**Raytheon Engineers &
Constructors**

Northrop Grumman

**Rockwell Aerospace
Rocketdyne**

Task No. 29

DOE/SF/20282-T4 REFERENCE
JUL 26 1995
JST.

ITER

Vacuum Vessel Design

(D201 Subtask 1.3 and Subtask 3)

Final Report

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MASTER

ITER/US/95/INV-VV-05

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DOE/SF/20282--T4

Acknowledgment

"This report is an account of work assigned to the U.S. Home Team under Task Agreement No. G 16 TD 33 within the Agreement among the European Atomic Energy Community, the Government of Japan, the Government of the Russian Federation, and the Government of the United States of America on Cooperation in the Engineering Design Activities for the International Thermonuclear Experimental Reactor ("ITER EDA Agreement") under the auspices of the International Atomic Agency (IAEA). The report has not been reviewed by the ITER Publications Office".

Disclaimer

"This report is an account of work undertaken within the framework of the ITER EDA Agreement. Neither the ITER Director, the Parties to the ITER Agreement, the U.S. DOE, the U.S. Home Team Leader, the U.S. Home Team, the IAEA or any agency thereof, or any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the parties to the ITER EDA Agreement, the IAEA or any agency thereof.

The views and opinions of authors expressed herein do not necessarily state or reflect those of the ITER Director, the Parties to the ITER Agreement, the U.S. DOE, the U.S. Home Team Leader, the U.S. Home Team, the IAEA or any agency thereof".

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MASTER

1000

Executive Summary

ITER Task Assignment, Vacuum Vessel Design (D201 Subtask 1.3 and Subtask 3), addresses local vacuum vessel stress concerns identified by the USHT (US Home Team) and the ITER Joint Central Team (JCT) at the Garching Joint Work Site. Task activities focus on identifying an effective design for the divertor port region which includes the highest stress intensities.

Existing stress analysis models and results for the vessel region in proximity to the divertor port were reviewed with the USHT stress analysts to obtain an overall understanding of the vessel response to the various applied loads. These reviews indicate that the reported stress intensities in the divertor port region are significantly affected by the loads applied to the vessel in adjacent regions. Consequently, Task D201 stress analyses include significant portions of the entire lower vessel region.

The industrial team of Raytheon Engineers & Constructors, Northrop-Grumman, and the Rocketdyne Division of Rockwell International Corporation has evaluated two stress reduction alternatives for the lower vessel region.

Alternative A proposes the use of a forged subassembly within the

local highly stressed regions. **Alternative B** proposes an approach that redistributes local stresses within the highly stressed region using increased shell and rib thickness' in combination with toroidal ribs. Design details, finite element stress analysis results, and an overview manufacturing assessment support the feasibility of each alternative.

The stress analysis models developed for the evaluation of Task D201 reinforcement alternatives are fine mesh local models. These models permit the identification of local stresses not available from the previously used coarser mesh models.

Stress analysis results indicate that both Alternatives A and B accomplish the intended objective of reducing stresses within the divertor port region.

The **Alternative A** subassembly concept appears to provide the maximum stress reduction potential. The use of a forging permits the load to transfer from the divertor port around to the vessel shell. Re-analysis with geometry refinements is recommended to reduce localized VDE stresses should this concept be acceptable to the JCT.

Alternative B represents a less radical deviation from the present JCT vessel design and also provides significant stress reduction benefits. The level of detail included within the Alternative B finite element model provides confidence in the accuracy of the analysis results. The primary region of concern in Alternative B is the port to vessel shell connection where very high local stresses are identified. The stress reduction benefits of local modifications such as curving the sides of the divertor port outward at the port to vessel connection and/or adjusting the rib geometry at this location should be considered for subsequent JCT evaluations.

The Alternative A and B stress analyses results provide a basis upon which to begin to evaluate connection reinforcement and detailed design. Further refinement of the reinforcement geometry to redistribute highly localized stresses and reanalysis are recommended as the detailed vessel design proceeds.

Vacuum Vessel Design
(D201 Subtask 1.3 and Subtask 3)
Final Report

Table of Contents

1. Introduction.....	1
1.1 Task Overview	1
1.2 Vacuum Vessel Description	2
2. Task Scope and Report Overview	3
3. Summary of Previous Structural Analysis Results.....	4
3.1 Model Description	4
3.2 Load Description and Stress Analysis Results	4
4. Divertor Port Stress Reduction Alternatives.....	10
<i>4.A Forged Subassembly Alternative.....</i>	<i>11</i>
4.A.1 Subassembly Description	12
4.A.2 Manufacturing Assessment.....	12
4.A.2.1 Discussion of Optional Manufacturing Processes	13
4.A.2.2 Industry Survey.....	14
4.A.2.3 Survey Results.....	15
4.A.2.4 Manufacturing Assessment Findings and Recommendations ...	15
4.A.3 Structural Analysis	18
4.A.3.1 Analysis Model Development.....	18
4.A.3.2 Local Model Description.....	19
4.A.3.3 Loading Conditions and Analysis Results.....	20
4.A.4 Summary and Conclusions.....	23
<i>4.B Lower Vessel Shell and Rib Reinforcement Alternative.....</i>	<i>24</i>
4.B.1 Reinforcement Description & Location.....	24
4.B.2 Manufacturing Assessment.....	25
4.B.3 Structural Analysis	26
4.B.3.1 Analysis Model Development.....	26
4.B.3.2 Local Model Description.....	27
4.B.3.3 Loading Conditions and Analysis Results.....	28
4.B.4 Summary and Conclusions.....	30
5. Task Summary and Recommendations	32
5.1 Manufacturing	32
5.2 Stress Results.....	34
5.3 Conclusion and Recommendations.....	36

**Appendix I: US Home Team Presentation at Vacuum Vessel
Workshop Naka October 30-November 2, 1995**

**Appendix II: Task 29 Vacuum Vessel Baseline Design Description
Document**

1. Introduction

1.1 Task Overview

ITER Task No. D201, Vacuum Vessel Design (Subtask 1.3 and Subtask 3), was initiated to propose and evaluate local vacuum vessel reinforcement alternatives in proximity to the Neutral Beam, Radial Mid-Plane, Top, and Divertor Ports. These areas were reported to be highly stressed regions based on the results of preliminary stress analyses performed by the USHT (US Home Team) and the ITER Joint Central Team (JCT) at the Garching JWS (Joint Work Site).

Initial design activities focused on the divertor port region which was reported to experience the highest stress intensities. Existing stress analysis models and results were reviewed with the USHT stress analysts to obtain an overall understanding of the vessel response to the various applied loads. These reviews indicated that the reported stress intensities in the divertor port region were significantly affected by the loads applied to the vessel in adjacent regions.

The port evaluation scope was subsequently modified and expanded (with the concurrence of the Technical Representatives of the USHT and the Garching JWS) to include the entire lower vessel region. This expanded evaluation added the attachment locations for the blanket, divertor, and the vessel gravity supports. Specific evaluations of the Top, Neutral Beam, and Mid-Plane Radial ports were eliminated with all parties agreeing that some subset of a suitable lower vessel region reinforcement design could potentially be adapted to other port regions if required.

The industrial team of Raytheon Engineers & Constructors, Northrop-Grumman, and the Rocketdyne Division of Rockwell International Corporation assessed two stress reduction reinforcement alternatives for the lower vessel region. One concept evaluated the use of a forged subassembly in the lower vessel region and a second evaluated the use of increased vessel shell and rib thickness' in combination with the addition of local toroidal ribs.

Task D201 design and stress analysis activities were coordinated with USHT efforts being performed by engineering personnel at the Oak Ridge National Laboratory (ORNL). Interim design and analysis results were summarized and included with the USHT material presented at the Vacuum Vessel Workshop held at the Naka Joint Work Site from October 30-November 2, 1995. The USHT presentation is provided as Appendix I of this report.

 

1.2 Vacuum Vessel Description

The ITER vacuum vessel, as the mechanical structure of the ITER machine, provides:

- an ultra-high vacuum boundary,
- structural support for all internal components, and
- the first safety confinement barrier for the ITER fusion reactor.

The vessel is designed to resist all credible loads without a loss of confinement. These include: gravity; pressure; electromagnetic forces due to plasma disruption; forces resulting from vertical displacement events; and other postulated event loads.

The vacuum vessel is a double walled toroidal shell structure joined by poloidal ribs. The volume between the inner and outer vessel shells is partially filled with stainless steel radiation shield plates. The shell wall interspace provides the flowpath and containment for the vessel coolant. Horizontal ports are located at the vessel midplane and in the lower vessel region. Vertical ports are located in the top vessel region.

The JCT has tentatively selected stainless steel Type 316LN for the vessel material. Engineering properties database for the Type 316LN wrought material are documented in the ITER Materials Properties Handbook.

Vacuum vessel fabrication, as presently described by the ITER Vacuum Vessel Design Description Document, will proceed with the initial manufacture of 40 fully welded vessel segments. Two segments will be subsequently assembled with a TF Coil to form 20 vessel segment/TF coil assemblies. The 20 vessel segment/TF coil assemblies will be transported to the machine floor for final machine assembly (Fig. 1.2-1 and Fig. 1.2-2).

A "Task 29 Vacuum Vessel Baseline Design Description Document" was prepared to describe the vessel design bases used for the Industrial Team's evaluations. (Task 29 is a US Home Team designation for the scope of work associated with ITER Task D201, Subtask 1.3 and Subtask 3.) A copy of this document is provided as Appendix II of this report. (The principal reference for the Task 29 Design Description Document was the Vacuum Vessel Design Description Document prepared by the JCT.)

Fig. 1.2-1

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

ASSEMBLY PLAN

SCALE: NTS

DL/IDR/ASSEMBLY_PLAN/01

ITER

DATE: 07-JUN-05
D. J. LIGHTOULERS

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

Fig. 1.2-2

DATE: 07-JUN-83
 D.J. LIGOTIUS
 WORKING PROPOSAL
 LEVEL : SKETCH
 GARCHING CO-CENTER

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VESSEL AND PORT FIELD JOINTS

SCALE: NTS DL/IDR/VESSEL AND PORT FIELD JOINTS/01

2. Task Scope and Report Overview

The scope of the design and analysis activities for ITER Task No. 29 (D201), focused on lower vessel reinforcement to reduce the reported high vessel stress intensities in that region. The reinforcement design proposed by the Task 29 Industrial Team has been developed to remain compatible with the fabrication and assembly requirements specified in the ITER JCT Vacuum Vessel Design Description Document. Reinforcement details also consider the vessel sector manufacturing proposals of the Japanese Home Team (JAHT) presented at the October 30 - November 2, 1995 Vacuum Vessel Workshop in Naka, Japan.

Two stress reduction alternatives developed and analyzed in this task are presented within this report. **Alternative A** proposes the use of a forged subassembly within the local highly stressed regions. **Alternative B** proposes an approach that redistributes local stresses within the highly stressed region using increased shell and rib thickness' in combination with toroidal ribs.

Alternative A - Forged Subassembly Alternative

Alternative A considers the use of a large, heavily reinforced subassembly. Concept drawings are presented. Results of an industry survey of suppliers is presented to demonstrate the feasibility of forging and machining such a large component. Preliminary finite element analyses results are provided for various loading conditions to assess the stress reduction benefits.

Alternative B - Lower Vessel Shell and Rib Reinforcement

Alternative B considers the use of increased shell and rib material thickness' over a broad region extending from the outboard blanket supports to the inboard blanket supports. Toroidal ribs are also provided at locations of high stress within the region. Concept drawings and preliminary finite element analysis results for various loading conditions are presented.

3. Summary of Previous Structural Analysis Results

3.1 Model Description

Multiple evaluations of the structural response of the vacuum vessel to various loads were performed by the USHT in conjunction with the JCT Garching JWS. Three finite element models of the vessel were developed and used for electromagnetic, static, and dynamic analyses (Fig. 3.1-1).

A 1.5° single rib model with a fine mesh was developed to evaluate the local bending effects due to coolant pressure and thermal gradients. An 18° single-wall composite shell model that included port extensions, vessel supports, and in-vessel components was developed to determine the transient response of the vessel and internals due to disruption forces. An additional 18° double-wall model was developed for performing static analyses and as the input geometry for three-dimensional electromagnetic analysis.

3.2 Load Description and Stress Analysis Results

The ITER vacuum vessel operating conditions and loads described within the JCT's Vacuum Vessel Design Description Document are summarized in Table 3.3-1 of this report. Stress analyses performed by the USHT evaluated several of these individual loads including gravity, coolant pressure, thermal gradients, plasma decay and disruption, and internal vessel pressure. Loading combinations were also analyzed for the following operating scenarios:

1. Normal Operation (operating state 4a - normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, and only gravity loads of in-vessel systems),
2. Off-normal Operation with Severe VDE (operating state 5a - normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems plus VDE loads),
3. Off-normal Operating with Pressure Pulse In-vessel (operating state 5c - normal operating temperatures and temperature gradients, normal vacuum in the cryostat, normal water pressure in the cooling channels, 0.5 MPa in the vessel interior, and gravity loads of in-vessel systems).

Figure 3.1-1: Vacuum Vessel Finite Element Models (USHT & JCT Analyses)

Table 3.3-1: Vacuum Vessel Loading Conditions

Operating State	Vessel Pressure (MPa)			Vessel Temperature (°C)		Blanket Temp. (°C)		Applied Loads (MN)			
	Internal	External	Coolant	Max Temp	Gradient	Avg Temp	Delta to VV	VV Wt.	Blkt Wt	Div Wt.	Other Loads
Fab/Assy											
- Pressure Test 1	0.10	0.10	2.75-3.44	20	0	n/a	n/a	~70	n/a	n/a	0
- Pressure Test 2	0.55	0.10	0.10	20	0	n/a	n/a	~70	n/a	n/a	0
Baking state	0.10	0.10	2.5	200	0	350	150	~70	~50	~15	0
Normal Plasma operating states											
- w/o disruption	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	0
- w/ disrup	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	~0.2 MPa Note 1
Off-normal operating states											
- Severe VDE	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	~150 MN Note 2
- TF coil quench	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	~0.4 MPa Note 3
- Pressure pulse in VV	0.5	0.0	2.0	200	5	250	50	~70	~50	~15	0
- Pressure pulse in cryostat	0.0	0.2	2.0	200	5	250	50	~70	~50	~15	0
- LOFA	0.0	0.0	2.0	200 - 300	5	550	350	~70	~50	~15	0
- LOCA	0.10	0.10	0.1-2.0 Note 5	~450 Note 6	5	550	350	~70	~50	~15	0
- Design basis earthquake, (level SL-2)	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	TBD
- Design basis earthquake, (level SL-1)	0.0	0.0	2.0	200	5	250	50	~70	~50	~15	0.2 g vert, 0.3 g horiz
Maintenance state	0.1	0.0	2.0	20	0	20	0	~70	0-50	0-15	TBD Note 4

Notes:

1. Load estimate due to centered disruption
2. Load estimate due to slow VDE with halo current
3. Load estimate due to toroidal field coil quench
4. Loads due to RH equipment and operations are TBD
5. One vessel cooling system is at 0.1 MPa and the other is at 2.0 MPa
6. Rough estimate of the vessel temperature. Value is TBD.

The electromagnetic loads used in the USHT analyses were obtained from the Tokamak Simulation Code (TSC) Case 940408H of a "slow" VDE simulation. In this simulation the plasma drifts vertically downward prior to the thermal quench. After the thermal quench, the plasma current decays from 25 MA to zero in about 400 milliseconds. The TSC computed loads were subsequently scaled to reflect the lower initial plasma current of 21 MA that is specified in the ITER General Design Requirements Document (GDRD).

The electromagnetic load distribution is spatial and temporal, and the peak loading of the blanket, vacuum vessel, and divertor do not occur simultaneously. To permit static analyses to be performed, the electromagnetic loads at 1.642 seconds after the start of the plasma drift phase were selected. This single point in time corresponds to the peak vacuum vessel stress taken from the dynamic analysis.

Results obtained from the USHT stress analyses were presented at Vacuum Vessel Workshops held in May 1995 at the Garching JWS and in October 1995 at the Naka JWS. The results reported at the May 1995 Garching JWS workshop are included in the ITER Vacuum Vessel Design Description Document.

Results Reported at the May '95 Vacuum Vessel Workshop (Garching)

The stress analysis results presented in May are summarized as follows:

- Case 1 (Normal Operation), vessel deformation was computed to be approximately 6-mm with stress levels at 75-MPa (50-MPa membrane). Maximum stresses up to 160-MPa were computed near the vessel support.
- Case 2, (Off-normal Operation with a Severe VDE), deformations increased to 10-15 mm. Resulting stresses in the support region increased to about 175-MPa (100-MPa membrane). A stress concentration of up to 260-MPa was found around the square corners of the divertor port.
- Case 3, (Off-normal Operating with Pressure Pulse), conditions resulted in a maximum displacement of 4-mm in the torus with a maximum stress intensity of 130-MPa (90-MPa membrane). Maximum stresses of 290-MPa were computed at the divertor port corners.

Stress analysis results for all three of the combined load cases are shown in the appropriate column of Table 3.3-2. The results of an additional loading combination case denoted as Operation with Disruption with "est. correction" are also presented in this Table. This case is a postulated condition of "Off-

normal Operation with 2 * Severe VDE, where an estimated correction factor of "2" is applied only to the electromagnetic loads from a "slow" VDE (i.e. not to gravity or pressure loads). This correction serves as a temporary approximation of a more severe loading condition until such a condition can be defined and analyzed more rigorously. The correction factor is intended to address:

1. a potential increase in the halo current from a value of about 6MA used in the "slow" VDE simulation, to the value of 8.4 MA specified in the GDRD;
2. toroidal asymmetry of the plasma forces by up to a factor of 2; and
3. vessel dynamic effects.

The selection of the magnitude of the correction factor considers that the halo forces account for about half of the total electromagnetic force, hence a factor of 1.2 was applied for the halo current correction. A factor of 1.5 was estimated to account toroidal asymmetry and a correction factor of 1.15 (based on the results of dynamic analysis) was used to address vessel dynamic effects. The combination of these individual factors results in a total estimated correction factor of ~ 2 (i.e. $1.2 \times 1.5 \times 1.15 = \sim 2$).

Review of the tabulated results indicates that the maximum vacuum vessel stresses for the Operation with Disruption case approach allowable stress limits. Inclusion of the estimated "correction factor" for disruption forces results in vessel stresses that exceed the allowable material stress limits.

Results Reported at the October '95 Vacuum Vessel Workshop (Naka)

Results of stress analyses performed subsequent to the May 1995 Garching Workshop were reported at the October 1995 Naka Workshop. These analyses are based on the double wall plate model of the vessel with some model modifications including:

- a minor support relocation from 7.3m to 7.7m from the vessel centerline;
- blanket support stiffness' consistent with the flexible plate blanket support concept; and
- VDE loads on the vacuum vessel applied as distributed loads. Figure 3.1-2 shows the variation in the application of the VDE forces.

