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**Design and thermo-structural analyses of
the DTT vacuum vessel, ports and
cryostat base systems.**

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Abstract

Nuclear fusion is regarded as one of the most promising energy sources for the years to come. However, achieving this goal is considered to be one of the most difficult tasks, due to the extreme physical and technological complexity involved. The European Roadmap toward commercial nuclear fusion energy is based on two main projects: ITER and DEMO. In addition to these primary projects, there are several smaller ancillary projects, each with a specific scientific objective. Among these projects, the work presented in this thesis focuses on Italy's next tokamak, the Divertor Tokamak Test (DTT), which is currently under construction at ENEA in Frascati. The scientific objective of the DTT is to study different divertor solutions for heat extraction that will be relevant to DEMO power plant. This thesis work is devoted to the thermo-mechanical analysis and structural integrity verifications of some of the main structural components of the DTT machine. The design of these components, carried out according to a well established Design By Analysis approach, is closely linked to compliance with the limits imposed by dedicated standards, the study and comparison of which was an integral part of this work. The study began with an analysis of the dynamic behavior of the machine toroidal complex. Then, the response of the system under the action of seismic actions was examined, with particular attention given to the displacements and stresses at the base of the cryostat. The investigation also involved the potential benefits of stress reduction through the incorporation of seismic isolation systems. Therefore, an analysis of the vacuum vessel was conducted in consideration of its various operational states. The design loads for this component were determined to be attributed to electromagnetic actions. Specifically, the case study on Vertical Displacement Events revealed potential critical concerns from the perspective of structural integrity. Utilizing the submodeling technique, a series of progressive strengthening interventions were assessed, culminating in compliance with the reference standards. Finally, the bolted flanged joint between the vacuum vessel and the gravity supports was examined. Given that the position and dimensions of the joint are constrained by spatial limitations, the objective was to identify an optimized configuration that was compatible with these boundary conditions.

Chapter 1

Introduction

1.1 Global energy outlook

The global energy crisis, particularly in Europe, has reached unprecedented levels following the advent of the pandemic and then the Russian invasion of Ukraine. The supply of natural gas to Europe from Russia has decreased by 80% in 2022 compared to previous years, resulting in a series of consequences for consumers associated with rising prices and emissions, due to a return to using coal and oil in place of natural gas [1]. It is therefore evident that a fundamental transformation of the energy system is required, one that encompasses not only the diversification of energy sources but also the guarantee of secure and economically sustainable access to energy services. The report [1] presents three distinct scenarios, each differing based on the political response to the crisis in question. The first scenario, designated the Stated Policies Scenario (STEPS), depicts the prospective trajectory of the energy system in the event that the prevailing policies remain in place. In the Announced Pledges Scenario (APS), it is assumed that all governments will fully implement all relevant energy policies meeting expected deadlines. The final scenario, Net Zero Emissions by 2050 (NZE), sets out specific targets, most notably the limitation of global warming to 1.5°C, as well as the strategies to achieve this, particularly the elimination of CO_2 emissions from energy and industrial processes by 2050. As illustrated in Figure 1.1, in 2021 approximately 65% of the coal and 40% of the natural gas were utilized in power generation, with this trend showing an upward trajectory. All the scenarios described predict a reduction in these percentages by 2030, albeit to a different extent.

Moreover, in light of the projected 30% increase in global energy consumption by 2050 and the anticipated doubling of electricity production [2], it is imperative to pursue the development of alternative, greenhouse gas-free technologies.

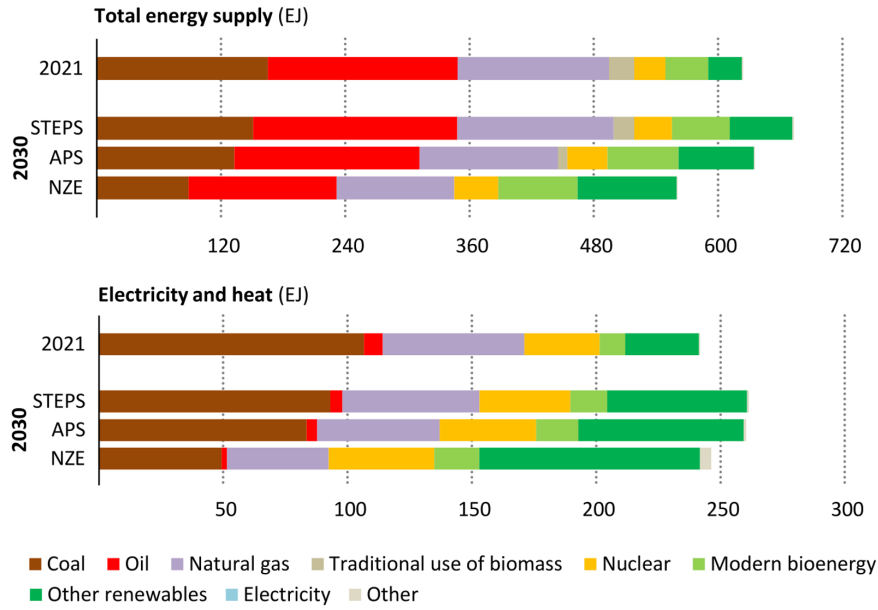


Figure 1.1: Global energy supply and demand by sector, scenario and fuel [1].