Results from these stress analyses indicate that:

- Case 1 (Normal Operation) vessel deformation was computed to be approximately 7-mm at the end of the divertor port. Maximum stresses were 118-MPa (membrane) and 133-MPa (membrane + bending) at the top corner of the divertor duct.

Figure 3.1-2: Application of Vacuum Vessel Forces in Previous USHT Analyses

- Case 2 (Off-normal Operation with a Severe VDE) reported increased deformations to 9-mm (12-mm from the dynamic analysis) at the end of the divertor port. The maximum membrane stress was 141-MPa. Maximum membrane + bending stresses of 159-MPa were computed at the corner of the divertor port and 181-MPa near the vertical support/divertor port.
- Case 3, (Off-normal Operating with Pressure Pulse), conditions resulted in a maximum deformation of 10-mm in the port duct and 7-mm in the torus. Maximum stress intensities of 204-MPa membrane and 228-MPa membrane + bending were computed at the upper, outer corner of the divertor port.

Stress analysis results for all three of the combined load cases as well as the additional "correction factor" combination case (Off-normal Operation with 2 * Severe VDE) are shown in the appropriate column within Table 3.3-2.

Review of the tabulated results indicates that the maximum vacuum vessel stresses approach or exceed the allowable stress limits for the case which includes the correction factor for disruption forces.

**Table 3.3-2: Vacuum Vessel Reported Stresses and Allowables
from Previous Analyses**

Load Case	Results Presented in May 1995 (Garching)			Results from analyses between May 1995 and October 1995 (Naka) ⁴		Allowable Stresses @ 200C	
	Mem-brane MPa	Mem-brane+ Bending MPa	Prim + Secndry MPa	Mem-brane MPa	Mem-brane + Bending MPa	Prim Membr MPa	Prim Membr+ Bending MPa
Normal Operation (gravity + coolant pressure)	50	75	160	118	133	131	196.5
Operation with Disruption (gravity + coolant pressure + slow VDE, symmetrically applied)	100	175	260	141	181	131	196.5
Operation with Disruption with est. correction (gravity + coolant press. + 2* slow VDE)	150	275	360	198	267	131	196.5
Off-Normal (gravity + coolant pressure + internal pressure of 0.5 MPa)	90	130	290	204	228	157.2	235.8

* correction = 2 x VDE stress

Notes:

1. The design criteria used to determine the allowable stress values is defined in section 2.2.3 of the ITER Design Description Document
2. The S_m , S_y , and S_u values used in this table are from the ASME Code Section III.
3. Material properties are for SS 316 LN.
4. Some stresses reported at the Naka meeting were found to be in error due to a bug in the postprocessing software that incorrectly calculated Tresca stress. The correct values are reported in this table and subsequent tables provided in this report

4.A Forged Subassembly Alternative

The forged subassembly alternative represents a significant deviation from the present vacuum vessel shell and rib design. This alternative seeks to reduce the magnitude of the stresses in the vicinity of the divertor port and vertical vessel support by maximizing the material mass within these regions. A schematic representation depicting the use of forged subassemblies at all port locations is shown in Figure 4.A-1.

Figure 4.A - 1: Vessel with Forged Subassemblies

Although the principle feature of a forged subassembly option is the maximization of material mass within the most highly stressed vessel regions, additional benefits include:

- elimination of subassembly welding and associated concerns (primarily distortion, tolerance control, and NDE) within a region which is characterized by significant changes in geometry,
- elimination of the additional supplementary shield plates within the region, since sufficient material would remain after post-machining operations to satisfy the vessel shielding requirements,
- flexibility of geometry to eliminate stress risers and redistribute local stresses afforded by the mandatory machining,

- flexibility in manufacturing as the use of a subassembly permits the parallel manufacture of significant portions of the vessel segment.

Other potential subassembly options, such as weldments or densified castings, are discussed in subsequent report sections. Although such options may be feasible, a forged subassembly was selected for initial evaluation, since in addition to maximizing the mass in the divertor port region, stress analysis indicates that its use provides substantial stress reduction benefits.

4.A.1 Subassembly Description

Various conceptual design sketches depicting the use of a forged divertor port subassembly are listed in Table 4.A-1. Copies of these drawings are included at the end of this report section.

The forged subassembly shown in SK 29-A-1001 provides a totally reinforced region around the divertor port itself that extends toward the base and beyond the region of the vertical vessel support. The pre-machined weight of the forged component is estimated at 40 metric tons.

Post-forging machining operations would be required to achieve the final desired configuration including the weld end preparations in the regions where the subassembly would be joined to the vessel shell. Cooling passages of considerable length, would be required to penetrate the most massive region of the component (SK 29-A-1002).

4.A.2 Manufacturing Assessment

The configuration and size of the large forged component required for the ITER vacuum vessel introduced the issue of manufacturing feasibility. A manufacturing assessment consisting primarily of an industry survey of suppliers was undertaken to establish a degree of confidence that such a component could be produced. Results of this survey are presented in the following sections.

Additional manufacturing considerations regarding the interface and compatibility of the forged component with the remainder of the vessel shell and rib fabrication were also reviewed. Particular concerns included the transition of the thick forged component with the thinner shell and rib plates and the compatibility of the forged subassembly with the layout of the shell facets.

In order to address these concerns, the forged subassembly conceptual design presented in this report includes machined transition sections at the shell and rib interfaces to permit the welding of similar thickness material. Additionally, the rib and shell facet arrangement shown in SK 29-A-1005 and SK 29-A-1006 was selected from various developed alternatives as it avoided a poloidal facet weld interface with the forged component.

Table 4.A-1: Forged Subassembly Design Sketches

Sketch No.	Date	Sketch Title
SK 29-A-1000	10/25/95	Task 29: Alternative A - Sketch Summary
SK 29-A-1001	10/25/95	Divertor Port Forging
SK 29-A-1002	10/25/95	Divertor Port Forging with Machined Surfaces
SK 29-A-1003	10/25/95	18° Segment with Forged Subassemblies at Ports
SK 29-A-1004	10/25/95	(2) 9° Vessel Sectors with Forged Subassemblies
SK 29-A-1005	10/25/95	Plan View - 18° Facet Arrangement
SK 29-A-1006	10/25/95	18° Segment Plan - Forged Subassembly Interface with Vessel Facets
SK 29-A-1007	10/25/95	Divertor Port Weldment Subassembly

4.A.2.1 Discussion of Optional Manufacturing Processes

Advantages and disadvantages of three manufacturing options considered for an Alternative A subassembly are summarized in Table 4.A-2. For this assessment overview, properties and manufacturing comparisons are general. A comparative cost analysis was not performed within the scope of this task.

Forging

As previously stated a forging allows maximum mass in the region of highest stress and compared to a weldment, eliminates fixtures, fabrication operations, and associated distortion concerns. In addition, as a wrought product form, it closely matches the shell plate material in engineering (mechanical and physical) and fabrication properties (machining, weldability, etc.).

Issues associated with the forged subassembly option include:

- handling of the heavy component during forging operations and
- the need for considerable machining to achieve the final configuration and to provide cooling passages, since the forging cannot be produced to a near net shape.

Weldment

A weldment subassembly fabricated from plate product possesses well characterized mechanical and physical properties and a developed fabricability (weldability, machinability) database. Benefits include ongoing welding development, established manufacturing methods, and a more manageable part size. A major disadvantage may be the potentially high cost of manufacturing.

A welded subassembly as depicted in SK 29-A-1007 is characterized by thick sections and may have manufacturing limitations. Fabrication is complicated by numerous ultra thick section (75 to 100 mm) welds, related machining, fitup and NDE operations and distortion control concerns. The configured weldment, although similar to a typical double wall shell section structure is considered significantly more difficult to produce due to its greater mass. Additional design refinements and stress analysis iterations may however, justify reduced plate thickness' that could result in a mass reduction.

Densified Casting

A densified casting is considered to represent the least acceptable subassembly option due primarily to difference in product form as compared to the wrought shell plate material of construction and the associated variability in engineering properties. In addition, a single casting which eliminates welding and inspection operations is most desirable however, it is not possible due to limited densification chamber size. The need to weld a segmented subassembly diminishes potential producibility advantages of a densified casting.

4.A.2.2 Industry Survey

A preliminary industry survey was conducted to determine the feasibility of producing large scale forgings. Ten forging suppliers were contacted. Each was requested to review an inquiry package describing material, size configuration and quantity with regard to producing the component as a single piece forging.

The need for long cooling passages, as well as other machining requirements, was considered a possible limitation. Three suppliers were contacted to determine machining feasibility.

4.A.2.3 Survey Results

Industry survey results are summarized in Table 4.A-3. Eight of ten forging suppliers lacked adequate handling or forging capacity, or both, to produce a single piece forging. Beth Forge possessed the necessary handling and forging capability. Creusot-Marrel was confirmed to possess melting and ingot size capabilities and near-forging capabilities.

Alternative multi-piece forgings are considered feasible and would certainly expand the field of suppliers. Also, information received from suppliers indicated other sources exist for both melting and forging capacity to produce a single piece component in Europe and Japan. If required these can be confirmed by further inquiry at some future date.

In the more limited machining feasibility study, three suppliers were contacted. One supplier reported that the machining requirements exceeded their capabilities. Precision Components Corp. reported that they have existing capabilities however, further study to confirm the feasibility of the ITER application will be required. Remmele Engineering reported that they possess both the capacity and experience to perform the required machining.

4.A.2.4 Manufacturing Assessment Findings and Recommendations

A forging or weldment is preferred to a densified casting since both possess engineering properties similar to or the same as the vessel construction material. Weldments may be difficult and costly to fabricate due to the anticipated multiple complex thick section assembly welds and required inspections. Distortion control issues are also a concern.

A forging alternative offers potential producibility benefits. Based upon the results of the preliminary manufacturing assessment, industry capability exists to produce a forged subassembly. The concept as presented however, places it near the limit of manufacturability in terms of weight and scale. Additional structural analyses to reduce the component scale to the extent possible will be required. Such analyses followed by a definitive manufacturing reassessment and cost estimate are recommended should the JCT wish to pursue this alternative.

Table 4.A-2: Manufacturing Processes Considered for an Alternative A Subassembly ¹

Process	Process Advantage	Process Disadvantage
Forging	<ul style="list-style-type: none"> • Engineering properties similar to vessel construction material • Potential manufacturing benefit compared to a welded structure 	<ul style="list-style-type: none"> • Forging and machining database not fully developed. • Properties in thickest sections need to be characterized. • Limited number of qualified suppliers. • Forging toroidal resistance may be lower than built up plate section
Weldment	<ul style="list-style-type: none"> • Engineering properties (mechanical, physical, fabrication) same as vessel material. • Fabrication experience benefits from Full Section Model Fabrication Study. • Most manageable in process fabrication part size. 	<ul style="list-style-type: none"> • Potentially high fabrication costs. • Multiple ultra thick section assembly welds • Distortion control issue.
Densified Casting	<ul style="list-style-type: none"> • Potentially lowest manufacturing cost if single piece. • Manufacturing database benefits from ongoing divertor internals casting and welding development. 	<ul style="list-style-type: none"> • Engineering properties differ from vessel construction material • Densification chamber size prohibits manufacture of single piece casting. • Material properties need to be characterized. • Limited number of qualified suppliers.

Note:

1. The Alternative A concept regardless of the selected subassembly manufacturing process, is generally considered more amenable to vessel fabrication by poloidal segmentation rather than by the built up fabrication described in the ITER Vacuum Vessel DDD. Depending upon the actual vessel fabrication approach selected by the ITER JCT, this consideration may either be a process advantage or disadvantage.

Table 4.A-3: Divertor Port Industry Survey of Forging and Machining Capabilities

FORGING SUPPLIER	LOCATION	SURVEY RESULT
Carlton Forge Works	Paramount, CA	Insufficient Capacity
Press Forge Co.	Paramount, CA	
Schlosser Forge	Cucamonga, CA	
Wyman Gordon	North Grafton, MA	
Jorgensen	Seattle, WA	
Ladish	Cudahy, WI	
Erie Forge	Erie, PA	
McInnes Steel	Corry, PA	
Creusot-Marrel	Wayne, PA	Melting Capability
Beth Forge	Bethlehem, PA	Forging Capability

MACHINING SUPPLIER	LOCATION	SURVEY RESULT
Dorries Scharmann	Erie, PA	Insufficient Capacity
Precision Components Corp.	York, PA	Capacity Exists Feasibility Study Req'd.
Remmele Engineering	St. Paul, MN	Capability Exists

4.A.3 Structural Analysis

Structural analyses to determine concept feasibility were performed for the lower vessel region that incorporated the forged subassembly. These preliminary analyses results were presented at the October 31-November 2, 1995, Vacuum Vessel Workshop in Naka, Japan.

More refined analyses to support this alternative were anticipated to be performed subsequent to the Naka Workshop however, these additional analyses and associated concept refinements were not pursued. The evaluation of a different lower vessel shell and rib reinforcement concept was initiated as it could be more readily incorporated into the design of the prototype vessel segments being prepared for manufacture by the JAHT.

Various design details associated with the subassembly concept were not included in Alternative A stress analysis model. These details are identified below.

- Blanket supports are not fully detailed. The analyses performed for this evaluation models the inboard and outboard blanket supports using beam elements.
- Divertor rails (similar to the blanket supports) are not fully detailed. These rails are represented with beam elements.
- Details of the forged subassembly such as machined transitions to accommodate welding to the vessel shells and ribs, as well as the machined cooling passages are not included to simplify modeling.
- Divertor port extension ribs are not detailed to simplify and reduce the initial model size.
- Inner shell volume shield plate weight was conservatively applied to the ribs to account for the additional mass.

4.A.3.1 Analysis Model Development

Local Model

The finite element code ANSYS was used for the Task 29 stress analyses. The local model of the lower vessel region used ANSYS capabilities to directly transform an IGES translation of CADD geometrical solids to ANSYS solid entities. The complicated solids were then meshed with tetrahedron elements within ANSYS to construct the finite element model. In order to

improve the quality of the analysis results, higher order 10 node tetrahedrons (4 corner nodes and 6 midside nodes along each of the 6 edges) were used exclusively. All nodes provide 3 translational degrees of freedom. The mid-side nodes allow for parabolic displacements along their edges and permit the recovery of stress variations within the element.

Global Model

To appropriately model the boundary conditions for the local model, an additional coarse global ANSYS model (Fig. 4.A.3-1) was developed and analyzed for the various load conditions developed by ORNL. Model geometry was obtained directly from an MSC PATRAN translation of the USHT model provided by ORNL and therefore generally includes the same nodes and elements that were used in the ORNL model. (The models are not identical as some differences between the ABAQUS (used at ORNL) and ANSYS analysis codes required minor modification of the composite elements used in the port regions.)

The coarse model analysis results provided appropriate boundary conditions along the "cut boundary planes" of the local model. (These "cut boundary planes" identified in Fig. 4.A.3-1 were specifically chosen to minimize boundary condition effects and also to keep the size of the model manageable.) The ANSYS "shell to solid cut boundary converter" permitted the vessel response displacements to be extracted from the global model analysis results and imposed at the local model boundaries. This feature converted the computed 3 translations and 3 rotations of the coarse global shell model to 3 translations for input to the local solid model.

4.A.3.2 Local Model Description

Figure 4.A.3-2 shows the ANSYS solid model. This model contains 18 ribs on the outboard side and 8 ribs on the inboard side. The face sheets and ribs are 40 mm thick. As previously noted the forging is modeled as a solid without machined features (see Figure 4.A.3-3). The vertical extension of the local model was taken far enough from the divertor port such that loads and boundary displacements from the global ANSYS model could be mapped onto the local model with minimum effects.

The ground supports as shown in Figure 4.A.3-4 were taken from the USHT global model and appended to the local finite element model. These supports are connected with beam elements and allow rotation about an axis consistent with the support representation in the global model.

1

Global Model Of Vacuum Vessel

ANSYS 5.2
JAN 5 1996
15:48:10
ELEMENTS
REAL NUM

XV =-1
YV =1
ZV =1
DIST=14061
XF =9458
ZF =-3399
VUP =Z
Z-BUFFER

Figure 4.A.3 - 1: ANSYS Global Model

Figure 4.A.3 - 2: Alternative A - ANSYS Solid Model

ANSYS 5.2
DEC 20 1995
08:20:54
VOLUMES
TYPE NUM

XV =-1
YV =1
ZV =1
DIST=4057
XF =10415
ZF =-4945
VUP =Z
Z-BUFFER

Figure 4.A.3 - 3: ANSYS Solid Model of Forging

Figure 4.A.3 - 4: Alternative A - Finite Element Model with Ground Supports

Repeatable toroidal edges of the local model segment were coupled with the degrees of freedom at these edges (rotated into the machine cylindrical system) and forced to have the same displacements. At the Z equals zero location, the surface boundary is constrained by the results of the global model. The configuration of the finite element model consisting of over 70,000 high order tetrahedron elements is shown in Figure 4.A.3-5.

The local model includes the weight of the radiation shielding between the vessel shells by increasing the density of the ribs. For the forged subassembly alternative, model element properties used an average density increase to account for the weight of the radiation shield plates.

4.A.3.3 Loading Conditions and Analysis Results

The finite element analyses performed for the Task 29 reinforcement alternatives used the same applied loadings as the USHT analyses results presented at the October 1995 Naka workshop. Re-evaluation or re-computation of the magnitudes of these applied loadings was not performed, therefore permitting a comparison of stress intensity results between the previously performed analysis and the Task 29 results.

Analyses of five individual load conditions were performed. These load conditions included:

- Vacuum Vessel Gravity
- Blanket Gravity
- Coolant Pressure (2.2-MPa)
- VDE
- Internal Pressure (0.5-MPa)

The analysis results of the individual load conditions were subsequently combined to provide the four load combinations previously reported by the USHT and the JCT (see Table 3.3-1 in Section 3.3 of this report). The use of these load combinations permits the comparison of the Task 29 local model results with the results previously reported by the USHT and the JCT. The four load combinations are as follows:

- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure
- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE
- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE

- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure (0.5-MPa)

Figure 4.A.3-6 shows the results of the Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure load combination. The weight of the lower divertor port and the fact that it extends away from the vacuum vessel adds a moment into the shell face sheets. This load is taken in hoop through the edges of the shell faces.

The outer face sheet Tresca stresses for this combination are shown in Figure 4.A.3-7. Note that in both Figures 4.A.3-6 and 4.A.3-7, the inner and outer face sheet maximum Tresca stresses are located near the corner of the forging. This is a result of the significant change in stiffness and most probably can be reduced by incorporating a machined transition into the forging design.

Figure 4.A.3-8 shows the results of the second load case combination (Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE). The very local maximum Tresca stresses shown on this figure near the inner divertor rail are suspect due to the use of beam elements to represent the inner rail. The modeling of a detailed inner rail attachment to the vessel should significantly reduce the maximum reported stresses. The stresses between the inner and outer divertor rails are probably realistic as this region is located a significant distance from the divertor rails and therefore not highly affected by the absence of detail.

Figure 4.A.3-9 shows the results for the postulated case of Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE loads. Qualitatively this figure shows similar results as the previous case with maximum Tresca stresses reported in the region of the divertor rail attachments.

Figure 4.A.3-10 shows the results for the Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure (0.5 MPa) load combination. These results show that similar to the combination case of gravity and coolant pressure, the large relative stiffness of the forging results in high local stresses in the inner and outer face sheets. The addition of a machined radius at the forging edge to transition down to the thickness of the shell face sheets should alleviate some of these local stresses near the forging corner.

Table 4.A-4 compares Task 29 Alternative A stress results with the USHT stress analysis results reported at the Naka Workshop in late October 1995. This comparison shows the relative reduction in maximum Tresca stress associated with this alternative. Additional stress comparisons and discussions associated with these comparisons are provided in **Section 5. Task Summary and Recommendations** of this report

Figure 4.A.3 - 5: Alternative A - Finite Element Model of Solid Elements

Figure 4.A.3 - 6: Alternative A - Inner Shell Tresca Stresses- Operation

ANSYS 5.2
 NOV 28 1995
 12:04:19
 PLOT NO. 1
 54.341
 14000
 28000
 42000
 56000
 70000
 84000
 98000
 112000
 126000

Figure 4.A.3 - 7: Alternative A - Outer Shell Tresca Stresses - Operation

Figure 4.A.3 - 8: Alternative A - Inner Shell Tresca Stresses - VDE

Figure 4.A.3 - 9: Alternative A - Inner Shell Tresca Stresses - 2*VDE

Figure 4.A.3 - 10: Alternative A - Inner Shell Tresca Stresses - Internal Pressure

Table 4.A-4: Vacuum Vessel Stress Comparison

	USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)		Detailed Model with Reinforcement (Task 29 Alternative A)
Load Case	Mem-brane (MPa)	Mem-brane + Bending (MPa)	Membrane + Bending ¹ (MPa)
Normal Operation Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure	118 (port corner)	133 (port corner)	94 (port corner) 132 (port centerline)
Operation with Disruption Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE	141	181 (port/supp interface)	108 (port corner) 112 (port centerline)
Operation with Disruption <i>With est. correction*</i> Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE	198 (div port interface)	267 (div port interface)	130 ² (inner shell)
Off-Normal Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure	204 (port corner)	228 (port corner)	176 (port corner) 178 ² (port centerline)

Notes:

1. The stress values reported in this column are the computed Tresca stresses obtained directly from the finite element analysis results. These stresses represent a total stress intensity (similar to that defined within the ITER Interim Structural Design Criteria - IRB 3224.5.4. These stresses have not been classified in primary, secondary, or peak categories.

2. This reported stress is less than the maximum value indicated in the finite element analysis results, as the maximum stress is considered unrealistic and a direct result of the absence of finite element model detail. A more representative stress has been substituted for the maximum computed stress for comparison with previous analysis results.

4.A.4 Summary and Conclusions

The manufacturing assessment indicates that it is feasible and potentially beneficial to use a forged subassembly in the region of the divertor port and vertical support. The forging alternative also provides integral radiation shielding, thereby eliminating the fabrication costs and any manufacturing feasibility concerns associated with the installation of separate shielding components.

Furthermore, preliminary stress analysis results indicate that beneficial stress reductions can be realized with a forged subassembly. Even with the absence of the variously noted design details, the maximum computed stresses obtained from the detailed local model are generally lower than the previous analysis results.

Maximum calculated Tresca stresses, where significant, appear highly localized and can be expected to be reduced with minor design modifications and re-analysis. The inclusion of a radius around the forging perimeter which tapers down to the thickness of the shell face sheet and increasing the local face sheet thickness between the divertor rails should provide added stress reduction benefits.

It was not possible to perform subassembly cost-benefit analyses within the scope of the evaluations. Such an assessment will be necessary if the JCT elects to seriously pursue further evaluation of this alternative.

ITER TASK 29
VACUUM VESSEL DESIGN (D201)
REINFORCEMENT ALTERNATIVE A
VACUUM VESSEL SKETCH SUMMARY

SK-29-A-1001
SK-29-A-1002
SK-29-A-1003
SK-29-A-1004
SK-29-A-1005
SK-29-A-1006
SK-29-A-1007

DIVERTOR PORT FORGING
DIVERTOR PORT WITH MACHINED SURFACES
18° SEGMENT WITH 3 SUBASSEMBLIES AT PORTS
(2) 9° VESSEL SECTORS WITH FORGED SUBASSEMBLIES
PLAN VIEW 18° FACET ARRANGEMENT
18° SEGMENT PLAN AT DIVERTOR PORT
DIVERTOR PORT WELDMENT SUBASSEMBLY

VOLUME = 5.140931E +09 CUMM
MASS = 39.7 MTONS

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN (02011)
REINFORCEMENT ATTACHMENT A
10, 11, 12

DIVERTOR PORT WITH MACHINED SURFACES

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN 102011
REINFORCEMENT ALTERNATIVE A
1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27, 28, 29, 30, 31, 32, 33, 34, 35, 36, 37, 38, 39, 40, 41, 42, 43, 44, 45, 46, 47, 48, 49, 50, 51, 52, 53, 54, 55, 56, 57, 58, 59, 60, 61, 62, 63, 64, 65, 66, 67, 68, 69, 70, 71, 72, 73, 74, 75, 76, 77, 78, 79, 80, 81, 82, 83, 84, 85, 86, 87, 88, 89, 90, 91, 92, 93, 94, 95, 96, 97, 98, 99, 100

18° SEGMENT WITH 3 SUBASSEMBLIES AT PORTS

RAYTHEON ENGINEERS AND CONSTRUCTORS	
ITER TASK 29	
VACUUM VESSEL DESIGN ID2011	
REINFORCEMENT ALTERNATIVE A	
SK-29-A-1001	10 25 95

(2) 9° VESSEL SECTORS WITH FORGED SUBASSEMBLIES

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN (D2011)
REINFORCEMENT ALTERNATIVE A
SK-29-A-1004 10-25-95

PLAN VIEW
 VESSEL CUT AT Z = 0°
 18° FACET ARRANGEMENT

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN 102101
REINFORCEMENT ALTERNATIVE A
FIG 29 A 100' 10 21 9'

18° SEGMENT PLAN AT DIVERTOR PORT
FORGED SUBASSEMBLY INTERFACE WITH VESSEL FACETS

RAYTHEON ENGINEERS AND CONSTRUCTORS	
ITER TASK 29	
VACUUM VESSEL DESIGN ID2011	
REINFORCEMENT ALTERNATIVE A	
SK-29-A-1006	10-25-95

DIVERTOR PORT WELDMENT SUBASSEMBLY

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN (D210)
REINFORCEMENT ALTERNATIVE A
SK-29-A-1007 10-25-95

4.B Lower Vessel Shell and Rib Reinforcement Alternative

This alternative seeks to redistribute the stresses within the highly stressed lower vessel region by increasing the shell and rib thickness' of the existing JCT vessel design and adding local toroidal stiffening ribs. The inclusion of local toroidal ribs in combination with the increased inner shell thickness redistributes the stresses in the regions adjacent to the blanket and divertor supports. Similarly, these local toroidal ribs combined with the increased outer shell thickness redistributes the stresses within the region of the vertical vessel support.

A schematic of the region of increased reinforcement is shown in Figure 4.B-1. Conceptual sketches developed for the shell and rib reinforcement alternative are listed in Table 4.B-1. Copies of these drawings are included at the end of this report section.

4.B.1 Reinforcement Description & Location

The region of shell and rib reinforcement is depicted in SK 29-B-2001 and SK 29-B-2002. It extends from approximately 18 cm above the top of outboard blanket support downward beyond the region of the vertical vessel support to approximately 45 cm above the top of the inboard blanket support. Within

this region, shell and rib thickness' are increased from 40-mm to 60-mm at all locations except for the weld joint splice plates. The splice plate thickness' remain at 40-mm based upon the JAHT Naka workshop presentations attended by the USHT Technical Representative.

Ten pairs of toroidal stiffening ribs are located between the poloidal ribs as shown in SK 29-B-2003. In order to permit coolant water flow these ribs are welded to the inner shell and extend only to mid-depth within the volume located between shells (SK 29-B-2004).

Table 4.B-1: Alternative B Design Sketches

Sketch No.	Date	Sketch Title
SK 29-B-2000	11/28/95	Task 29 Alternative B - Sketch Summary
SK 29-B-2001	11/28/95	18° Reinforced Lower Vessel Region
SK 29-B-2002	11/28/95	Lower Vessel Elevation
SK 29-B-2003	11/28/95	Isometric View - Rib Arrangement
SK 29-B-2004	11/28/95	Rib Layout Plan View
SK 29-B-2005	11/28/95	Vessel and Divertor Port Rib Arrangement
SK 29-B-2006	11/28/95	Plan View - 18° Facet Arrangement
SK 29-B-2007	11/28/95	Poloidal Rib Volumes
SK 29-B-2008	11/28/95	Poloidal Volumes Between Ribs
SK 29-B-2009	11/28/95	18° Full Vessel Segment

4.B.2 Manufacturing Assessment

The shell and rib reinforcement alternative which increases the shell and rib thickness' in the lower vessel region by 50%, remains very similar to the existing JCT design. Issues associated with fixturing, fabrication, and handling of the thicker materials are considered to be within the existing database of manufacturing experience.

The increase in weld deposit volume for the thicker material increases both costs (welding and inspection) and the potential for weld induced distortion.

Distortion control should benefit however, by the present plan to manufacture and weld the thicker materials in the shop rather than in the field (assuming that the long poloidal weld joint splices between segments remain at the specified 40-mm thickness). Additionally, all fabrication operations can be expected to benefit from the ongoing ITER manufacturing process and weld development activities associated with the JCT specified 316LN plate product.

4.B.3 Structural Analysis

Structural analyses were performed for the lower vessel region with increased shell and rib thickness'. Task schedule and resources permitted analyses with only a single geometry, consequently analysis iterations with design refinements were not performed.

The Alternative B stress analysis does incorporate various vessel geometry and analysis refinements not included in the previous Alternative A analyses. These are:

- Detailed blanket and divertor rail supports that provide a more realistic stress distribution of the applied concentrated forces. Concept details obtained from the Garching JWS were incorporated into both the CADD and analysis models.
- Ribs located within the local inner and outer faces of the divertor port extension have been included in the models. The rib arrangement is very conceptual and does not fully consider all necessary requirements for coolant flow. The stress results however, provide design feedback that may be beneficial to future more detailed design initiatives.
- Inner shell volume shield plate weight was more accurately applied to the ribs to account for the additional mass. SK 29-B-2007 and 2008 were prepared to identify the computed volumes between ribs where the radiation shield plates will be installed.

These geometry refinements improve the interpretation of the local stresses.

4.B.3.1 Analysis Model Development

Local and Global Models

The same technique described in Section 4.A.3.1 was used to develop the Alternative B stress analysis model. To briefly reiterate, ANSYS features were used to directly transform an IGES translation of CADD geometrical

solids to ANSYS solid entities. These ANSYS solids were subsequently meshed with 10 node tetrahedron elements to construct the local finite element model.

The global model also described in Section 4.A.3.1 was modified to reflect 60-mm change in shell and rib thickness'. Analysis results using this model provided the appropriate boundary conditions along the "cut boundary planes" of the local model.

4.B.3.2 Local Model Description

Figure 4.B.3-1 shows the ANSYS solid model. Figure 4.B.3-2 shows a solid view of the inner face sheet, the inner divertor port shell and the vessel/port ribs. This model contains 14 vessel ribs on the outboard side and 6 vessel ribs on the inboard side. The inner and outer vessel shell face sheets are 60-mm thick except at field joint locations which are 40-mm. The divertor port extension inner and outer shell thickness' are both 40 mm. Vessel poloidal and toroidal ribs are 60-mm thick. Divertor port ribs are 40-mm in thickness. The vertical extension of the local model was taken far enough from the divertor port such that loads and boundary displacements from the global ANSYS model could be mapped onto the local model with minimum effects.

The lines on the coolant side of the inner shell shown in Figure 4.B.3-3 are generated as part of the model solid entity "sewing" process which connects the ribs to the sheets. These lines are actually common lines shared between the solids. These lines can be seen in the finite element results and help in relating the stresses to the geometry.

The vessel ground supports are the same used for the Alternative A model and described in Section 4.A.3.2. These supports are connected with beam elements and allow rotation about an axis consistent with the support representation in the global model.

Repeatable toroidal edges of the local model segment were coupled with the degrees of freedom at these edges (rotated into the machine cylindrical system) and forced to have the same displacements. At the model locations above the inboard and outboard blanket supports, the surface boundary is constrained by the results of the global model. The configuration of the finite element model consisting of over 70,000 high order tetrahedron elements is shown in Figure 4.B.3-4.

1

ANSYS 5.2
DEC 20 1995
12:03:39
VOLUMES
TYPE NUM

XV =-1
YV =1
ZV =1
DIST=6563
XF =9458
ZF =-5091
VUP =Z
Z-BUFFER

Solid Model Configuration B

Figure 4.B.3 - 1: Alternative B - ANSYS Solid Model

Figure 4.B.3 - 2: Alternative B - View of Solid Model: Inner Shell And Ribs

ANSYS 5.2
DEC 20 1995
13:06:12
VOLUMES
TYPE NUM
XV =1
YV =.5
ZV =-1
DIST=5206
XF =8424
ZF =-3988
VUP =Z
Z-BUFFER

Figure 4.B.3 - 3: Alternative B -View of Solid Model: Coolant Side of Inner Shell

Figure 4.B.3 - 4: Alternative B - Finite Element Model of Solid Elements

The local model includes the weight of the radiation shielding between the vessel shells by increasing the density of the ribs. Unlike the Alternative A approach which used an average density increase to account for the weight of the radiation shield plates, the Alternative B analyses input the individual rib densities as 65% of the adjacent volumes using the data supplied on SK 29-B-2007 and 2008.

4.B.3.3 Loading Conditions and Analysis Results

The loading conditions previously described in Section 4.A.3.3 were applied to the Alternative B stress analysis model with one noted exception. ORNL analysts directly mapped the VDE forces to the finite element model grid locations. This refinement eliminated the need to interpolate applied VDE forces.

The individual load conditions included:

- Vacuum Vessel Gravity
- Blanket Gravity
- Coolant Pressure (2.2-MPa)
- VDE
- Internal Pressure (0.5-MPa)

Combined load cases are as follows:

- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure
- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE
- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE
- Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure (0.5-MPa)

Figures 4.B.3-5 through 4.B.3-10 show the results of the Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure load combination. Figure 4.B.3-5 shows the Tresca stress results on the inside of the inner shell. The stresses are generally low, however, at the interface between the shell and the divertor port, a maximum Tresca stress of 148755. kPa exists. Figure 4.B.3-6 shows the coolant side of the inner face sheet. Maximum Tresca stresses occur at the divertor port interface and at the splice joint where the vessel shell thickness decreases to 40 mm. The coolant side of the outer shell,

Figure 4.B.3-7, shows similar results. Figure 4.B.3-8 shows the resulting stresses for the vessel ribs. The stresses are generally low except for the locations where the ribs are attached to the divertor port shells.

The stresses in the divertor port shells and ribs are shown in Figure 4.B.3-9 and 4.B.3-10. These stresses are generally low except at the locations of the rib attachments. The rib stiffeners that were proposed in this alternative appear undersized and require modification. The divertor port rib connections to the vessel ribs and face sheets show high stresses. These stresses are in general due to the local geometry changes.

Tresca stresses for the combined load case of Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE loads are given in figures 4.B.3-11 and 4.B.3-12. The maximum Tresca stress shown in Figure 4.B.3-11 is in the divertor port region. This stress is primarily due to operational loads. The maximum stress on the vacuum vessel inner shell near a blanket support corner is 97155 kPa. The maximum computed stress between the divertor rails is 64818 kPa. The added modeling detail of the divertor rails combined with the increase in inner shell thickness to 60-mm reduces the large stresses near the inboard divertor rail that were present in the Alternative A results. Figure 4.B.3-12 shows the Tresca stresses near both the inner and outer divertor rails.

The results of the combined load case of Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2*VDE loads are given in Figures 4.B.3-13 and 4.B.3-14. The maximum stress location at the divertor port shell for this load case is the same as the VDE load case. The magnitude of the stress is only slightly higher than the VDE case indicating that these stresses are dominated by the operating loads. The stresses in the vacuum vessel shell are more affected by the 2*VDE loads. This is evidenced by the large increase in the Tresca stress on the inner shell locations [i.e. the inner shell stresses between the divertor rails increase 53% from 64818 kPa (VDE) to 99302 kPa (2*VDE), while the divertor rail stresses increase only 4% from 144200 kPa (VDE) to 149380 kPa (2*VDE)]. The stress near the inner divertor rail is also more affected by the 2*VDE loads as the stress in this region nearly doubles when the VDE loads are increased by a factor of 2, 53504 kPa (VDE) to 103940 kPa (2*VDE).

Figures 4.B.3-15 to 4.B.3-18 show the results for the Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure (0.5 MPa) load combination. The maximum computed Tresca stress shown in Figure 4.B.3-15 on the Coolant side of the inner shell occurs at the splice joint where the shell thickness decreases to 40 mm. The figure also shows some local peaking at the corners of the face and the divertor port. The coolant side of the outer shell (Figure 4.B.3-16) shows similar type results except that the

Figure 4.B.3 - 5: Alternative B -Inner Shell Tresca Stresses (Inside Face) - Operation

Figure 4.B.3 - 6: Alternative B -Inner Shell Tresca Stresses (Coolant Side) - Operation

Figure 4.B.3 - 7: Alternative B -Outer Shell Tresca Stresses (Coolant Side) - Operation

Figure 4.B.3 - 8: Alternative B -Tresca Stresses Vessel Ribs - Operation

Figure 4.B.3 - 9: Alternative B - Divertor Port Shell Tresca Stresses - Operation

Figure 4.B.3 - 10: Alternative B - Divertor Port Rib Tresca Stresses - Operation

Figure 4.B.3 - 11: Alternative B - Tresca Stresses -VDE

Figure 4.B.3 - 12: Alternative B - Tresca Stresses (Plan View) - VDE

Figure 4.B.3 - 13: Alternative B - Inner Shell Tresca Stresses - 2*VDE

Figure 4.B.3 - 14: Alternative B - Tresca Stresses (Plan View) - 2*VDE

Figure 4.B.3 - 15: Alternative B - Inner Shell Tresca Stresses (Coolant Side) - Internal Pressure

Figure 4.B.3 - 15: Alternative B - Inner Shell Tresca Stresses (Coolant Side) - Internal Pressure

Figure 4.B.3 - 16: Alternative B - Outer Shell Tresca Stresses (Coolant Side) - Internal Pressure

Figure 4.B.3 - 17: Alternative B - Tresca Stresses Vessel Rib - Internal Pressure

Figure 4.B.3 - 18: Alternative B - Divertor Port Shell Tresca Stresses - Internal Pressure

stresses are a maximum at the corner attaching the shell to the divertor port. The splice joint also shows higher stress than the adjacent thicker shell. Figure 4.B.3-17 shows the poloidal rib Tresca stresses. The stress levels are generally low except at the connections to the divertor port. The divertor port shell and rib Tresca stresses are shown in figure 4.B.3-18. The highest stress occurs at the corner between the divertor port outer shell and the vacuum vessel outer shell.