1.2 Nuclear fusion

In view of the above, nuclear fusion has the potential to emerge as an important alternative source of energy in the future, particularly given its ability to produce energy without emitting carbon dioxide. The nuclear fusion reaction has the potential to release up to four million times more energy than a chemical reaction, such as the combustion of coal, oil or gas. Furthermore, it has the capacity to release up to four times more energy than that obtained from nuclear fission processes [3]. In order to achieve a nuclear fusion process on Earth, the fuel must be heated to temperatures of approximately 150 million degrees Celsius, which is 10 times the temperature of the Sun. This is required for the fuel to become plasma [4]. To this end, the most promising fuel mixture for use in fusion reactions is a mixture of deuterium and tritium, two isotopes of hydrogen. The reaction between deuterium and tritium results in the formation of a helium-5 nucleus, which subsequently undergoes a decay process, leading to the creation of a helium-4 nucleus, one free neutron, and the release of 17.6 MeV of energy [5]. The procurement of the fuel necessary for the reaction represents a significant challenge for those engaged in scientific research into fusion. Deuterium is a plentiful element in nature, whereas tritium can be produced through the interaction between neutrons released by a fusion reaction and lithium, which is also available in nature. Nevertheless, the production and extraction of tritium in sufficient quantities to sustain the fusion reaction are yet to be fully resolved, representing a significant challenge to the commercial viability of fusion as an energy alternative. In any case, pursuing fusion continues to be worthwhile, since, among other positive aspects, the process of fusion

does not produce high-level and long-lasting nuclear waste in the same way that nuclear fission processes do. Furthermore, in contrast to fission, fusion is inherently safe, as it is a process that can only be achieved under highly controlled conditions. If these conditions are not met, the plasma will rapidly cool, leading to the sudden halt of the reaction.

1.3 Plasma confinement

The exceptionally high temperatures reached by the plasma necessitate the utilisation of sophisticated confinement techniques to prevent it from colliding with the surrounding metallic structures within the vacuum chamber. To date, two principal technologies have been employed for the purpose of plasma confinement. The first of these is based on inertial confinement [6], which exploits the inertia of the fuel mass, while the second relies on magnetic confinement, which is achieved through a rotational transformation of the toroidal magnetic field. The methods by which this transformation is pursued differentiate the two principal magnetic confinement devices, namely the tokamak and the stellarator. In the case of the tokamak, whose external magnets are axisymmetric, the twisting of the toroidal magnetic field is produced by the toroidal current within the plasma. In contrast, in the stellarator, non-axisymmetric external magnets are used [7]. The configuration of the magnets results in a greater plasma volume within the tokamak than within the stellarator. Conversely, as the current in the tokamak is induced by a transformer, these devices are susceptible to instabilities associated with the current itself, rendering them unable to operate in a steady state, in comparison to the stellarator, which is capable of doing so.

1.4 Tokamak

Tokamak is a Russian acronym for *Toroidalnaya kamera and manitnaya katushka* that means toroidal chamber with magnetic coils. This device employs three distinct sets of magnets: Toroidal Field Coils (TFC), Poloidal Field Coils (PFC), and a Central Solenoid (CS). The operational principle is analogous to that of a transformer, wherein the primary circuit is constituted by the central solenoid, while the secondary circuit, in which the current is induced, consists of the plasma. The interaction between the plasma current and the toroidal magnetic field generated by the TFC, within which the current flows in a poloidal direction, enables the twisting of the field that is necessary to prevent the plasma from touching the walls of the vacuum chamber. The poloidal magnets, which permit the current to flow toroidally, are responsible for the functions of vertical plasma confinement, plasma shaping, heating and the driving of the current in the plasma. A schematic representation

of the acting magnetic fields, which result in an helicoidal field on the plasma, is presented in Figure 1.2 [8].