Table 4.B-4 compares stress result magnitudes from the Task 29 Alternative B analyses with the previously reported USHT/JCT stress analysis results. A comparison of these results indicates that the port corner stresses have been significantly reduced as a result of the Alternative B reinforcements. Very high localized stresses are identified by the Alternative B results however, at the vessel to port connection where the shells and ribs converge.

4.B.4 Summary and Conclusions

The proposed Alternative B shell and rib reinforcement alternative remains very similar to the current JCT vacuum vessel design. The manufacturing assessment indicates that the replacement of the 40-mm thick plate with 60-mm thick plate in the lower vessel region is considered feasible and within the existing database of manufacturing experience.

The Alternative B stress analysis results show that the combination of model geometry refinements and increased shell thickness' in the lower vessel, reduce the VDE vessel stresses (as compared to the vessel design with 40-mm shell and ribs throughout). The geometry of the divertor port however, creates a discontinuous load path with respect to the port weight which results in high stresses at the connections between the port shell, ribs, and the vacuum vessel inner shell. This indicates that the load is taken out in hoop through the vacuum vessel shell plate. These stresses are of particular concern as they occur at potential weld locations making local geometry transitioning unpredictable and possibly ineffective. Prototyping and testing or redesign of this region may be required.

The stress results also indicate the divertor port ribs and wall thickness' as proposed in the design, are inadequate. Redesign of the duct using increased material thickness' of reduced rib spacing should be considered in future design initiatives.

Table 4.B-4: Vacuum Vessel Stress Comparison

	USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)		Detailed Model with Reinforcement (Task 29 Alternative B)	
Load Case	Mem-brane (MPa)	Mem-brane + Bending (MPa)	Membrane + Bending¹ (at multiple locations) (MPa)	
Normal Operation Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure	118 (port corner)	133 (port corner)	109 (port corner) 280 (port rib)	313 (div port interface) 183 (port shell)
Operation with Disruption Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE	141	181 (div. port interface)	308 (div port interface) 144 (port shell)	70 (port corner)
Operation with Disruption <i>With est. correction*</i> Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE	198 (div port to vessel interface)	267 (div port interface)	307 (div. port interface)	89 (port corner) 149 (port shell)
Off-Normal Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure	204 (port corner)	228 (port corner)	108 (port corner) 159 (port shell)	248 (div port interface) 125 (port centerline)

Notes:

1. The stress values reported in this column are the computed Tresca stresses obtained directly from the finite element analysis results. These stresses represent a total stress intensity (similar to that defined within the ITER Interim Structural Design Criteria - IRB 3224.5.4. These stresses have not been classified in primary, secondary, or peak categories.

ITER TASK 29
VACUUM VESSEL DESIGN (D201)
REINFORCEMENT ALTERNATIVE B
VACUUM VESSEL SKETCH SUMMARY

SK-29-B-2001	18° REINFORCED LOWER VESSEL REGION
SK-29-B-2002	LOWER VESSEL ELEVATION
SK-29-B-2003	ISOMETRIC VIEW RIB ARRANGEMENT
SK-29-B-2004	RIB LAYOUT PLAN VIEW
SK-29-B-2005	VESSEL & DIVERTOR PORT RIB ARRANGEMENT
SK-29-B-2006	18° FACET ARRANGEMENT
SK-29-B-2007	POLOIDAL RIB VOLUMES
SK-29-B-2008	POLOIDAL VOLUMES BETWEEN RIBS.
SK-29-B-2009	18 DEGREE FULL VESSEL SEGMENT

RAYTHEON ENGINEERS AND CONSTRUCTORS	
ITER TASK 29	
VACUUM VESSEL DESIGN (D210)	
REINFORCEMENT ALTERNATIVE B	
SK-29-N-2001	11 28 95

OUTBOARD BLANKET
SUPPORT

INBOARD BLANKET
SUPPORT

DIVERTOR RAILS

VERTICAL VESSEL SUPPORT

LOWER VESSEL ELEVATION

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN ID2011
REINFORCEMENT ALTERNATIVE B
11/20/05

ISOMETRIC VIEW RIB ARRANGEMENT

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN ID2011
REINFORCEMENT ALTERNATIVE B
5K-29-13 2001 11 20 95

RIB LAYOUT PLAN VIEW

VIEW A-A

RAYTHEON ENGINEERS AND CONSTRUCTORS
 ITER TASK 29
 VACUUM VESSEL DESIGN ID2011
 REINFORCEMENT ALTERNATIVE B
 WK 29 11-2004 11-28-95

VESSEL & DIVERTOR PORT RIB ARRANGEMENT

PLAN VIEW
VESSEL CUT AT $Z = 0^\circ$
18° FACET ARRANGEMENT

POLOIDAL RIB VOLUMES

RAYTHEON ENGINEERS AND CONSTRUCTORS	
ITER TASK 29	
VACUUM VESSEL DESIGN (D201)	
REINFORCEMENT ALTERNATIVE B	
5K-29 B 2007	11-28-95

RAYTHEON ENGINEERS AND CONSTRUCTORS
 ITER TASK 29
 VACUUM VESSEL DESIGN ID2011
 REINFORCEMENT ALTERNATIVE B
 SK-29-02008 11 28 95

TOTAL VOLUME OF VOIDS
 = 6.108352E +09 CU/MM

POLOIDAL VOLUMES BETWEEN RIBS

18 DEGREE FULL VESSEL SEGMENT

RAYTHEON ENGINEERS AND CONSTRUCTORS
ITER TASK 29
VACUUM VESSEL DESIGN (D201)
REINFORCEMENT ALTERNATIVE B
REV 01 11/2004 11/20/04

5. Task Summary and Recommendations

Alternatives A and B represent two distinctly different approaches to reduce stresses in the lower vessel region. Alternative A is a notable deviation from the present JCT vessel design as it incorporates a prefabricated subassembly into the vessel segment. Alternative B provides a less radical deviation from the existing JCT by increasing the shell and rib thickness' within the lower vessel region in combination with the addition of local toroidal ribs. Within the scope of a limited manufacturing assessment, both alternatives are considered feasible.

Stress analysis results comparisons indicate that both Alternatives A and B accomplish the objective of reducing stresses within the divertor port region. Differences in modeling techniques (element type, mesh density, etc.) and model details among Alternatives A, B, and the previous USHT analyses however, make it difficult to perform a direct comparison of stress results.

The relative comparison of stress intensity magnitudes provide evidence that the local finite element models developed for the evaluation of the A and B reinforcement alternatives have successfully identified various localized stresses that could not be recovered with the previously used coarser models. These Alternative A and B analyses results therefore provide a basis upon which to begin to evaluate connection reinforcement and detailed design. Refinement of the reinforcement geometry to redistribute highly localized stresses and reanalysis is recommended as the detailed vessel design proceeds.

A comparison of relative costs of the evaluated alternatives was not performed within the Task D201 (29) scope. Likewise, manufacturing schedule considerations and potential impacts were not assessed. Such comparisons considering factors including manufacturing and assembly labor, fixturing, material cost, and material availability are recommended to achieve a more comprehensive understanding of the relative advantages and disadvantages of the Alternative A and B reinforcement approaches.

5.1 *Manufacturing*

Alternative A

The potential vacuum vessel manufacturing benefits of the Alternative A option can be summarized as:

- elimination of complicated welding and associated concerns (distortion, tolerance control, NDE, etc.) in the region where the port ducts, vessel shells, and vessel ribs all converge,
- elimination of the additional supplementary shielding within a region characterized by complex geometry,
- addition of the maximum reinforcement mass (with the forging) at the port to vessel interface,
- flexibility of geometry to eliminate stress risers and redistribute local stresses afforded by the mandatory machining,
- flexibility of geometry to permit smooth transitions from the torus into the port shell along the field assembly joint (if required to simplify the path of the welding and inspection equipment during initial assembly and subsequent remote operations), and
- flexibility in the manufacture of significant portions of the vessel, as the use of a subassembly permits parallel manufacturing by potentially two or more vendors.

The principal manufacturing disadvantages of the Alternative A option (particularly the forging concept) are primarily associated with:

- the subassembly weight and scale, as it places this component near the limit of forging capacity and
- the need for supplementary machining after forging to achieve the required geometry within specified tolerance.

It is recommended that manufacturing process development activities be initiated if the Alternative A subassembly concept is pursued. Process repeatability considering the quantities required for the vessel provides justification for such development activities.

Alternative B

Manufacturing advantages of increasing the shell and rib material thickness' as proposed by the Alternative B option include:

- a design very similar to the current JCT vacuum vessel design which will, in principal, require no specific additional R&D activities, and
- specification of plate thickness' that remain within the existing database of manufacturing experience.

The primary manufacturing disadvantage of Alternative B is that it requires welding within a region of very complex geometry which may be further complicated by the addition of supplemental shield plates.

5.2 Stress Results

Previously noted differences in modeling techniques (element type, mesh density, etc.) and geometry details (particularly at the blanket and divertor attachments to the vessel) make direct stress result comparisons difficult. Subjective comparisons of stress analysis results can be made, however, to assess the relative stress reduction benefits of Alternatives A and B.

Table 5-1 presents JCT specified material allowable stresses and summarizes USHT (Naka Workshop), Alternative A, and Alternative B stress analysis results at various vessel regions. Figure 5-1 schematically depicts the vessel regions identified in the Table.

Vessel to Divertor Port Connection

This connection location is not a concern with the Alternative A monolithic forging. USHT and Alternative B results show that significant stresses are present within this vessel to divertor port connection region.

USHT results show that these stresses approach or slightly exceed the material allowable stresses. The refined Alternative B local model however, shows maximum stresses which greatly exceed the JCT material allowable at

313 MPa. The geometry within this region will require redesign to reduce these maximum stresses and permit an acceptable connection design.

Upper Divertor Port Corner at Vessel

Both Alternatives A and B have successfully reduced the USHT reported stresses at the divertor port corner. This region is no longer a concern with the Alternative A monolithic forging. Alternative B reduces the previously reported USHT stresses by 30-50%.

Inner Shell Between Divertor Rails

In the vicinity of the inner divertor rail, a general review of the stress plots for the four (4) load combinations provides an indication that the maximum computed Tresca stresses for Alternative A are likely to be associated with the absence of model detail in this region.

The stresses within this region are similar in magnitude and within the material allowables. The level of divertor and blanket attachment support detail modeled in Alternative B provides confidence in these reported results.

Inner Shell at Blanket Supports

The USHT reported stresses for the "Normal Operation" and "Operation with Disruption" cases are similar in magnitude to those obtained with the refined Alternative B model. The "2*VDE" case and "Off-Normal" cases show that the Alternative B stress results are at least 30% lower than the USHT results. All stresses remain well within the material allowables.

Vessel Segment Splice Joint Regions

Alternative A and B local model stress results in the regions of the vessel splice joints show increased intensities over those reported in USHT results. The tabulated results for the Operation with Disruption Case (VDE), both with and without the "correction factor", show stress reductions for both Alternatives A and B. Alternative A maximum stresses due to the VDE loads can be decreased by increasing the local face sheet thickness between the divertor rails. As previously noted the maximum computed stresses under the divertor rails are due to lack of detail in the model and are expected to be significantly reduced with alternate modeling. Alternative B stress results demonstrate that inclusion of the divertor rail details and thickening of the vacuum vessel face sheet between the rails reduce the VDE stresses considerably.

5.3 *Conclusion and Recommendations*

In summary, two stress reduction alternatives are proposed for the divertor port region. The results of stress analyses performed to evaluate these alternatives substantiate the stress reduction benefits.

The Alternative A forging permits the load to transfer from the divertor port around to the vessel shell. The presently analyzed geometry however, creates discontinuous load paths between the divertor port and the vessel shells. This can be eliminated with some local modification including the addition of a radius around the forging which tapers down to the thickness of the shell. Re-analysis with geometry refinements at the divertor and blanket supports and local increases in material thickness the adjacent shell are also recommended to reduce the localized VDE stresses.

Alternative B likewise provides significant stress reduction benefits. The increased level of detail included within the finite element model provides confidence in the accuracy of the analysis results. The primary region of concern in Alternative B is the port to vessel shell connection where very high local stresses have been identified. The benefits of local modifications such as curving the sides of the divertor port outward at the port to vessel connection and/or adjusting the rib geometry at this location should be investigated.

Table 5-1: Stress Comparison Summary (Sht. 1 of 5)

Load Case & Stress Location	USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)		Detailed Model with Forged Subassembly (Task 29 Alternative A)	Detailed Model with Increased Shell and Rib Thickness' (Task 29 Alternative B)	Allowable Stresses @ 200 °C JCT Design Description Document	
	Mem-brane ⁴ (MPa)	Mem-brane Bending ⁴ + (MPa)	Membrane + Bending ⁴ (MPa)	Membrane + Bending ⁴ (MPa)	Primary Mem-brane (MPa)	Primary Mem-brane + Bending (MPa)
Normal Operation Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure	(LC11.59) Integration	Point Stresses				
Vessel to Divertor Port Connection (Ribs)	67	95	Not Applicable	313	131	196.5
Upper Divertor Port Corner at Vessel:					↓	↓
Inner Shell	118	133	94 ²	109		
Outer Shell	101	115	82 ²	74		
Inner Shell Between Divertor Rails	45	52	40	34		
Inner Shell at Blanket Supports	74	79	Unavailable Due to Lack of Model Detail	75		
9° Vessel Segment Splice Joint Regions:	IB/OB ¹	IB/OB ¹	IS/OS ³	IS/OS ³		
Port Centerline	45/43	51/51	60/132	65/59		
TF Coil Centerline	45/56	52/67	86/84	80/77		

Table 5-1: Stress Comparison Summary (Sht. 2 of 5)

Load Case & Stress Location		Mem- ⁴ brane ⁴ (MPa)	Mem- ⁴ brane ⁴ + Bending ⁴ (MPa)	Detailed Model with Increased Shell and Rib Thickness ⁷ (Task 29 Alternative B)		Primary Mem- brane (MPa)	Primary Mem- brane + Bending (MPa)
		USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)	Membrane + Bending ⁴ (MPa)			Membrane + Bending ⁴ (MPa)	Primary Mem- brane + Bending (MPa)
9° Vessel Segment Splice Joint Regions: Port Centerline TF Coil Centerline	Upper Divertor Port Corner at Vessel:	141	159	106 ²	70	308	131
	Inner Shell	99	109	108 ²	42		
	Outer Shell	73	76	82 ⁵	65		
	Inner Shell Between Divertor Rails	89	105	Unavailable Due to Lack of Model Detail	97		
	Inner Shell at Blanket Supports	IB/OB ¹	IB/OB ¹	IS/OS ³	IS/OS ³		
	Port Centerline	116/39	133/44	57/112	56/57		
	TF Coil Centerline	73/59	76/69	108/71	69/67		
	Vessel to Divertor Port Connection (Ribs)	132	181	Not Applicable			196.5
	Operation with Disruption Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + VDE	(LC11.54)					

Table 5-1: Stress Comparison Summary (Sht. 3 of 5)

Load Case & Stress Location	USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)		Detailed Model with Forged Subassembly (Task 29 Alternative A)	Detailed Model with Increased Shell and Rib Thickness' (Task 29 Alternative B)	Allowable Stresses @ 200 °C JCT Design Description Document	
	Mem-brane ⁴ (MPa)	Mem-brane + Bending (MPa)	Membrane + Bending ⁴ (MPa)	Membrane + Bending ⁴ (MPa)	Primary Mem-brane (MPa)	Primary Mem-brane + Bending (MPa)
Operation with Disruption (With est. correction)* Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + 2* VDE	(LC12.81)					
Vessel to Divertor Port Connection (Ribs)	198	267	Not Applicable	307	131	196.5
Upper Divertor Port Corner at Vessel:						
Inner Shell	165	244	96 ²	89		
Outer Shell	96	111	91 ²	37		
Inner Shell Between Divertor Rails	106	112	130 ⁵	99		
Inner Shell at Blanket Supports	105	162	Unavailable Due to Lack of Model Detail	122		
9° Vessel Segment Splice Joint Regions:	IB/OB ¹	IB/OB ¹	IS/OS IB-IS ³	IS/OS ³	↓	↓
Port Centerline	187/36	216/48	54/92 107	49/56		
TF Coil Centerline	108/86	110/103	68/58 117	58/69		

Table 5-1: Stress Comparison Summary (Sht. 4 of 5)

Load Case & Stress Location	USHT Results from Static Model without Reinforcement (Oct. '95 Naka Workshop)		Detailed Model with Forged Subassembly (Task 29 Alternative A)	Detailed Model with Increased Shell and Rib Thickness' (Task 29 Alternative B)	Allowable Stresses @ 200 °C JCT Design Description Document	
	Mem-brane ⁴ (MPa)	Mem-brane Bending ⁴ + (MPa)	Membrane + Bending ⁴ (MPa)	Membrane + Bending ⁴ (MPa)	Primary Mem-brane (MPa)	Primary Mem-brane + Bending (MPa)
Off-Normal Vacuum Vessel Gravity + Blanket Gravity + Coolant Pressure + Internal Pressure	(LC11.56)					
Vessel to Divertor Port Connection (Ribs)	62	130	Not Applicable	248	157.2 ↓	235.8 ↓
Upper Divertor Port Corner at Vessel:						
Inner Shell	131	137	176 ²	108		
Outer Shell	204	228	134 ²	101		
Inner Shell Between Divertor Rails	83	97	65 ⁵	67		
Inner Shell at Blanket Supports	104	154	Unavailable Due to Lack of Model Detail	75		
9° Vessel Segment Splice Joints:	IB/OB ¹	IB/OB ¹	IS/OS ³	IS/OS ³	↓	↓
Port Centerline	79/103	88/122	107/178 ⁵	125/112		
TF Coil Centerline	80/101	90/132	139/141	110/109		

Table 5-1: Stress Comparison Summary (Sht. 5 of 5)**Notes:**

¹ IB = Inboard Region; OB = Outboard Region

² The reported divertor port corner stress for Alternative A actually occurs at the forging to shell interface which is some distance from the port corner. As the present model geometry does not include local model transitioning these stresses are directly attributable to local geometry discontinuity and can be expected to decrease with the geometry refinement and re-analysis.

³ IS = Inner Shell; OS = Outershell; IB = Inboard Region: For the case including 2*VDE loads the inboard locations of Alternative A show levels of stress higher than the outboard locations.

⁴ The stress values reported in this column are the computed Tresca stresses obtained directly from the finite element analysis results. These stresses represent a total stress intensity (similar to that defined within the ITER Interim Structural Design Criteria - IRB 3224.5.4. These stresses have not been classified in primary, secondary, or peak categories.

⁵ This reported stress is less than the maximum value indicated in the finite element analysis results, as the maximum stress is considered unrealistic and a direct result of the absence of finite element model detail. A more representative stress has been substituted for the maximum computed stress for comparison with previous analysis results.