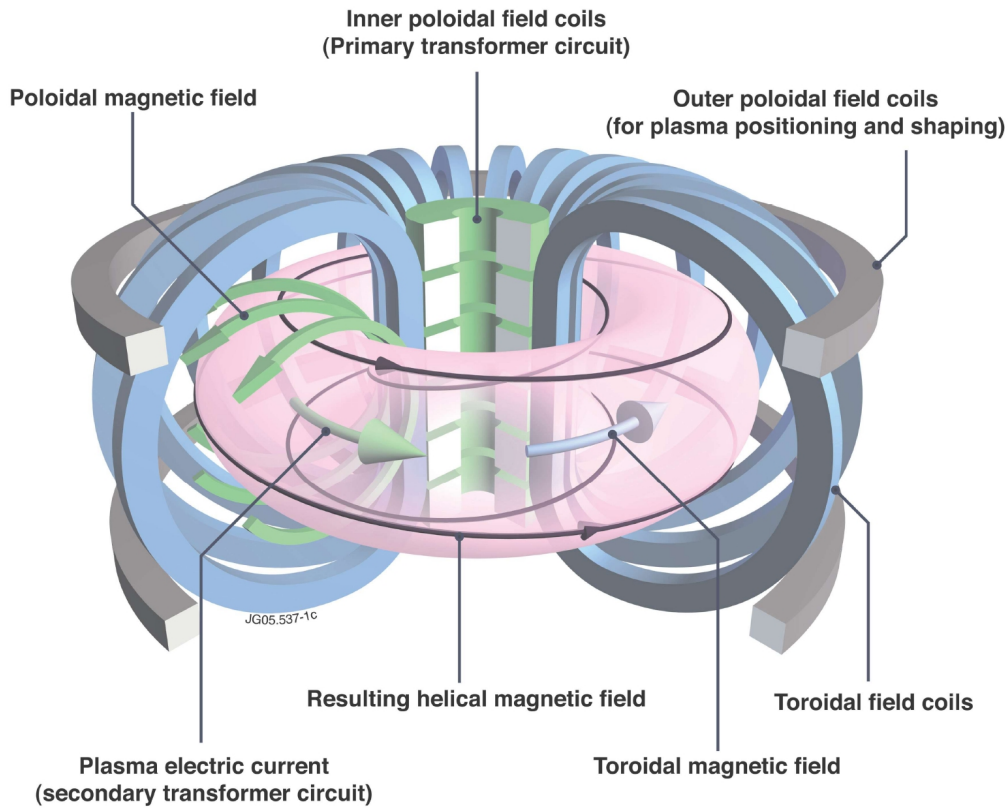


Figure 1.2: Schematic representation of a tokamak magnetic system [8].

1.5 EUROfusion roadmap

The target date for the integration of electricity generated by fusion into the European grid is 2050. This target was established in the document issued in 2012 by the European Fusion Development Agreement (EFDA), the predecessor of EUROfusion, entitled “European roadmap towards fusion electricity”. The contents of the roadmap are schematised in Figure 1.3 .

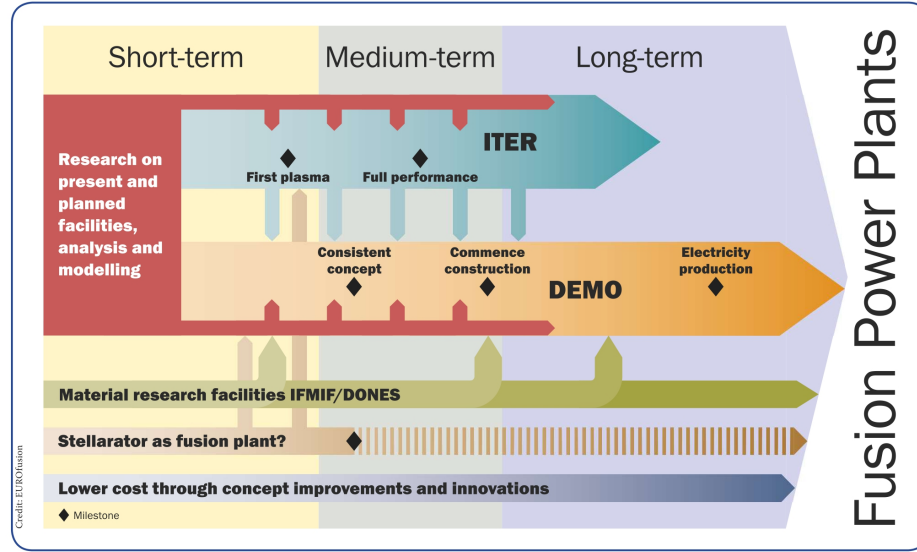


Figure 1.3: Summary of the European Roadmap toward the commercial energy by nuclear fusion [9].

Two machines in particular serve as the cornerstones of the roadmap. The International Thermonuclear Experimental Reactor (ITER) represents the largest tokamak currently under construction worldwide. The project aims to achieve several specific objectives. The primary goal is to demonstrate the scientific and technological feasibility of nuclear fusion as an energy source. Moreover, ITER will serve as a pivotal facility for the advancement and assessment of materials. The Demonstration Power Plant (DEMO), instead will be responsible for feeding electricity into the grid, operating with a closed fuel cycle. The roadmap is divided into three main stages:

- **First period (up to 2030).** The primary objectives of this phase can be summarized as follows: the completion of ITER and other ancillary projects, primarily JT-60SA [10], and the construction of IFMIF-DONES [11], a fusion neutron source. With regard to DEMO, the principal objective is the completion of the conceptual design of the device.
- **Second period (2031-2040).** The main goals are related to the exploitation of ITER up to its maximum performance and the preparation of DEMO construction.
- **Third period (beyond 2040).** By the conclusion of the final period, the exploitation of ITER is anticipated to have been completed. Furthermore, the IFMIF-DONES will be duly updated, and finally, the construction of the DEMO will have been completed.