**Appendix I: US Home Team Presentation at Vacuum
Vessel Workshop
Naka October 30-November 2, 1995**

*not enough info
to conf - jep*

US Home Team Presentation
Vacuum Vessel loading, stress analysis, and reinforcement concept

B. Riemer, R. Sayer, P. Fogarty, *ORNL*
John Swanson, B. Paul, W. Fragetta, *Raytheon Engineers and Constructors*
T. Schultheiss, *Northrop Grumman Aerospace*
J. Maslakowski, *Rocketdyne Division, Rockwell International*

presented by B. Nelson

Vacuum Vessel Workshop
Naka Joint Work Site
October 30 - November 2, 1995

US Home Team Presentation
Vacuum Vessel loading, stress analysis, and reinforcement concept

B. Riemer, R. Sayer, P. Fogarty, *ORNL*
John Swanson, B. Paul, W. Fragetta, *Raytheon Engineers and Constructors*
T. Schultheiss, *Northrop Grumman Aerospace*
J. Maslakowski, *Rocketdyne Division, Rockwell International*

presented by B. Nelson

Vacuum Vessel Workshop
Naka Joint Work Site
October 30 - November 2, 1995

Presentation Outline

- Introduction and summary
- Loading conditions, including disruption loads
- Dynamic response of structure and comparison with static analysis
- Non-Axisymmetric loading conditions
- Global static analysis of combined loads
- Reinforcement concept and detailed analysis

Introduction and summary

- The vacuum vessel is the focus of this presentation
 - critical component, provides the first safety barrier
 - must withstand normal and fault conditions without losing confinement
- Plasma disruptions cause the most severe loading conditions, especially if non-axisymmetric conditions are considered
- Static and dynamic analyses have been performed assuming linear, elastic material behaviour
- Analysis to date shows the vessel design should be adequate but may require some local reinforcement

Vacuum Vessel Design Loads

- Loads include

- Normal operation:

gravity	
coolant pressure	<i>0 to 2.2 MPa</i>
nuclear heating/temperature gradients	<i>negligible</i>
differential temperature between vessel and blanket/shield	<i>up to 150C</i>

- Plasma disruptions and/or fast discharge of coils:

directly applied electromagnetic loads	<i>~0.3 MPa</i>
reacted loads from in-vessel components	<i>up to 150 MN</i>

- Fault conditions

internal overpressure	<i>0.5 MPa</i>
external overpressure	<i>0.2 MPa</i>
seismic	<i>~0.2 g</i>
TF and PF coil faults	<i>TBD</i>

Plasma Disruption Loads

- Plasma disruption loads are a function of vessel and in-vessel component design, as well as plasma conditions (e.g. halo temperature) and are difficult to quantify
- Experimental results from DIII, JET, C-MOD, JT-60U and other machines indicate high Halo currents with toroidal asymmetry up to a factor of two
- Various ITER simulations have been performed using TSC and MAXFEA that also predict high Halo currents
- All simulations are axisymmetric, and do not address toroidal peaking
- ITER General Design Requirements Document (GDRD) specifies 3 disruption scenario cases: radial, vertical and VDE. Case IV is a "fast" VDE

Plasma Disruptions

- A disruption is a disturbance that results in the uncontrolled decay of plasma current
- Vertical Displacement Episode (VDE), typically the most severe, has several phases:

Current quench

Thermal quench:

Drift phase:

Slow vertical motion of plasma induces toroidal currents opposing motion of plasma, results in net vertical force on the structure

Diamagnetic flux change from beta collapse induces poloidal currents in structure and halo, radial motion of plasma induces toroidal currents to oppose motion

Plasma current decay induces toroidal and poloidal currents in conducting structure and halo, halo currents interact directly with structure.

Plasma Disruption Simulation Models

Simulation	Type	I_i	β_{pol}	Thalo (eV)	Toroidal Resistance (μ -ohms)			L/R toroidal (ms)		
					VV	FW	Backplate	VV	FW	Backplate
TSC (1) 940408H	slow VDE	0.96		20	27	328	33	188	~20	~160
MAXFEA(2) S 82 MD 10 94-05-26 W 1	slow VDE	0.9	0.7	n/a	26	16	16	~195	~425	~325
TSC 950114D	fast rad Case I	1.04	0.56	0	27	10	13.5	190	683	387
TSC 950407E	vertical Case II	1.04	0.56	0	27	10	13.5	190	683	387
TSC 950315H	slow VDE Case III	1.04	0.56	25	27	10	13.5	190	683	387
MAXFEA S 73 MD 12 95-04-06 W1.1	slow VDE	0.9	0.86	n/a	13.2	20	14.2	~390	~340	~370
TSC	fast VDE Case IV	1.04	0.56	0.5	27	10	13.5	190	683	387
Design Nov 94	all			n/a	27	10	13.5	~190	~700	~390
Design Aug 95	all	<1.2		n/a	12.6	6.9 (combined)		~410	~850 (combined)	

1. Tokamak Simulation Code, S. Jardin

2. MAXFEA, P. Barabaschi

File 2

Fig. 2 TSC Filaments for the ITER vessel (VV),

Fig. Overlay of plasma boundaries for "radial" and VDE TSC disruption cases - 950114D and 950315H

"Radial" Disruption Simulation, Case I

0.75 ms

ITER Radial Disruption 950114d
Evolution of Plasma Current, β , and Z_{mag}

POLSEGPC POL Current t = 0.75 ms I_p = 21.13 MA

$t = 0.75 \text{ ms}$ $I_p = 27.73 \text{ MA}$

Forces due to Poloidal Current

"Slow" VDE Disruption , Case III

R. O. Sayer 27 Apr 95

IVVFWSBvst_950315H

Fig. . ITER EDA. Evolution of toroidal current for the first wall, strongback, and vacuum vessel for the TSC VDE Case 950315H ($\langle di/dt \rangle_p = -0.06$ MA/ms).

Fig. 2. Extreme EM Pressures on ITER conducting structures for 4 TSC "Reference" Disruption Simulations for the TAC-6 Geometry.

Plasma Disruption Simulation Summary

Simulation	Type	I_{plasma} (MA)	$\langle dI_p/dt \rangle$ (MA/ms)	ΔZ (drift) (m)	I_{halo}^* (MA)	I_{poloidal} in struct. (MA)	Vertical Force (MN)			
							VV+Div	FW+BP	Max Total	from pol currents only
TSC (1) 940408H	slow VDE	25	~ -0.06	-1.6	6.1	6.7	-122	-133	-188	
MAXFEA(2) S 82 MD 10 94-05-26 W I	slow VDE	24	~ -0.04	-1.6	7.2	-	-29	-116	-157	-128
TSC 950114D	fast rad Case I	24	-2.1	0.		3.3	-	-	-4	0
TSC 950407E	vertical Case II	24	-0.2	-0.2		<3.3	-	-	-11	0
TSC 950315H	slow VDE Case III	24	-0.06	-1.23	4.6	<3.3	-74	-84	-113	-70
MAXFEA S 73 MD 12 95-04-06 W1.1	slow VDE	21	-0.055	-	7.0	-	-51	-99	-147	-96
TSC 950428G	fast VDE Case IV	24	-1.6	-1.23	3.5	-	-17	-67	-60	-42
GDRD	Case II	24	≤ -2.5		4.8					
GDRD	Case III	21	≤ -2.5		8.4					

* I_{halo} = peak toroidal current in plasma halo
 1. Tokamak Simulation Code, S. Jardin
 2. MAXFEA, P. Barabaschi

"Slow" VDE Disruption , Case III

R. O. Sayer 27 Apr 95

FZvst_950315H

Fig. . Evolution of Net Vertical Forces on the ITER first wall, strongback, vacuum vessel, divertor assembly, and all passive structure for TSC VDE Case 950315H.

"Fast" VDE Disruption , Case IV

R. O. Sayer 20 Sep 95

FZvst_950428G

Fig. Evolution of Net Vertical Forces on the ITER first wall, strongback, vacuum vessel, divertor assembly, and all passive structure for TSC VDE Case 950428G.

Dynamic Analysis

- Simplified model includes 18 degree sector of blanket/shield assembly, vacuum vessel, and supports
- Double wall vessel represented by composite shell elements

Composite shell vessel

First wall/shield/
backplate model

Views not to scale

"A" FRAME
SUPPORT ARRAY

MARCH 20, 1988

Vacuum Vessel A-Frame Supports and FEM

Dynamic Analysis - Harmonic Modes

- Mode shapes and frequencies calculated for zeroth harmonic (all sectors that same) using ABAQUS

zeroth harmonic mode	frequency (Hz) (single sector, symmetric)
1	2.0
2	6.8
3	9.3
4	11.3
5	17.7
6	18.3

- Non-symmetric, lateral mode (calculated using cyclic symmetry) is lower, ~1.7 Hz

Dynamic Analysis - EM loads for slow VDE

LOAD

- "slow" VDE, case 949408H scaled from 25 to 21 MA + gravity
- Total vertical load = -156 MN, pressure on vessel walls ~ 0.3 MPa
- VDE loads mapped onto model directly from TSC analysis, includes halo loads

MODEL

- 18 degree sector, composite wall vessel with blanket/shield
- Single A-frame vessel support, flexible plate blanket supports

RESULTS

- Max Tresca stress = 110 MPa @ 1.33 s, top of divertor port opening
= 112 MPa @ 1.64 s, bottom of divertor port opening
- Max deformation = 14 mm @ 1.33 s, 10 mm @ 1.64 s
- Vertical support reaction = 1.1 x applied load

ISSUES

- Asymmetry of halo currents not included
- Vertical shear loads on blanket/shield (due to non-continuous first wall) not included

Job G3A

Tresca Stress (section 9) @ 1.6431 s

112059.	100978.	89897.	78816.	67735.	56654.	45573.	34492.	23411.	12330.	1249.
[kPa]

Job G3A

Tresca Stress (section 1) @ 1.3327 s

110527.
[kPa]

99783.

89038.

78294.

67550.

56805.

46061.

35317.

24572.

13828.

3084.

"Slow" VDE (Case III): Stress and vertical reaction vs time

BWR

4/19/95

Vacuum Vessel Maximum Tresca Stress

Job: G3A
Section 04-01 shape
A frame Lower support w/shear panel
Flexible Plate Blanket Supports

Vertical Applied & Reaction Force

Job: G3A
Section 04-01 shape
A frame Lower support w/shear panel
Flexible Plate Blanket Supports

Slow VDE deflected shape (TSC 940408H)

Section 04-01

Deformed Shape at 1.6431 s (time of max. vessel stress)

Job G3A

Dmag=250

Max. Def. ($t = 1.3256 \text{ s}$)

Dynamic Analysis - EM loads for fast VDE

LOAD

- "fast" VDE, case 950426F scaled from 24 to 21 MA + gravity
- Total vertical load = -52 MN, pressure on vessel walls $\leq 0.2\text{MPa}$
- VDE loads mapped onto model directly from TSC analysis, includes halo loads

MODEL

- 18 degree sector, composite wall vessel with blanket/shield
- Single A-frame vessel support, flexible plate blanket supports

RESULTS

- Max Tresca stress = 100 MPa , 0.15s after plasma current decay
- Max deformation = 12 mm , 0.15s after plasma current decay
- Vertical support reaction = 1.28 x applied load for VV support
= 2.7 x applied load for blanket supports

ISSUES

- Asymmetry of halo currents not included
- Vertical shear loads on blanket/shield (due to non-continuous first wall) not included

Job G4B, Fast VDE + Gravity (TSC 950428G)

Tresca Stress @ section 1

0.1525 s after t_{quench}

"Fast" VDE (case IV): Stress and vertical reaction vs time

BWR

7/14/95

BWR

7/14/95

Fast VDE deflected shape (TSC 950426F)

Deformed shape at $t = 2.5015$ s (beginning of thermal quench)

Job G4A: Fast VDE, TSC case 950426F, Section 04-01

Dmag=250

Deformed shape at $t = 2.5277$ s (time of max. deformation)

Job G4A, Fast VDE, TSC case 950426F, Section 04-01

Dmag=100

Blanket support reactions vs EM load, "fast" VDE

Grav. + Fast TSC Loads, Case 950428G

*Applied force does not include gravity

**Outer Shield Support Force &
Applied Load, Vertical**Job: G4B, Section 04-01 shape
A frame Lower support w/shear panel
Flexible Plate Blanket Supports

Comparison of dynamic and static response

- Simplified model analysis compared for slow and fast VDE, symmetrical loads

Parameter	Unit	"Slow" VDE		"Fast" VDE	
Analysis type		static	dyna- mic	static	dyna- mic
VV deflection	mm	9.8	13.6	9.3	12.1
Blkt/shld assy deflection	mm	7.9	15.1	16.9	26.0
Inboard Blkt/shld assy support vertical reactions (tot for 40)	MN	67.1	77.1	19.1	52.2
Outboard blkt/shld assy. support vertical reactions (tot for 40)	MN	76.2	92.1	48.1	129.3
Vacuum vessel vertical support reactions (totfor 20)	MN	293.1	306.1	184.3	235.1
Max tresca "stress" in VV torus	MPa	100.	112.	72.	105.

Non-axisymmetric Halo currents and loads

- Non-axisymmetric halo currents and resulting loads have been observed in several tokamak experiments.
- Factors of two on peak-to-average loading have been reported
- Simple $1-\cos(\phi)$ distribution of halo current loads was applied to cyclic symmetric model for "slow" VDE load case (net lateral force of ~ 30 MN)
- Reactions at vessel supports compared with symmetric load application

Maximum reaction at single vessel support (1 of 20)	For symmetric distribution of halo loads (MN)	For $\cos(\phi)$ distribution of halo loads (MN)
vertical component	~ 15	~ 21
toroidal component	$\ll 1$	~ 3

- Maximum vertical reaction at vessel support increased by $\sim 40\%$

Forces due to Poloidal Current
Slow VDE, $t = 1.560$ s

221532.

[N]

199379.

177226.

155072.

132919.

110766.

88613.

66460.

44306.

22153.

.01953

vv support halo reactions

ITER vacuum vessel support reactions'								
Static analysis, halo loads with (1-cos) distribution								
Degree P	Radial	Toroidal	Vertical		Node ID	FX	FY	FZ
9	-4.99E+03	-5.79E+08	-5.83E+09		1	9.06E+07	-5.72E+08	-5.83E+09
27	-3.74E+03	-1.41E+09	-4.37E+09		2	6.38E+08	-1.25E+09	-4.37E+09
45	-1.50E+03	-2.09E+09	-1.75E+09		3	1.48E+09	-1.48E+09	-1.75E+09
63	1.52E+03	-2.57E+09	1.78E+09		4	2.29E+09	-1.17E+09	1.78E+09
81	5.02E+03	-2.80E+09	5.87E+09		5	2.76E+09	-4.38E+08	5.87E+09
99	8.67E+03	-2.75E+09	1.01E+10		6	2.71E+09	4.30E+08	1.01E+10
117	1.21E+04	-2.43E+09	1.41E+10		7	2.16E+09	1.10E+09	1.41E+10
135	1.50E+04	-1.86E+09	1.75E+10		8	1.32E+09	1.32E+09	1.75E+10
153	1.70E+04	-1.12E+09	1.99E+10		9	5.08E+08	9.97E+08	1.99E+10
171	1.80E+04	-2.61E+08	2.10E+10		10	4.09E+07	2.58E+08	2.10E+10
189	1.79E+04	6.24E+08	2.09E+10		11	9.76E+07	-6.16E+08	2.09E+10
207	1.66E+04	1.45E+09	1.94E+10		12	6.58E+08	-1.29E+09	1.94E+10
225	1.44E+04	2.14E+09	1.68E+10		13	1.51E+09	-1.51E+09	1.68E+10
243	1.14E+04	2.62E+09	1.33E+10		14	2.33E+09	-1.19E+09	1.33E+10
261	7.88E+03	2.84E+09	9.20E+09		15	2.81E+09	-4.45E+08	9.20E+09
279	4.24E+03	2.79E+09	4.95E+09		16	2.76E+09	4.37E+08	4.95E+09
297	8.09E+02	2.47E+09	9.45E+08		17	2.20E+09	1.12E+09	9.45E+08
315	-2.07E+03	1.91E+09	-2.41E+09		18	1.35E+09	1.35E+09	-2.41E+09
333	-4.11E+03	1.16E+09	-4.80E+09		19	5.28E+08	1.04E+09	-4.80E+09
351	-5.11E+03	3.06E+08	-5.97E+09		20	4.79E+07	3.02E+08	-5.97E+09
Note: All loads in milli-newtons.					Total	2.83E+10	-1.61E+09	1.51E+11

Static Slow VDE Force at t=1560 ms

Reaction forces at supports

Cyclic symmetry model

Halo force asymmetry: $(1-\cos)$

Toroidal component

Rev.3

Vertical component

Halo force asymmetry: (1-cos)

Static Slow VDE Force at t=1560 ms

Reaction forces at supports

Cyclic symmetry model

Blanket Support Toroidal Shear Force

Slow VDE with Toroidal Asymmetric Halo Force ($1-\cos(\Theta)$), Static at $t=1.560$

Blanket Support Radial Shear Force

Slow VDE with Toroidal Asymmetric Halo Force ($1-\cos(\Theta)$), Static at $t=1.560$

Static Slow VDE Force at t=1560 ms

Reaction forces at supports

Cyclic symmetry model

Halo force asymmetry: (1-cos)

Radial component

1.802+01

[N]

1.570+01

1.339+01

1.108+01

8.764+00

6.452+00

4.139+00

1.826+00

-4.870-01

-2.800+00

-5.113+00

Rev.3

Blanket Support Axial Force

Slow VDE with Toroidal Asymmetric Halo Force (1-cos(Θ)), Static at t=1.560

Blanket Support Bending Moment TT

Slow VDE with Toroidal Asymmetric Halo Force (1-cos(Θ)), Static at t=1.560

Blanket Support Bending Moment RR

Slow VDE with Toroidal Asymmetric Halo Force (1-cos(Θ)), Static at t=1.560

Blanket Support Torque

Slow VDE with Toroidal Asymmetric Halo Force ($1-\cos(\Theta)$), Static at $t=1.560$

VDE Load Correction factors

- **Halo Current Magnitude**

- Specified as 40% of I_p , simulations predict 24% to 33% of I_p
- Assume correction factor of: 1.2

- **Halo Current Toroidal Peaking**

- Specified as factor of 2
- Assume correction factor of 1.5

- **Dynamic effects**

- For slow VDE, stress amplification is about 15%
- Must be applied to static analysis

Recommendation: correction factor of 2 on stresses due to VDE loads

Combined loads - normal operation

LOAD

- coolant pressure 2.2 MPa
- gravity loads

MODEL

- 18° sector, double wall vessel

RESULTS

- Max membrane stress = ~120 MPa at top corner of divertor port,
- Max membrane + bending stress = ~130 MPa same place
- Max deformation = ~7 mm at end of divertor port

ISSUES

- none

Combined loads - "slow" VDE (+ normal oper. loads)

LOAD

- "slow" VDE , total vertical load = -150 MN + coolant pressure and gravity loads
consistent with TSC case 940408H scaled 21 MA or MAXFEA 94-04-06
- loads distributed over vessel facesheets, divertor rails, and blanket supports

MODEL

- 18° sector, double wall vessel

RESULTS

- Max membrane stress intensity = ~120 MPa
- Max membrane + bending stress = ~140 MPa at corner of divertor port
- Max membrane + bending stress = ~150 MPa near vert support and div port
- Max deformation = ~9 mm at end of divertor port
~12 mm (from dynamic analysis)

ISSUES

- Correction factor not included for specified halo current peaking, magnitude

Fringe: LC=11.54-RES=1.1-P3/PATRAN R.1-(Tresca)-ABAQUS-21-Jun-95 08:44:21

Shimizu model R4

VDE loads @ 1.642 s, Bkt forces @ 1.6431 s, + Gravity + Coolant Pres.