1.6 Divertor Tokamak Test (DTT)

The roadmap is populated with a multitude of additional projects, each with a more circumscribed scope than ITER and DEMO. These projects are tasked with addressing specific issues in order to transfer the knowledge thus acquired into reference machines. In particular, the Divertor Tokamak Test experiment [12], currently under construction at the ENEA research centre in Frascati, and whose toroidal complex and vacuum vessel are the subject of the work presented in this thesis, will have the task of studying alternative divertor solutions aimed at heat extraction [13], which can then be adopted within the DEMO machine [14]. The current status of the design has been shared in [15].

1.6.1 DTT machine description

The DTT Vacuum Vessel (VV) is a double-walled torus of D-shaped configuration made of austenitic stainless steel, AISI 316LN (1.4406 in [16]), which comprises 17 sectors [17]. Of these, 14 are standard sectors, featuring a toroidal extension of 20° . Additionally, there are two sectors with a 30° toroidal extension and one special sector with a 20° toroidal extension. Normally, each sector features five ports, enumerated from one at the top to five at the bottom. However, in six sectors, the fifth port is replaced by a Gravity Support (GS), and the special sector is free of ports. Six gravity supports are equipped with spring plates to accommodate radial thermal expansion. Each gravity support is composed of a vertical leg and is positioned on a ring pedestal. An electrical insulation plate is situated between the gravity supports and the pedestal ring, serving to prevent the formation of current loops. At the upper extremity of the leg, a flange is attached to the lower portion of the vessel via a bolted joint. The VV ports are structurally connected and electrically isolated with respect to the Cryostat (CRS) port stubs. Expansion bellows are placed between the vessel and the cryostat, thereby ensuring the relative displacements produced by machine operating phases. The interspace between the shells of the vessel, which houses a system of poloidal and toroidal reinforcing ribs, can be filled with water that performs cooling and heating functions in accordance with the specific operating state. This water flows from the inlet located at port 5 and the outlet at port 1, enhancing the neutron shielding capability of the vessel [18]. In Figure 1.4 the toroidal complex is depicted, with the cryostat system included.

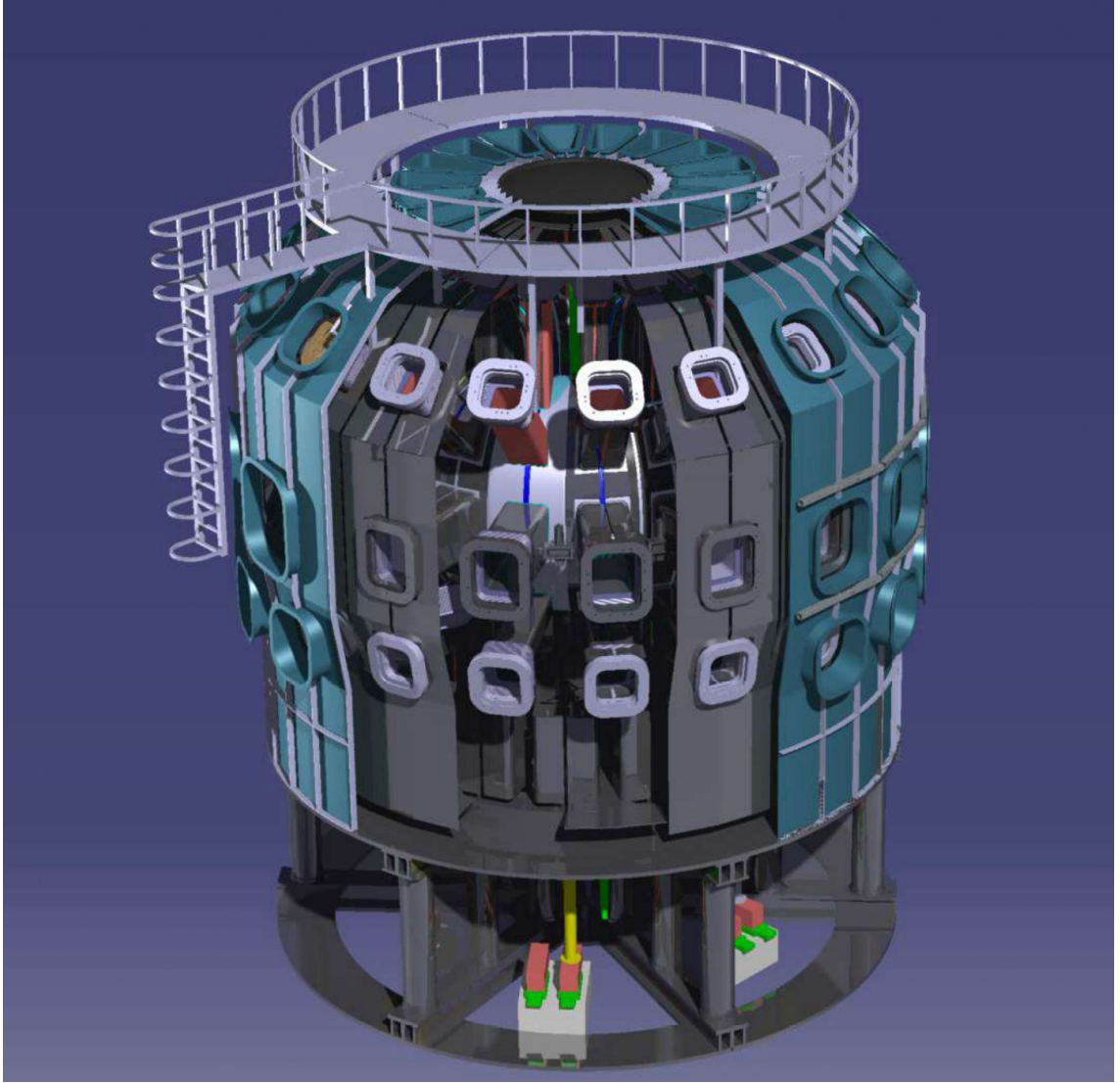


Figure 1.4: DTT toroidal complex overview.

The cryostat can be subdivided into three principal sub-assemblies: the Cryostat Base (CBS), the Cryostat Cylindrical Body (CCB), and the Cryostat Top Lid (CTL). The CCB is constructed with a single wall of AISI 304LN (1.4311 in [16]) material and is bolted to the main plate. The CBS is constituted by a main plate, comprising two welded sectors, which serve to connect the gravity supports of the VV. Additionally, the plate incorporates a support ring for the 18 gravity supports of the magnets, as well as apertures for the routing of port ducts and instrumentation. At the centre of the assembly, a booth with an inlet manhole is connected to the plate through a large circular opening. Vacuum tightness is guaranteed by the use of seal welds within the component. The plate is supported by six custom-profiled radial beams, with the wings of the section connected by three vertical cores. The area situated beneath the radial beams has undergone modifications, which have yet to be incorporated into the machine's overall CAD design and are therefore not visible

in Figure 1.4. A new system of 12 columns, two for each radial beam, is employed to provide support, with bolted joints connecting the beams and columns. Each pair of columns is equipped with a bracing system comprising a central plate that is welded to four cylindrical elements, which in turn are welded to flanges through which the bolted connection between the columns and the beams is made. Finally, the columns are fixed to the concrete foundation via anchor bolts [19].

1.7 Thesis objectives and structure

The engineering design of fusion machines is inherently complex, particularly in light of the multiphysics phenomena that occur during operation. The present work is focused on the realization of complex models aimed at studying the structural behavior of the DTT machine, considering stresses arising from thermal, mechanical, and electromagnetic loads. These stresses are considered both in normal operating states of the machine and in emergency situations, which are of greater relevance for structural integrity evaluations. It is not possible to conduct this type of assessment without a comprehensive understanding of the regulatory requirements pertaining to the design of such devices. In Chapter 2, a comparison is made between the ASME and RCC-MRx codes with respect to the aspects relevant to the presented work. This analysis identifies points of convergence and divergence between the two codes. The remaining chapters present the results of the analysis conducted on the machine at varying levels of detail regarding the system under consideration. Chapter 3 is focused on the toroidal machine complex as a whole, with a particular emphasis on the cryostat base system, where the machine's critical components are housed. The analysis is centred on the dynamic behavior of the complex, particularly on the cryostat base in terms of displacement and stresses during seismic events. Furthermore, a preliminary design of a seismic isolation system is presented. Chapter 4 addresses the thermo-mechanical analysis of the vacuum vessel, which was examined for structural integrity comparing both the codes described in Chapter 2. Chapter 5 is dedicated to a detailed examination of a specific sub-assembly of the vacuum vessel, namely the Gravity Support (GS) system. In particular, the chapter focuses on the analysis and design optimization of the bolted joint that interfaces the GS and the VV.