Face sheets + top port

Tresca stress at section 2

[kPa]

105159.

93488.

81817.

70146.

58476.

46805.

35134.

23463.

11793.

121.9

Combined loads - fault internal pressure (+norm oper.)

LOAD

- internal fault pressure of 0.5 MPa
- coolant pressure and gravity loads

MODEL

- 18° sector, double wall vessel

RESULTS

- Max membrane stress intensity = ~140 MPa
- Max membrane + bending stress = ~180 MPa at edge of divertor port
- Max deformation = ~10 mm in port duct, ~7 mm in torus above div. port

ISSUES

- Dynamic effects of very fast pressure rise may need to be investigated, e.g. explosion

Fringe: LC=11.56-RES=1.1-P3/PATRAN R.1-(Tresca)-ABAQUS-21-Jun-95 08:52:03

Shimizu model R4

Gravity + Coolant press. + internal press.

Face sheets + top port

Tresca stress at section 2

142403.

[kPa]

128248.

114094.

99940.

85785.

71631.

57476.

43322.

29167.

15013.

858.7

Vacuum vessel static stress vs allowables

Load Case	Maximum Calculated Vessel Stress (MPa)		Allowable Stress (MPa) @ 200C (safety factor)	
	Mem-brane	Mem-brane + Bending	Primary Mem-brane	Primary Mem-brane + Bending
Normal Operation (Gravity + coolant pressure)	118	133	131 (1.1)	197 (1.5)
Operation with Disruption (Gravity + coolant press. + slow VDE, symmetrically applied)	117	152	131 (1.1)	197 (1.3)-
Operation with Disruption <i>With est. correction*</i> (Gravity + coolant press.+ 2* slow VDE)	187	193	131 (0.7)	197 (1.0)
Off-Normal (Gravity + coolant press.+ internal pressure of 0.5 MPa)	142	183	157 (1.1)	236 (1.3)

Segmentation of Sector

Field weld joints are shown in orange, they will join the individual, 18 degree sectors together. The shop weld joint, shown in green, joins the two halves of a single, 18 degree sector.

Task D201: Vacuum Vessel Design

**Evaluation of Stress Reduction Alternative in the Region of the
Vessel Support and the Divertor Port**

Interim Status

presented by B. Nelson

US Industrial Consortium Participants:

Raytheon Engineers & Constructors - J. Swanson, B. Paul, W. Fragetta

Northrop Grumman Aerospace - T. Schultheiss

Rocketdyne Division - Rockwell International - J. Maslakowski

ITER Vacuum Vessel (D201)

- Stress reduction alternatives presently being evaluated include:
 - addition of local reinforcement in the region of the vessel support and at port to vessel interfaces
 - minor relocation of the vessel support toward divertor port (7.7m to 8.2m)
 - rounding port corners

- **Local Reinforcement**
 - Initially maximize reinforcement at the port-to-vessel connection by adding large material mass
 - Perform stress analysis & evaluate results
 - Modify reinforcement to optimize design

ITER Vacuum Vessel (D201)

- **Proposed reinforcement is prefabricated component that:**
 - accommodates vessel shell faceting (shell facets located at port center & edges of reinforcement)
 - remains compatible with the port-center segmentation concept
 - provides passages for cooling medium
 - includes the required supplementary neutron shielding
 - utilizes transitioned sections to permit welding to shells
 - can be made compatible with either the DDD fabrication approach or the poloidal segmentation concept proposed for vessel sector prototype

Component Wgt. ~40 mTons

	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95	96	97	98	99	100
0	0	1	4	9	16	25	36	49	64	81	100	121	144	169	196	225	256	289	324	361	400	441	484	529	576	625	676	729	784	841	900	961	1024	1089	1156	1225	1296	1369	1444	1521	1600	1681	1764	1849	1936	2025	2116	2209	2304	2401	2500	2601	2704	2809	2916	3025	3136	3249	3364	3481	3600	3721	3844	3969	4096	4225	4356	4489	4624	4761	4900	5041	5184	5329	5476	5625	5776	5929	6084	6241	6400	6561	6724	6889	7056	7225	7396	7569	7744	7921	8100	8281	8464	8649	8836	9025	9216	9409	9604	9801	10000

Vacuum Vessel Arrangement with Reinforced Divertor Port

Plan View Below Midplane Port

Parallel Activities Initiated to Evaluate Local Reinforcement Design & Fabrication Options

Fabricability Assessment

- Investigate alternate processes to prefabricate large reinforced region
 - Forging (single or multiple segments)
 - Weldment
 - Densified Casting
- Evaluate compatibility with vessel segment fabrication
 - facet arrangement
 - rib layout
 - port center segmentation and field joint welding

Finite Element Analysis

- Develop double wall fine mesh local model of lower vessel region (includes vessel support and divertor port)
 - Utilize global model geometry originally prepared by Dr. Shimizu (JCT-Garching) & modified slightly by USHT at ORNL
 - Include design features associated with port reinforcement
 - Define model boundaries to represent global vessel response
- Analyze load cases and load combinations presented in DDD

Preliminary Fabricability Assessment

- Forging or weldment preferred to densified casting
 - wrought product form
 - properties similar to vessel construction material
 - mechanical
 - physical
 - fabrication (machining, welding, inspection)
 - fewer restrictions on single segment component size (HIP chamber size limitations)
- 8 US Vendors contacted
 - capacity exists to forge as single unit at Beth Forge - Bethlehem, PA. (confirmation of ingot availability required)
 - specialty machining vendors presently being contacted to assess machining capacity of forged or welded component

Preliminary Fabricability Assessment (cont'd)

- Welding of prefabricated reinforced port region (forging or weldment) to vessel segment shells and ribs generally considered feasible
 - *Advantages*
 - prefabricated components can be post-machined to maintain tolerances and control fit-up with mating port segment
 - components can be fabricated in parallel with vessel segment assembly
 - vessel segment welding reduced during fabrication
 - *Difficulties*
 - large (~40 mton) component handling and alignment may be difficult

Finite Element Model Developed for Lower Vessel Region

CAD Model

Detail Model Of Lower Divertor Port Region

Finite Element Model

Detail Model Of Lower Diverter Port Region

ANSYS 5.2
 OCT 26 1995
 07:38:06
 PLOT NO. 1
 ELEMENTS
 DSYS=1
 TYPE NUM

XV =-1
 YV =1
 ZV =1
 DIST=10239
 XF =9610
 ZF =-7665
 VUP =Z
 PRECISE HIDDEN

ANSYS 5.2
 OCT 26 1995
 07:25:49
 PLOT NO. 1
 ELEMENTS
 DSYS=1
 TYPE NUM

 XV =-1
 YV =1
 ZV =1
 DIST=7276
 XF =9610
 ZF =-4366
 VUP =Z
 PRECISE HIDDEN

Detail Model of Lower Inverter Port Region

06:22:38
 PLOT NO. 1

54.341
14000
28000
42000
56000
70000
84000
98000
112000
126000

Load Case
 Operation
 Gravity, Coolant Pressure

Lower Divertor Port Region Fresco Stress MPa

Lower Divertor Port Region Fresco Stress MPa

ANSYS 5.2
OCT 26 1995
08:44:06
PLOT NO. 1

15.209
14000
28000
42000
56000
70000
84000
98000
112000
126000

Lower Inverter Port Fastener Area Stress: MPa

ANSYS 5.2
 OCT 26 1995
 09:18:36
 PLOT NO. 1

39.147
 14000
 28000
 42000
 56000
 70000
 84000
 98000
 112000
 126000

Load Case
 Operational/Support
 Weight, Correction
 (Gravity, Constraint)

Power Hydrogen Fuel Element Stress, Lb/in

126000

179910. 11.9a

(1) The following international provisions)

Page: 14 of 14 Print Post Comment Close

Additional Task Activities

- Complete fabricability assessment
- Review stress analysis results
- Modify design to:
 - reduce localized peak stresses
 - reduce reinforcement mass
 - eliminate sharp discontinuities
- Reanalyze modified design
- Assess modified design compatibility with vessel sector fabrication

Vacuum vessel stress comparison.

Load Case	Static model without reinforcement		Detailed model with reinforcement	
	Mem- brane	Mem- brane + Bending	Mem- brane + Bending absolute peak	Mem- brane + Bending compara ble peak
Normal Operation (Gravity + coolant pressure)	118	133	94	82
Operation with Disruption (Gravity + coolant press. + slow VDE, symmetrically applied)	117	152	120	82
Operation with Disruption <i>With est. correction*</i> (Gravity + coolant press.+ 2* slow VDE)	187	193	220	130
Off-Normal (Gravity + coolant press.+ internal pressure of 0.5 MPa)	142	183	176	176

* correction factor of two on VDE loads is assumed

Summary

- Plasma disruptions and the resulting dynamic loads are critical to the design of the vacuum vessel and require careful analysis.
- Analysis to date includes:
 - Plasma disruption simulations to quantify range of em loads
 - Dynamic analysis of simple model for two VDE load cases
 - Basic static analysis of single and double wall models
- For symmetrical loading, no peaking factors and 6 to 7 MA halo current, the vacuum vessel stress approaches allowable stress limit during VDE and overpressure events
- Including a correction factor of 2 on em loads to account for peaking effects, the stresses in the vessel exceed the allowable limits
- Reinforcement concept based on “collar” made by forging or weldment reduces stresses, but additional analysis is needed around divertor rails
- Non-axisymmetric loading conditions need to be further quantified and evaluated

Five Vessel Supports Compared

(4 single support, 1 double support configuration)

- Modes
- Response under dynamic VDE
 - Maximum vessel stress
 - Selected nodal displacements

Effect of Vessel Support Position on Nodal Displacements

Gravity + Slow VDE (TSC 940408H)

Effect Of Vessel Support Radial Position on Maximum Stress

Gravity + Slow VDE (TSC 940408H)

Appendix II: Task 29 Vacuum Vessel Baseline Design Description Document

Task 29: Vacuum Vessel Design (D201)

Baseline Design Description

for the

Design & Analysis

of the

ITER Vacuum Vessel Ports

Revision	Description	Prepared By:	Issued By:	Date
0	Initial Issue	J. Swanson		8/31/95

Contents

1. PURPOSE	3
2. SCOPE	3
3. INTRODUCTION.....	3
3.1 Vacuum Vessel Description.....	3
3.2 Vacuum Vessel Heating and Cooling.....	5
4. FUNCTIONAL DESCRIPTION	6
5. PORT DESIGN AND ANALYSIS REQUIREMENTS.....	9
5.1 Shop Fabrication and Welding	9
5.2 Field Assembly and Welding	10
5.3 Remote Handling	11
5.4 Vessel Loads and Loading Conditions.....	11
5.5 Design Codes and Allowable Stresses	13
5.6 Results from Previous Analyses	15
6. REFERENCES	18
6.1 Reference Documents:.....	18

1. Purpose

The purpose of this document is to establish the criteria for design and analysis activities associated with ITER Task 29: Vacuum Vessel Design (D201). This criteria will document the baseline design requirements and establish guidance for the activities performed by the Raytheon, Northrop Grumman, and Rocketdyne Industrial Team.

2. Scope

The scope of ITER Task 29 (D201) is to generate design details for the vacuum vessel port structures and inner port extensions. Structural analyses of the ports considering the applicable vessel loading criteria and allowable stresses will also be performed.

3. Introduction

(Many of the descriptions and figures contained within this document have been extracted directly from the ITER Vacuum Vessel Design Description Document. These descriptions present a general overview of the current ITER machine.)

3.1 Vacuum Vessel Description

The Vacuum Vessel is a part of the ITER Tokamak-Basic Machine and provides the primary high vacuum and tritium boundary for the plasma (see Figures 2.0-1 and 2.6-1). The main vessel is torus shaped and located inside the cryostat. It is supported from its lower curved shell by steel structures that pass through the cryostat to the concrete floor below. Twenty (20) toroidal field (TF) coils fit around the vessel and define the outer machine envelope and the space available for ports, supports, and utility connections.

The Vacuum Vessel system is comprised of various sub-components. Four of the primary sub-components include: the main vessel, the port assemblies, the support structures (both the internal in-vessel components supports, as well as, the external vessel gravity supports), and a pressure suppression system. Descriptions are provided below for each of these named components.

The **main vessel** is a double wall welded structure with inner and outer shells fabricated with 40 mm thick 316LN stainless steel (see Figure 2.0-2). The minimum and maximum radii of the vessel torus are 4.1 m and 13 m respectively. The overall vessel height is 14.5 m (see Figure 2.0-3). The inner

Fig. 2.0-1

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

TOROIDAL CROSS SECTION OF VV ASSEMBLY

SCALE: NTS 1/1000 TOP CROSS SECT OF VV ASSEMBLY

ITER

DATE: 07-JUN-95
D.J. LIGHTOVLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

Fig. 2.6-1

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VACUUM VESSEL INTERFACES

SCALE: NTS | DT / IDR / VACUUM VESSEL INTERFACES / 01

ITER

DATE: 07-JUN-95
D.J. LIGHTOVLERS

WORKING PROPOSAL
LEVEL: SKETCH
GARCHING CO-CENTER

Fig. 2.0-2

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VACUUM VESSEL SECTOR

SCALE: NTS

DL/IDR/VACUUM_VESSEL_SECTOR/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

and outer shells are connected with stiffening ribs which give the vessel its required mechanical strength (see Figure 2.1.1.1-1). The total thickness of the vessel shell cross-section is typically in the range of 0.45-0.83 m.

The vacuum vessel provides radiation shielding using an array of plate inserts (see Figure 2.0-4) installed within the space between the shells. The plates will be attached to the ribs such that the total toroidal electrical resistance of the vacuum vessel is sufficient to limit the induced eddy current flow while allowing magnetic field penetration. Water flowing in the space between the plates is required to remove a nuclear heat deposition of 3 - 5 MW.

Vacuum vessel assembly will feature 20 vessel segment/TF coil sub-assemblies. Vessel segments will be joined by field welding at the central plane of the ports (see Figures 2.0-5 and 2.1.1.1-19). The structure of the sector field joint will allow two sector replacement cycles and accommodate a mismatch between adjacent vessel sectors.

The **port assemblies** include all or some of the following subassemblies: the vessel port structures; the inner port ducts and the port duct cover plates; utility feedthroughs; and associated bellows. The vessel port structures are the structural components of each port that are located between the vessel inner and outer shell. The inner port ducts extend from the vessel port structure to the cryostat port interface location. Port duct cover plates cover the end of the inner port ducts (except at particular open locations such as NBI ports and diagnostic accesses) and are part of the vessel pressure boundary. The upper port structures, have integral port ducts and cover plates. Utility feedthroughs to in-vessel components and small top ports penetrate the port duct cover plates.

The **support structures** are used to mechanically support the vessel and internals and must withstand structural loads due to gravity, earthquakes, and induced electromagnetic loads. Each vessel sector is supported externally by a single support located at the bottom of the sector between the TF coils (Figure 2.1.1.1-9). A-frame structures located below the TF coils are used to transfer the mechanical loads to the cryostat and cryostat supports.

Internal vessel supports include the blanket and divertor supports. The inboard and outboard blanket segments are supported from brackets attached to the inner wall of the vacuum vessel. The support structures must allow for differential thermal expansion yet, be sufficient to withstand a large total vertical force of 150-200 MN due to vertical displacement events (VDE). The divertor cassettes are supported by two rails that extend around the bottom of the vessel. The divertor rails are not continuous between adjacent vessel sectors.

The **pressure suppression system** is a large tank used to limit the maximum vessel pressure that may occur during an off normal event. The connection to the vacuum vessel, is by four 1 m² ducts located on the bottom of four divertor

Fig. 2.0-4

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

SCHEMATIC VIEW OF DOUBLE WALL CONCEPT

SCALE: NTS

DL/IDR/DOUBLE_WALL_CONCEPT/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

FIG. 2.1.1.1 - 1

DIMENSIONS RELATE TO ROOM TEMPERATURE

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VESEL RIB LAYOUT

SCALE: NTS DL/IDR/VESEL RIB LAYOUT_CONCEPT/01

DATE: 07-JUN-95
D.J. LICHTONLERS

WORKING PROPOSAL
LEVEL : SKETCH

GARCHING CO-CENTER

Vacuum Vessel Loading conditions and assumptions DRAFT												
Operating State	Vessel Pressure (MPa)			Vessel Temp. (C)		Blanket Temp. (C)		Gravity loads (MN), all sectors				
	Internal	External	Coolant	Max Temp.	Gradient	Avg. Temp.	Delta to VV	VV	Blanket, IB	Blanket, OB	Div, IB	Div, OB
Fab/Assy												
-assembly loads on temporary rail	0.1	0.1	0.1	20	0	n/a	n/a	-70	0 to 12	0 to 28	0	0
- Pressure Test 1	0.1	0.1	2.75-3.44 note 1	20	0	n/a	n/a	-70	0	0	0	0
- Pressure Test 2	0.55 note 2	0.1	0.1	20	0	n/a	n/a	-70	0	0	0	0
Commissioning	0.1	0.1	0.1	20	0	20	0	-70	-15	-35	-5	-10
Off State	0.1	0.1	0.1	20	0	20	0	-70	-15	-35	-5	-10
Baking State	0	0	2.5	200	0	350	150	-70	-15	-35	-5	-10
Normal Plasma Operating State												
- w/o disruption	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
- with disruption (note 3)	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
Off-normal state												
- Severe VDE (note 4)	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
- VDE with non-axisymmetric halo currents (note 5)	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
sector 1 loads												
sector 6 loads												
sector 11 loads												
sector 16 loads												
- Pressure Pulse in VV	0.5	0	2	200	5	250	50	-70	-15	-35	-5	-10
Pressure pulse in cryostat	0	0.2	2	200	5	250	50	-70	-15	-35	-5	-10
- LOFA	0	0	2	200 to 300	5	250-550	-50 to 350	-70	-15	-35	-5	-10
- LOCA	0.1	0.1	0.1 or 2.0 note 6	200 to 450	5	250-550	-200 to 350	-70	-15	-35	-5	-10
- Design basis earthquake, (level SL-2)	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
sector 1 loads												
sector 6 loads												
sector 11 loads												
sector 16 loads												
- Design basis earthquake, (level SL-1)	0	0	2	200	5	250	50	-70	-15	-35	-5	-10
Maintenance State	0.1	0	2	20	0	20	0	-70	-15	-35	-5	-10
Notes												
1. design pressure is 1.1*normal operating pressure, test pressure is 1.25* design pressure												
2. test pressure is 10% more than design pressure												
3. Disruption types include radial and vertical as specified in GDRD												
4. VDE loads are for slow VDE, as specified in GDRD												
5. Non-axisymmetric loads from halo currents with peaking factor of 2, total lateral load of 30 MN												
6. One vessel cooling system at 0.1 MPa, the other at 2.0 MPa												
7. Pressure loads from plasma disruptions are not uniform and are distributed on both face sheets												
8. Dynamic effects are included in blanket and vessel reaction loads

VACUUM VESSEL SHOP WELD ASSEMBLY

SHOP WELD ASSEMBLY

VACUUM VESSEL FIELD JOINT

Fig. 2.1.1.1-19

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

ASSEMBLY PLAN

SCALE: NTS

DL/IDR/ASSEMBLY_PLAN/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOVLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

FIELD JOINT

PORT
EXTENSION

Fig. 2.0-5

DATE: 07-JUN-95
D-J. LICHTWILLERS

ITER

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VESSEL AND PORT FIELD JOINTS

SCALE: NTS DL/IDR/VESSEL AND PORT FIELD JOINTS/01

SOLID CONCEPT

BOX STRUCTURE CONCEPT

Fig. 2.1.1.1-9

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VACUUM VESSEL SUPPORT CONCEPT

SCALE: NTS

DL/IDR/VAC_VES_SUPPORT_CONCEPT/01

ITER

DATE: 07-JUN-85
D.J. LIGHTOWLERS

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

ports. The ducts, which lead to the suppression tank itself are fitted with rupture discs that isolate the tank from the vessel during normal operation.