1.8 Disseminations of research results

The work carried out during the three years of PhD and reported in the following thesis has been presented at various conferences related to the nuclear fusion research:

- FuseNet PhD Event 2022:

M. Fulici et al., *Thermo-mechanical analyzes and verification of DTT vacuum vessel*

- 32nd Symposium on Fusion Technology (SOFT2022):
M. Fulici et al., *Thermo-structural coupled analyzes and verification of the DTT vacuum vessel*
- 15th International Symposium on Fusion Nuclear Technology (ISFNT 2023)
M. Fulici et al., *Effects of seismic isolation on the dynamic behavior of the DTT torus complex during earthquake events*

Parts of this work have been also included in publications in scientific journals:

- F. Giorgetti et al., *Vertical displacement events analysis using MAXFEA code in combination with ANSYS APDL in the final design stage of the DTT vacuum vessel*, Fusion Eng. Des., 184, August 2022, (<https://doi.org/10.1016/j.fusengdes.2022.113273>)
- M. Fulici et al., *Effects of seismic isolation on the dynamic behavior of the DTT torus complex during earthquake events*, Fusion Eng. Des., 203, April 2024, (<https://doi.org/10.1016/j.fusengdes.2024.114476>)
- F. Romanelli et al., *Divertor Tokamak Test facility project: status of design and implementation*, Nucl. Fusion, 64, September 2024, (<https://doi.org/10.1088/1741-4326/ad5740>)

Chapter 2

Multicode approach for the design of nuclear fusion facilities

This chapter aims to provide an overview of the main codes used in the design of components and elements employed within nuclear facilities, namely the ASME code and the RCC-MRx code, highlighting the similarities and differences between them. The procedure used for the design of components, which will be detailed in the following chapters, are based on a Design By Analysis approach (DBA). Such kind of approach relies on techniques such as Finite Element Analysis (FEA) to ensure design meets specific safety and performance criteria. In the chapter, the contents of the codes will be presented limited to the type of analysis performed, namely linear static analyses. Further details can be found directly in the standards.

As for the ASME code, the sections referenced are as follows:

- *Section III - RULES FOR CONSTRUCTION OF NUCLEAR FACILITY COMPONENTS Division 1 - Subsection NB Class 1 components* [20]
- *Section III - RULES FOR CONSTRUCTION OF NUCLEAR FACILITY COMPONENTS Subsection NCA General Requirements for Division 1 and Division 2* [21]
- *Section VIII - RULES FOR CONSTRUCTION OF PRESSURE VESSEL Division 2 Alternative Rules* [22]

The Subsection NCA has been used as reference for the Service Level definition whilst in Appendix 5.A of Section VIII the stress linearization procedure is detailed. The elements for carrying out DBA according to the RCC-MRx are all contained in a single document, which is the *Section III - Tome 1 - Subsection B: CLASS N1 Rx REACTOR COMPONENTS, ITS AUXILIARY SYSTEMS AND SUPPORTS* [23]. In the design of the components described in the following chapters, additional codes were also used as a supplement:

- *Eurocode 3* [24] for the design of flanged joints, which is detailed in Chapter 5
- *Eurocode 8* [25] and the italian code *Norme tecniche per le costruzioni* [26] for seismic design of the DTT toroidal complex, which is detailed in Chapter 3.

These codes are not included in this chapter as they are not specific to nuclear facilities and the aspects useful in the design of the specific components are directly mentioned in the related chapters.

2.1 ASME vs RCC-MRx

This section presents a comparative analysis of the two reference codes utilized in the design of nuclear facilities.

2.1.1 Design by analysis

2.1.1.1 Analysis related terms

The stresses obtained by elastic analysis can be classified in primary, secondary and peak stresses.

- **Primary stress**

ASME

Primary stress is any normal stress or shear stress developed by an imposed loading that is necessary to satisfy the laws of equilibrium of external and internal forces and moments. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure or, at least, in gross distortion.

RCC-MRx

The primary stress is defined as the fraction of the total stress which does not disappear after small permanent deformation. Small deformation is taken to mean deformation which does not lead to either appreciable geometrical modifications (large displacements) nor to notable elongation (large local deformation).

The primary stresses can be divided into:

- **General membrane stress (P_m)**

ASME

A general primary membrane stress is one that is so distributed in the structure that no redistribution of load occurs as a result of yielding.

RCC-MRx

The general primary membrane stress is the mean value of the primary stress tensor within the thickness of the wall.

– **Local membrane stress (P_L)**

ASME

Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a discontinuity would, if not limited, produce excessive distortion in the transfer of load to other portions of the structure.

RCC-MRx

Close to a geometrical or loading discontinuity, that is in a small zone adjoining the discontinuity, mechanical stress (elastic follow-up will be included in the mechanical loads) should be distinguished from among the stresses which are not primary stresses; the membrane stress associated with this stress is noted L_m . The local primary membrane stress P_L is the stress equal to the sum of stress L_m and the general primary membrane stress P_m : $P_L = P_m + L_m$.

– **Bending stress (P_b)**

ASME

Bending stress is the component of normal stress that varies across the thickness. The variation may or may not be linear.

RCC-MRx

Primary bending stress is the stress distributed linearly within the thickness which has the same moment as the primary stress.

• **Secondary stress**

ASME

Secondary stress is a normal stress or a shear stress developed by the constraint of adjacent material or by self-constraint of the structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur and failure from one application of the stress is not to be expected.

RCC-MRx

Secondary stress is the fraction of the total stress which can disappear as a result of small permanent deformation, minus the peak stresses.

• **Peak stress**

ASME

Peak stress is that increment of stress that is additive to the primary plus secondary stresses by reason of local discontinuities or local thermal stress including the effects, if any, of stress concentrations. The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture.

RCC-MRx

The peak stress is the fraction of the total stress which meets the following two conditions: it is the additional stress applied by a geometrical discontinuity of the structure or by non linearity in the distribution of stresses within the thickness, this additional stress, generally very localized, cannot, should it be redistributed, cause deformation of the whole structure.

- **Total stress** The total stress is the sum of the primary, secondary, and peak stress contributions.

2.1.2 Failure mechanism related to stress categories

According to [27], the established limits on primary stresses are designed to prevent plastic deformation. To avoid excessive distortion a limit of S_y was considered for local primary stresses. This limit is 50% higher than the limit on general membrane stress, though it allows to prevent from excessive yielding. The primary plus secondary stress limits are intended to prevent excessive plastic deformation leading to incremental collapse. This limit ensures that the structure is able to shake down to an elastic state within few cycles, even if the stress range exceeds the yielding. In the end the limit is to prevent that incremental plastic growth occurs [28]. Usually collapse and excessive plastic deformation are used interchangeably. In [28] the two concepts are defined separately. The collapse is the condition of unbounded deformation, whilst excessive plastic deformation is defined as any permanent plastic deformation caused by mechanical loads that will not shake down under repeated application. Should the yield strength be exceeded, only strain-hardening, gross plastic deformation, or restructuring of the geometry could prevent the structure from collapse. Nonetheless, materials that undergo strain-hardening under monotonic loads may not necessarily undergo the same process under cyclic loads. Furthermore, if these materials undergo strain-hardening under cyclic loads, it does not necessarily imply that they will undergo shake-down. In this regard, primary stresses differ from secondary stresses. Looking at Figure 2.1 [27], in the case of unloading, secondary stresses that exceed yielding produce a residual stress state without permanent plastic deformation (a). In contrast, in the case where primary stresses exceed yielding, following unloading there are no residual stresses but a permanent plastic deformation that accumulates in the case of cyclic

loading and shake down will not occur even if the sum of primary and secondary stresses is less than $2S_y$, which represents the maximum value of calculated secondary elastic stress which will shake down to purely elastic action (b).