3.2 Vacuum Vessel Heating and Cooling

Vacuum vessel temperatures will be maintained using a dedicated fluid system that employs water for both cooling and baking. Coolant inlet and outlet ducts located on the divertor ports will be used to direct coolant flow around the vessel within the interspace located between the inner and outer vessel shells. The typical vessel coolant flow path is shown in Fig. 2.1.1.1-10. An overall cooling schematic is shown in Figure 2.1.1.1 - 11. Two independent cooling loops connect 10 sectors each (every other one) in series within a given loop. A coolant flow schematic near a midplane port is shown in Figure 2.1.1.2-2.

The heating/cooling system design parameters are summarized in Table 3.2-1.

Table 3.2 - 1 - Vacuum Vessel - Operating Conditions

Parameter	Value
Cooling (Normal Operation)	
Total Heat Load	3 MW
Maximum Heat Deposition Rate	~0.04 MW/m ³
Total Flow Rate	~65 kg/s (0.065 m ³ /s)
Inlet/Outlet Temperature	~100 C / 111 C
Coolant Velocity	~0.01 - 0.3 m/s
Inlet/Outlet Pressure	~2.0/1.9 MPa
Maximum Vessel Wall Temperature	~160 C
Cooling (Off-Normal Operation)	
Total Heat Load	5 MW
Maximum Heat Deposition Rate	~0.07 MW/m ³
Total Flow Rate (natural convection - DZ-6 m)	~36 kg/s (0.036 m ³ /s)
Inlet/Outlet Temperature	~150 C / 182 C
Coolant Velocity	~0.005 - 0.15 m/s
Inlet/Outlet Pressure	~2.0/1.9 MPa
Maximum Vessel Wall Temp.	~280 C
Baking (Normal Operation)	
Maximum Heat Up Time	~100 hrs
Maximum Heat Deposition Rate	~0.0 MW/m ³
Maximum Inlet Temperature	~200 C
Inlet/Outlet Pressure	~2.5/2.4 MPa
Maximum Vessel Wall Temp.	~200 C

FIG. 2.1.1.1-10

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VESSEL COOLING CONCEPT

SCALE: NTS

DL/IDR/VESSEL_COOLING_CONCEPT/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

Fig. 2.1.1.1-11

DATE: 07-JUN-85
BB / DJL

WORKING PROPOSAL
LEVEL : SKETCH

GARCHING CO-CENTER

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VACUUM VESSEL COOLING SCHEMATIC - SECTORS

SCALE: NTS DL/IDR/VV_COOLING_SCHEMATIC/01

Fig. 2.1.1.2-2

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

PORT RIB LAYOUT

SCALE: NTS | DL/IDR/PORT_RIB_LAYOUT/01

NER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

Vacuum vessel bake-out will be performed at the maximum temperature of 200 C at a pressure of 2.5 MPa. The baking flow conditions are the same as for cooling. The system design will allow the vessel to be heated to 200 C in less than 100 hrs.

4. Functional Description

The interface of the vessel ports with the vacuum vessel will require the evaluation of constructability concerns (fabrication, assembly, and welding) in combination with vessel stresses and associated local stress concentrations. The port design must also consider the interface of the inner port duct with the port extension bellows.

The ITER vacuum vessel has 20 top, midplane, and divertor ports which are used for equipment installation and maintenance, utility feedthroughs, and vacuum pumping. The cover plates for the upper ports are used as feedthroughs for the blanket cooling pipes, a viewing system, and diagnostics. A bellows between the pipes and the cover plate provide the vacuum/pressure seal and absorb the differential thermal expansion between the blanket and the vacuum vessel (see Figure 2.1.1.1 - 6).

The midplane ports are used for replacement of blanket test modules and blanket modules integrated with the ion cyclotron radiofrequency (ICRF) and/or electron cyclotron heating (ECH) system, diagnostics, the installation of remote handling tools and potentially neutral beam injectors (NBI).

The lower horizontal ports are used for replacement of divertor cassettes and vacuum pumping. Vessel cooling water supply and return flowpaths to the ex-vessel water system components are located within the interspace between the divertor port inner and outer shells. Shield plugs are installed in both the midplane and divertor ports which provide shielding for the coils and form part of the first boundary of the vessel (see Figure 2.1.1.1 - 7). Beyond the shield plugs are circular bellows and the outer port extensions which extend to the cryostat. These components form part of the second pressure boundary for the system.

Port assignments/allocation for the upper and lower ports are specified in Table 4-1. Table 4-2 summarizes similar information for the mid-plane port considering the 3 different layout options presently being considered by the Garching Joint Work Site.

The basic configuration of the vessel ports is shown in Figure 2.1.1.2 - 1 and port sizes are summarized in Table 4-3. Due to space constraints, the upper port is a single wall construction, 80 mm thick. A space of 170 mm is maintained between the outside of the port and the TF coil for the thermal shield and

Fig. 2.1.1.1-6

THIS IS PROVISIONAL WORKING INFORMATION ONLY.	
PORT FIELD JOINTS	
SCALE: NTS	DL/IDR/PORT_FIELD_JOINTS/01

ITER
 WORKING PROPOSAL
 LEVEL : SKETCH
 GARCHING CO-CENTER

DATE: 07-JUN-95
 D.J.LIGHTOWLERS

MIDPLANE AND DIVERTOR PORT

MIDPLANE PORT

DIVERTOR PORT

FIG. 2.1.1.1 - 7

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

MIDPLANE AND DIVERTOR PORTS

SCALE: NTS

DL/IDR/INTEGRAL_DIV-PORTS_CONCEPT/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

KEY TO SECTIONS

Fig. 2.1.1.2-1

DIMENSIONS RELATE TO ROOM TEMPERATURE

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

VACUUM VESSEL PORT SECTIONS AT ROOM TEMPERATURE

SCALE: NTS: DL/IDR/VV_PORTS_SECTIONS_AT_ROOM_TEMP/01

ITER DATE: 23-MAY-85
D.J. LIGHTOWERS
WORKING PROPOSAL
LEVEL: SKETCH
GARCHING CO-CENTER

assembly space. A stainless steel (SS) tube is attached to the outside of the port for cooling and baking. Both the midplane and divertor ports are of double wall construction with stiffening ribs between the walls (see Figure 2.1.1.2 -2 in Section 3.2 of this document). As with the upper port, a 170 mm clearance is maintained between the outside of the port and the TF coil.

Table 4-1: Current Upper and Lower Port Allocations

Port Number	Upper Port	Lower Port
1	Diagnostics and blanket cooling piping	Cryopumps
2	Diagnostics and blanket cooling piping	Cryopumps
3	Diagnostics and blanket cooling piping	Cryopumps
4	Diagnostics and blanket cooling piping	Remote handling / diagnostics
5	Diagnostics and blanket cooling piping	Cryopumps
6	Diagnostics and blanket cooling piping pump	Cryopumps
7	Diagnostics and blanket cooling piping	Cryopumps
8	Diagnostics and blanket cooling piping	Cryopumps
9	Diagnostics and blanket cooling piping	Remote handling / diagnostics
10	Diagnostics and blanket cooling piping	Cryopumps
11	Diagnostics and blanket cooling piping	Cryopumps
12	Diagnostics and blanket cooling piping	Cryopumps
13	Diagnostics and blanket cooling piping	Cryopumps
14	Diagnostics and blanket cooling piping	Remote handling / diagnostics
15	Diagnostics and blanket cooling piping	Cryopumps
16	Diagnostics and blanket cooling piping	Cryopumps
17	Diagnostics and blanket cooling piping	Cryopumps
18	Diagnostics and blanket cooling piping	Cryopumps
19	Diagnostics and blanket cooling piping	Remote handling / diagnostics
20	Diagnostics and blanket cooling piping	Cryopumps

Table 4-2: Current Midplane Port Allocations (Three Layout Options)

Port Number	NBI/ICRF Layout	NBI/ECH Layout	ICRF/ECH Layout
1	ICRF	Diagnostics	Diagnostics
2	Test module	Test module	Test module
3	Test module	Test module	Test module
4	Remote handling	Remote handling	Remote handling
5	Test module	Test module	Test module
6	Pellet injector / Torus rough pump	Pellet injector / Torus rough pump	Pellet injector / Torus rough pump
7	Diagnostics	Diagnostics	Test module
8	Test module	Test module	Diagnostics
9	Remote handling	Remote handling	Remote handling
10	Diagnostics	Unallocated	Diagnostics
11	NBI	NBI	Diagnostics
12	NBI	NBI	ICRF
13	NBI	NBI	ICRF
14	Remote handling	Remote handling	Remote handling
15	ICRF	Diagnostics	ECH start-up
16	ICRF	ECH	ECH
17	ICRF	ECH start-up	ICRF
18	Diagnostics	Diagnostics	ICRF
19	Remote handling	Remote handling	Remote handling
20	Diagnostics	Diagnostics	Diagnostics

Table 4-3: Vessel Port Size Summary

Port Location	Dimension Location	Dimension (mm)
Upper Port	Top Width	340
	Bottom Width	1356
	Length	3500
Middle Port	Top Width	1800
	Bottom Width	1800
	Length	2998
Lower Port	Top Width	1647
	Bottom Width	965
	Length	2545

5. Port Design and Analysis Requirements

The vessel port design and analysis activities shall consider the following:

- shop fabrication and welding
- field assembly and welding
- remote handling
- applicable loads and loading conditions
- design codes and allowable stresses
- previous analysis results

5.1 Shop Fabrication and Welding

The vacuum vessel shall be designed for manufacture by conventional and established methods. To reduce and/or minimize the final fabrication/assembly time on site, a half sector of the vessel (9-degree sector) or a full sector (if possible), should be completed in the factory shop, and transported to the site. **(Note: Transportation regulations governing the maximum component shipping size have not been established pending site selection by the JCT. It is assumed that transport of either a 9-degree or 18-degree sector is considered feasible.)**

Vessel fabrication options have been proposed by both the Garching JWS and the Japanese Home Team (JHT). The Garching JWS preferred concept for vessel fabrication is shown in Figure 2.1.1.1 - 16. This approach features completion of the inner shell first. Rib welding to the inner shell and the welding of the outer shell is then completed in a manner such as to minimize weld distortion and control misalignment. The basic fabrication procedure is specified in Appendix 1.

The Garching JWS has also considered the use of T-beam and I-beam sections in combination with inner and outer shell closure plates as alternatives for the manufacture of the double wall vessel. This option may reduce weld distortion and misalignment concerns, but may increase fabrication cost. An assembly scheme is shown in Figure 2.1.1.1 - 17.

The JHT has proposed a vessel segmentation approach that utilizes a poloidal segmentation concept for the manufacture of its 18⁰ sector prototype. This concept approach features the manufacture of several complete shell subassemblies (i.e. inner and outer shells with ribs) that are welded together to form the vessel segment. A concept sketch is shown in Figure 5.1-1.

Fabrication tolerances considered feasible for the vessel are:

- Sector overall height and width maintained within ± 30 mm of specified value,
- Sector wall thickness (distance from inner to outer wall) maintained within ± 5 mm of specified value,
- Sector interior and exterior surfaces maintained within ± 30 mm of specified value,
- Field joint edges, on each side of the sector, must be flat within ± 2 mm,
- The edge location dimensions of all field weld joints (from all 20 sectors) must be the same ± 5 mm, and
- Shell plate material thickness tolerance is -0.3 mm and $+1.8$ mm.

5.2 Field Assembly and Welding

The vacuum vessel is segmented to permit pre-assembly within the TF coils and final assembly on the machine floor. General assembly requirements include the following:

- the number of field joints shall be minimized,
- field welds shall be reweldable after 1 MW a/m^2 at the first wall,
- remotely welded field joints must be double contained and leak detection systems provided,
- bellows must be double contained,
- field joint edges, on each side of the sector, must be radially and toroidally positioned within ± 5 mm of specified location,
- in-vessel components support structures shall be positioned to the following accuracy:
 - a. blanket mount: ± 10 mm (limiter to vessel shell)
 - b. divertor mount: ± 10 mm (target plate to vessel shell), and
- access to primary vacuum shall be possible from the outside without venting the cryostat vacuum.

All conventional manual, automatic, and most advanced welding processes suitable for use on SS 316 LN vessel are being considered for the vacuum vessel. ITER specifications identify Gas Tungsten Arc Welding (GTAW) as the preferred process to be used for the vessel and port welds however the JHT is proposing electron beam welding for the vessel prototype subassemblies.

A vessel of the size and complexity of the ITER vessel can be expected to utilize a combination of weld processes depending upon the type of weld joint, accessibility, and volumetric examination required. Both automated and manual welding are expected to be used during the actual vessel manufacturing.

Various Garching JWS concepts for the field joints are shown in Figures 2.1.1.1-3, 2.1.1.1-4, 2.1.1.1-7, 2.1.1.1-8, and 2.1.1.1-18

5.3 Remote Handling

The vacuum vessel is a semi-permanent component of the machine however ITER requires an established method of remotely assembling and disassembling a vessel sector together with corresponding TF coil. The vessel has been designated a Remote Handling Class 3 system. Specific remote handling requirements for the vacuum vessel include:

- the ITER machine shall be designed with provisions to replace the TF/vessel sector remotely with minimum cutting of cooling pipes and ports.
- field joints must be designed to be remotely leak tested,
- gripping points capable of supporting their full weight over the full range of motion required at installation or removal must be provided on the components,
- the structural supports, coolant lines joints, instrumentation leads and all other (dis)assembly interfaces must be compatible with the capability of the remotely operated tools, and
- all liquid and gas pressure joints must be capable of being leak detected by remote means.

5.4 Vessel Loads and Loading Conditions

(Electromagnetic and the other loading conditions described below for the vacuum vessel are largely dependent on the mechanical and electrical characteristics of the in-vessel components. As the design of these systems evolves, loading conditions will be re-evaluated in order to ensure that consistent reaction forces exist between components. The present evaluation is based on the information provided by the JCT.)

Vacuum vessel design loads during normal operation will include gravity forces, coolant pressure, nuclear heating, and loading due to temperature differences between the vessel and in-vessel components. Plasma disruption will result in directly applied electromagnetic loads as well as reaction forces from in-vessel components. Off-normal events may also produce internal or external torus pressure, electromagnetic forces, or seismic loading.

The vacuum vessel has seven operational states. Brief descriptions of these conditions are provided below and summarized in Table 5.4-1.

1. Commissioning State - Ambient temperature and pressure. Seventeen sectors welded together and three not in place. Under this condition the

SHIELDING PLATE ASSEMBLY

SHELL ASSEMBLY
USING BEAM SECTION

ASSEMBLY SCHEME 1

ASSEMBLY SCHEME 2

Fig. 2.1.1.1-17

DIMENSIONS RELATE TO ROOM TEMPERATURE

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

BEAM ELEMENTS FOR THE CONSTRUCTION OF THE VESSEL (OPTION 2)

SCALE: NTS

DL/IDR/BB_DETAIL_COMPILATION/01

ITER

DATE: 23-MAY-95
BB / DJL

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

Figure 5.1-1

FIGURE 1 Segmentation Plan of Full Scale Fabrication Model

Fig.2.1.1.1-3

DIMENSIONS RELATE TO ROOM TEMPERATURE

THIS IS PROVISIONAL WORKING INFORMATION ONLY.	
VESSEL FIELD JOINT CONCEPT	
SCALE: NTS	DL/IDR/VESSEL_FIELD_JOINT_CONCEPT/01

ITER
 DATE: 07-JUN-95
 DJL/JA
 WORKING PROPOSAL
 LEVEL : SKETCH
 GARCHING CO-CENTER

DETAIL Z

PLASMA SIDE

FIG. 2.1.1.1 - 4

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

SCHEMATIC VIEW OF DOUBLE WALL CONCEPT

SCALE: NTS

DL/IDR/DOUBLE_WALL_CONCEPT/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL

LEVEL : SKETCH

GARCHING CO-CENTER

Appendix 3 - Design Reports

Reference Number	Report Number	Title
1.		R.O. Sayer, "Design and Calculation for Vacuum Vessel and Port Structure - Part A: Plasma Disruption Simulations," final report for VV task D7, January, 1995.
2.		B. W. Riemer, "ITER Vacuum Vessel Dynamic Structural Analysis from Disruption Loading, Task D11," final report for VV task D11, January, 1995.
3.		G. Saji, "The Need for Development and Testing of Low Decay Heat Inconels (Rev. 1) ", July 7, 1994
4.		F. Elio and G. Malavasi, "The Flexible Support for the Blanket", Rev. 3, 18 May 1995.

MIDPLANE PORT

DIVERTOR PORT

MIDPLANE AND DIVERTOR PORT

FIG. 2.1.1.1 - 7

ITER
DATE: 07-JUN-95
D.J. LICHTENBERG
WORKING PROPOSAL
LEVEL: SKETCH
GARCHING CO-CENTER

THIS IS PROVISIONAL WORKING INFORMATION ONLY.	
MIDPLANE AND DIVERTOR PORTS	
SCALE: NTS	DL/IDR/INTEGRAL_DIV-PORTS_CONCEPT/01

SECTION A-A

FIG. 2.1.1.1-8

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

UPPER PORT FIELD JOINT DETAILS

SCALE: NTS

DI./IDR/UPPER PORT FIELD JOINT DETAILS/01

ITER

DATE: 07-JUN-95
DJL/JA

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

WELD TYPE	A	B	C	D	E	F	G	H
	X			X				X
	X			X				X
					X	X	X	
					X	X	X	
					X	X	X	
			X					
		X						

Fig. 2.1.1.1-18

THIS IS PROVISIONAL WORKING INFORMATION ONLY.

OPTIONS FOR WELDED JOINTS

SCALE: NTS

DL/IDR/WELD_JOINT_OPTIONS/01

ITER

DATE: 07-JUN-95
D.J. LIGHTOWLERS

WORKING PROPOSAL
LEVEL : SKETCH
GARCHING CO-CENTER

- blanket backplates are installed. Gravity loads for the blanket backplates and fixturing are included.
2. Off State - Ambient temperature and pressure with only gravity loads of in-vessel systems
 3. Baking State - Baking temperature, normal vacuum and water pressure conditions, and only gravity loads of in-vessel systems
 4. Normal Plasma Operating States
 - a) W/O Disruption - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, and only gravity loads of in-vessel systems
 - b) With Disruption - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems, and electromagnetic loads.
 5. Off-Normal Operating States
 - a) Severe VDE - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems plus VDE loads.
 - b) Toroidal Field Coil Quench - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems, and electromagnetic loads.
 - c) Pressure Pulse in-vessel - Normal operating temperatures and temperature gradients, normal vacuum in the cryostat, normal water pressure in the cooling channels, 0.5 MPa in the vessel interior, and gravity loads of in-vessel systems.
 - d) Pressure Pulse in-cryostat - Normal operating temperatures and temperature gradients, normal vacuum in the vessel interior, normal water pressure in the cooling channels, 0.2 MPa in the cryostat, and gravity loads of in-vessel systems.
 - e) LOFA - Operating temperatures and temperature gradients resulting from decay heat removal by natural convection cooling only, normal vacuum in the vessel interior, normal water pressure in the cooling channels, normal vacuum in the cryostat, and gravity loads of in-vessel systems.
 - f) LOCA - Operating temperatures and temperature gradients resulting from decay heat removal by conduction and radiation only, through the system to the cryostat, 0.1 MPa pressure in the vessel interior, 0.1 MPa water pressure in the cooling channels of one circuit and 2.0 MPa in the other, 0.1 MPa pressure in the cryostat, and gravity loads of in-vessel systems.

- g) Design Basis Earthquake (Level SL-2) - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems, plus SL-2 level seismic loads
 - h) Operating Basis Earthquake (Level SL-1) - Normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems, plus SL-1 level seismic loads
6. Maintenance State - Ambient temperatures, normal vacuum in the cryostat, normal water pressure in the cooling channels, atmospheric pressure in the vessel interior, gravity loads of in-vessel systems, and RH equipment loads.

5.5 Design Codes and Allowable Stresses

The ITER vessel is a SIC-2 component and shall be designed by using a well established design code such as ASME Sec. III Subsection NC with design by analysis. Other applicable codes and standards include:

- Interim ITER Structural Design Criteria, Draft 1.
- ITER General Design Requirement Document
- ASME Code, Code Case N-47
- RCC-MR Design Code
- ASTM Standards
- ISO Standards

The allowable stress and deformation limits to be applied for the vessel port design activities will utilize the present criteria provided within the ITER Vacuum Vessel Design Description Document. Allowable stresses have been determined by classifying each of the described vessel operating states and defining acceptable damage limits. The ITER Interim Structural Design Criteria has defines operational condition classifications A, C, and D where:

- Level A protects a component against excessive deformation, plastic instability, fracture, fatigue, etc.,
- Level C is similar to A except that small levels of overall deformation could occur, and
- Level D is similar to C but with lower safety margins.

The Garching JWS preliminary classifications of the operational states of the vacuum vessel are summarized in Table 5.5-1. (Note: The structural design criteria defined in this table is based on requirements from the ASME Code Section III, however the vessel must be designed to codes and standards deemed to be representative for the Four Parties. Additionally, the design must consider the possibility that the host country may require conformance to different codes and standards. Vessel port design and analysis activities will

proceed using this criteria unless otherwise directed by the Garching JWS or the JCT.)

Table 5.4-1 - Vacuum Vessel Operating Conditions & Loads

Operating State	Vessel Pressure			Vessel Temperature		Loads			
	Int. VV Pressure (MPa)	Ext. VV Pressure (MPa)	Coolant Pressure (MPa)	Max. Temp. (C)	Max. Temp. Grad. (C)	VV Weight (MN)	Blanket Weight (MN)	Divertor Weight (MN)	Other Loads (MN)
1) Commissioning	0.1	0.1	0.1	20	0	~70	~50	~15	0
2) Off State	0.1	0.1	0.1	20	0	~70	~50	~15	0
3) Baking State	0	0	2.5	200	0	~70	~50	~15	0
4) Normal Plasma Operating States									
4a) W/O Disruption	0	0	2.0	200	5	~70	~50	~15	0
4b) With Disruption	0	0	2.0	200	5	~70	~50	~15	~0.2 MPa (Note 1)
5) Off-Normal Operating States									
5a) Severe VDE	0	0	2.0	200	5	~70	~50	~15	~150 MN (Note 2)
5b) Toroidal Coil Quench	0	0	2.0	200	5	~70	~50	~15	~0.4 MPa (Note 3)
5c) Pressure Pulse in-vessel	0.5	0	2.0	200	5	~70	~50	~15	0
5d) Pressure Pulse in-cryostat	0	0.2	2.0	200	5	~70	~50	~15	0
5e) LOFA	0	0	2.0	~300	5	~70	~50	~15	0
5f) LOCA	0.1	0.1	0.1 or 2.0 (Note 5)	[~450] (6)	5	~70	~50	~15	0
5g) Design Basis Earthquake (Level SL-2)	0	0	2.0	200	5	~70	~50	~15	TBD g
5h) Design Basis Earthquake (Level SL-1)	0	0	2.0	200	5	~70	~50	~15	0.2 g -vert. & 0.3 g horz.
6) Maintenance State	0.1	0	2.0	20	0	~70	~50	~15	TBD (Note 4)

Notes:

1. Load estimate due to a centered disruption.
2. Load estimate corresponding to a halo and eddy current value of 7.4 MA that is toroidally symmetric.
3. Load estimate due to a toroidal field coil quench.
4. Loads due to RH equipment and operations are TBD.
5. One vessel cooling system is at 0.1 MPa and the other is at 2.0 MPa.
6. Rough estimate of the vessel temperature. Value is TBD.

Table 5.5-1 - Vacuum Vessel Allowable Stress Limits

Operating State	Operational Condition Classification	Acceptable Consequences	Structural Design Criteria ^{1,2}			
			General Primary Membrane	Local Primary Membrane	Primary Memb. + Bending	Primary + Secondary
1) Commissioning	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
2) Off State	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
3) Baking State	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
4) Normal Plasma Operating States						
4a) W/O Disruption	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
4b) With Disruption	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
5) Off-Normal Operating States						
5a) Severe VDE	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
5b) Toroidal Coil Quench	C	Local Yielding	Note 3	Note 4	Note 4	
5c) Pressure Pulse in-vessel	C	Local Yielding	Note 3	Note 4	Note 4	
5d) Pressure Pulse in-cryostat	C	Local Yielding	Note 3	Note 4	Note 4	
5e) LOFA	C	Local Yielding	Note 3	Note 4	Note 4	
5f) LOCA	C or D	TBD	Note 3 or 5	Note 4 or 6	Note 4 or 6	
5g) Design Basis Earthquake	D	TBD	Note 5	Note 6	Note 6	
5h) Design Basis Earthquake	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$
6) Maintenance State	A	No Damage	$<S_m$	$<1.5 S_m$	$<1.5 S_m$	$<3 S_m$

Notes:

1. S_m is the lesser of $2/3 S_y$ or $1/3 S_u$.
2. S_m values are for irradiated base or weld materials at the maximum expected temperature. This criteria value assumes that the irradiated material has acceptable ductility.
3. Greater of $1.2 S_m$ and $1.0 S_y$.
4. Greater of $1.8 S_m$ and $1.5 S_y$.
5. Minimum of $2.4 S_m$ and $0.7 S_u$.
6. Minimum of $3.6 S_m$ and $1.05 S_u$.

5.6 Results from Previous Analyses

An evaluation of the structural response of the vessel to individual loadings was performed by the US Home Team in conjunction with the Garching JWS. Three finite element models of the vessel were developed and used to perform electromagnetic, static, and dynamic analysis (Figure 2.1.1.1 - 12). These models and the analysis methods are discussed in Appendix 2.

Gravity loads were evaluated using an 18-deg single-wall composite -shell model with the reference vessel support and flexible-plate type blanket/shield supports.

SINGLE RIB MODEL

SINGLE-WALL COMPOSITE

DOUBLE-WALL MODEL

Fig. 2.1.1.1-12 - Vacuum Vessel Finite Element Models

The applied gravity force of 130-MN causes the vessel to tilt slightly outboard, producing a displacement of 7.7-mm at the end of the divertor duct extension. The maximum stress in the vessel for this condition is 58-MPa (membrane).

Coolant pressure effects were evaluated using the 1.5-deg segment model comprised of the face sheets and one rib. For a coolant pressure of 2.2-MPa, the vessel experiences a global dilation of approximately 2-mm and a maximum stress in the outboard region of 65-MPa (35-MPa membrane, 30-MPa bending). Due to curvature effects, the bending stress in most areas is less than that approximated by a flat-plate.

The effect of a temperature difference between the vacuum vessel and blanket/shield was analyzed using the 18-deg single-wall composite-shell model with the blanket geometry. Given a ΔT of 175 C (an initial estimate for normal operation which now appears to be high) between the blankets and vessel, stresses of 10-120 MPa are produced in the vessel at the blanket support interface. For this case, which assumes a flexible type blanket support, the stress depends largely on the assumed radial stiffness of the support. Temperature gradients and stress due to heating in the vessel itself were found to be small.

GDRD Section 2.2.7.5 defines three reference plasma disruption scenarios for analysis: radial disruption, vertical disruption, and vertical displacement events (VDE). The most severe mechanical loading occurs during a downward VDE, when the plasma can move as much as 1.6-m off-axis before the current begins to decay. This produces large vertical forces on the in-vessel components, followed by a peak electromagnetic pressure of 0.2-MPa in the vessel wall due to induced currents (Figure 2.1.1.1 - 13). VDE loads are specified in Reference 1 (see Appendix 3).

Dynamic analysis was performed for both a slow and fast decay of plasma current following the motion (Figure 2.1.1.1 - 14). In both cases, the maximum displacement at the divertor duct end is 10-15 mm and the stress intensity is 112-MPa in the region between the divertor duct and the support. This stress is ~ 15 % higher than is obtained when the loads are applied statically. Based on these results, the dynamic loading factor for the vessel is assumed to be 1.15. Dynamic analysis results are given in Reference 2 (see Appendix 3).

A final individual loading condition is associated with internal pressure due to accidents within the torus chamber. For an internal pressure of 0.5-MPa, the outboard radial deflection of the torus is 2-mm and maximum stress intensity is 85-MPa (65-MPa due to membrane stress). Stresses were found to be higher (95-MPa) in the vertical port duct walls.

Figure 2.1.1.1-13 - Vacuum Vessel Disruption Forces

Figure 2.1.1.1-14 - Vacuum Vessel Dynamic Response

5.6.1 Combined Loads

Load combinations were evaluated for three possible operating scenarios:

- 1) normal plasma operation w/o disruption (operating state 4a - normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, and only gravity loads of in-vessel systems)
- 2) off-normal operation - severe VDE (operating state 5a - normal operating temperatures and temperature gradients, normal vacuum and water pressure conditions, gravity loads of in-vessel systems plus VDE loads), and
- 3) off-normal operating - pressure pulse in-vessel (operating state 5c - normal operating temperatures and temperature gradients, normal vacuum in the cryostat, normal water pressure in the cooling channels, 0.5 MPa in the vessel interior, and gravity loads of in-vessel systems).

During normal operation, the vessel deformation is approximately 6-mm and stress levels are about 75-MPa (50-MPa membrane) with some stress peaking up to 160-MPa in the region around the vessel support (Fig. 2.1.1.1 - 15).

The forces due to a severe VDE (assuming a halo current of 7 MA), increase the deformation to 10-15 mm and stress to about 175-MPa (100-MPa membrane) in the support region. An additional stress concentration of up to 260-MPa is found around the square corners of the divertor duct. The analysis does not include toroidal peaking of the force due to plasma "halo" currents.

For the pressure pulse in-vessel condition, the maximum displacement is about 4-mm in the torus and stress intensity is 130-MPa (90-MPa membrane) with peaking in the divertor duct corners to 290-MPa. Further mesh refinement is required in this area to improve accuracy of the results.

5.6.2 Evaluation

Table 5.6.2 - 1 provides a stress summary for each of the 3 combined load cases described in Section 5.6.1. Results for additional "corrected" severe VDE operating state are also provided to address halo currents to 8.4 MA, the toroidal asymmetry of plasma forces as specified in GDRD table 2.2-6(b), and dynamic effects.

*[The correction of 2 was computed considering 1.2 for the halo current (8.4 MA / 7 MA=1.2), 1.5 for the toroidal asymmetry (first approximation - the affects of the asymmetry may be larger than this approximation), and 1.15 for dynamic effects. Combining the three corrections results in a total correction factor of ~2 (1.2*1.5*1.15 =~2).]*

Figure 2.1.1.1-15 Vacuum Vessel Stress Due to Load Combinations

Table 5.6.2- 1 Vacuum Vessel Maximum Stress and Allowables

Load Case	Maximum Vessel Stress			Allowable Stress @ 200C / (SF)			Allowable Stress @ 300C / (SF)			Allowable Stress @ 400C / (SF)		
	Primary Membr	Primary Membr + Bnding	Primary + Secndry	Primary Membr	Primary Membr + Bnding	Primary + Secndry	Primary Membr	Primary Membr + Bnding	Primary + Secndry	Primary Membr	Primary Membr + Bnding	Primary + Secndry
Operation (4a) (Gravity + coolant pressure)	50	75	160	131 (2.6)	196.5 (2.6)	393 (2.5)	116 (2.3)	174 (2.3)	348 (2.2)	105 (2.1)	157.5 (2.1)	315 (2.0)
Operation/Disruption (5a) <i>As calculated</i> (Gravity + coolant + VDE)	100	175	260	131 (1.3)	196.5 (1.1)	393 (1.5)	116 (1.2)	174 (1.0)	348 (1.3)	105 (1.1)	157.5 (0.9)	315 (1.2)
Operation/Disruption (5a) <i>With est. correction*</i> (Gravity + coolant + VDE)	150	275	360	131 (0.9)	196.5 (0.7)	393 (1.1)	116 (0.8)	174 (0.6)	348 (1.0)	105 (0.7)	157.5 (0.6)	315 (0.9)
Off-Normal (5c) (Gravity + coolant + 0.5 MPa internal pressure)	90	130	290	157.2 (1.7)	235.8 (1.8)	472 (1.6)	139.2 (1.5)	208.8 (1.6)	418 (0.9)	126 (1.4)	189 (1.5)	378 (1.3)

* correction = 2 x VDE stress

Notes:

1. The design criteria used to determine the allowable stress values is defined in section 2.2.3.
2. The Sm, Sy, and Su values used in this table are from the ASME Code Section III.
3. Material properties are for SS 316 LN.

The tabulated results indicate that vacuum vessel stresses approach allowable limits for the VDE case. The corrected VDE case exceeds allowable limits.

6. References

6.1 Reference Documents:

The following documents contain criteria applicable to the design and analysis of the vacuum vessel ports:

- ITER Task Assignment No. 29
- Vacuum Vessel Design Description Document dated 6/2/95

Appendix 1 - Vacuum Vessel Fabrication Procedure

Listed below is the basic Garching JWS fabrication procedure for the vessel. A main feature of this procedure is that the inner shell is completed first.

0. Prepare assembly fixtures

1. Sub-assembly of inner shell, including port extension (mid-plane / divertor)

- 1.1 Cutting and measurement of procured plates
- 1.2 Bending plates, machining and setting plates on fixture
- 1.3 Welding plates
- 1.4 Inspection
- 1.5 Repairing weld joints and inspection (if required)
- 1.6 Preparation for welding ribs and stiffeners

2. Sub-assembly of poloidal, local stiffeners and port stiffeners

- 2.1 Cutting / measurements / machining plates
 - 2.2 Preparation for welding ribs / stiffeners on the inner shell
 - 2.3 Setting temporal fixtures)
 - 2.4 Welding ribs / stiffeners
 - 2.5 Dimensional checks and adjustment ribs / stiffeners
 - 2.6 Inspection
 - 2.7 Repairing weld joints and inspection (if required)
 - 2.8 Machining ribs / stiffeners for welding the outer shell)
- * Local stiffeners : structures around supports for in-vessel components, and local toroidal ribs provided for bifurcation of flow path

3. Sub-assembly of internal shields

- 3.1 Cutting / bending / machining plates
- 3.2 Surface treatment if necessary)
- 3.3 Measurements
- 3.4 Mechanically joining shield plates to ribs (or stiffeners)

4. Sub-assembly of outer shell, including port extension (mid-plane / divertor)

- 4.1 Cutting and measurement of plates
- 4.2 Bending, machining plates and weld preparation
- 4.3 Welding plates
- 4.4 Inspection
- 4.5 Repairing weld joints and inspection (if required)

Appendix 2 - Analysis Methods and Models

Analysis Method

Disruption forces, being the most difficult of the design loads to quantify, have been evaluated by two methods: 1) mapping of forces from an axisymmetric plasma simulation, and 2) performing three-dimensional electromagnetic analysis.

The first method involves mapping of force results from a plasma simulation code directly onto the dynamics analysis model. By this method, plasma "halo" current forces are included as a distributed load that is consistent with other plasma forces.

Disruption forces have also been estimated using a three-dimensional finite element electromagnetics code. This program calculates the element current and pressure distribution on a shell-element representation of the three-dimensional structure. A multifilament model of plasma motion and current decay is used to generate transient forces for structural analysis. Plasma field input is taken from data generated by the axisymmetric plasma simulation codes. By this method, three-dimensional eddy current effects are considered.

Following the development of disruption loads, linear static analysis is performed to determine the structural response of the vessel. This method is appropriate for all individual loads except disruption transients, which are evaluated by dynamic analysis. An equivalent static load is used, however, to evaluate the load combinations.

Analysis Models

Three finite element models of the vessel have been developed for electromagnetic, static, and dynamic analysis. The first is a 1.5-deg toroidal rib-and-face-sheet model with a fine mesh for evaluating local bending effects due to coolant pressure and thermal gradients. The second is an 18-deg single-wall composite-shell model that includes port extensions, vessel supports, and in-vessel component geometry's. This model has been used to determine the transient response of the vessel and internals to disruption forces. The third model is an 18-deg double-walled sector with an appropriate mesh for combined loads static analysis. This model has also been used as the input geometry for three-dimensional electromagnetic analysis.