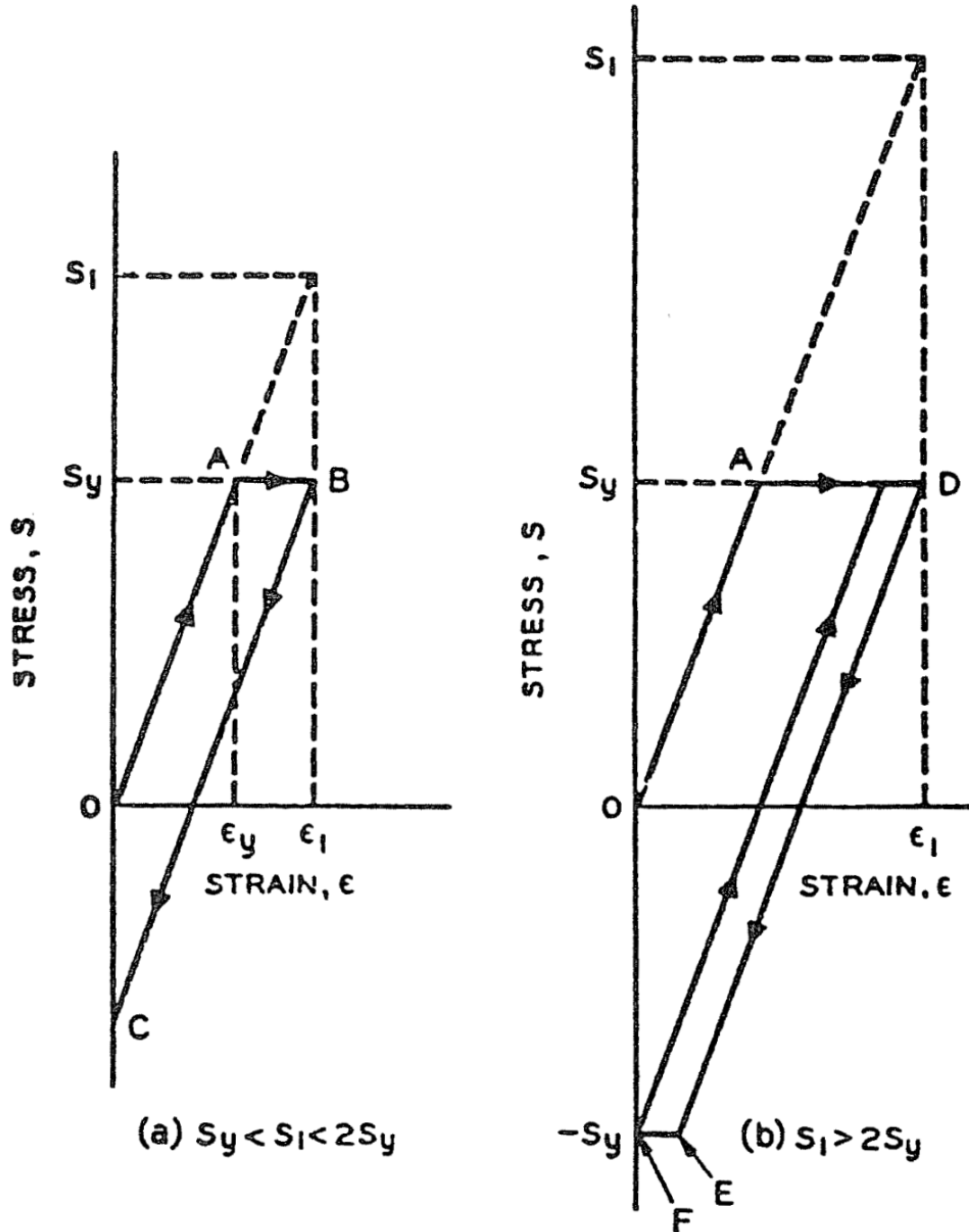


Figure 2.1: Strain history beyond yield. Shake down explanation [27].

2.1.3 Service Level approach

According to ASME Section III Subsection NCA, components and supports of a nuclear power system may be subjected to plant and system operating and test conditions that are required to be considered in the design and overpressure protection of the components and the design of supports in order to satisfy applicable systems safety criteria. Plant and system operating conditions are commonly referred to as normal, upset, emergency, and faulted. Service Level A is associated to normal operating conditions, the Level B to upset operating conditions, the Level C to emergency conditions and finally the Level D to faulted conditions. As for the ASME Code, Level B criteria are identical to Level A criteria except that the allowable primary stresses may be increased to 110% of their Level A values when an Upset pressure loading exceeds the Design pressure. Following a conservative approach the RCC-MRx removes the distinction between Level A and Level B, with the latter not mentioned in the code. The service levels of interest in this work are Level A and Level C. According to the RCC-MRx, the aim of Level A criteria is to protect the component against the following types of damage:

- immediate or time-dependent excessive deformation,
- immediate or time-dependent plastic instability,
- time-dependent fracture,
- elastic or elastoplastic instability, immediate or time-dependent,
- progressive deformation,
- fatigue.

The aim of Level C criteria is to protect the component against the following type of damages:

- immediate or time-dependent excessive deformation,
- immediate or time-dependent plastic instability,
- time-dependent fracture,
- elastic or elastic-plastic instability, immediate or time-dependent.

RCC-MRx states that when a component has been subjected to loadings which, whilst satisfying Level C criteria, exceed Level A criteria, it is possible that small levels of overall deformation could occur. Consequently, it could be necessary to inspect a component subjected to these types of loading before re-using it, to ensure that its condition is satisfactory. About the Level C criteria, it is recalled that fatigue

analyses are not required and consequently the number of stress cycles which are only limited by Level C criteria, should not exceed 10.

The ASME code doesn't give specific definitions or aims about the service levels as the RCC-MRx does. Nonetheless it gives a better description of the limits in terms of stress cycles for the definition of Level C criteria. For all specified service conditions for which Level C Limits apply the loads shall not cause more than 25 stress cycles having an S_a value greater than that for 10e6 cycles from the applicable fatigue design curves.

2.1.4 Stress limits for components

Different service levels correspond to different stress limits. The allowable values depend on the membranal stress, denoted by S_m for components and S_{mB} for bolts. In general, S_m is a quantity that is a function of material, temperature and, in the case of RCC-MRx, the irradiation. The limits presented below are also function of the minimum yielding strength S_y , in the case of the ASME code, and the minimum tensile strength $(R_m)_{min}$, for the RCC-MRx.

2.1.4.1 Allowable stresses

ASME

As for ASME *II Appendix 2*, the value of the allowable stress, S_m , is equal to the minimum of the following quantities:

- 2/3 for the yield strength at 20°C;
- 2/3 for the yield strength at temperature or 0.9 for the yield strength in case of austenitic stainless steel with a ratio between the yield and the tensile strength less than 0.625;
- 1/3 for the tensile strength at 20°C;
- 1/3 for the tensile strength at operating temperature.

RCC-MRx

As for RCC-MRx *Appendix A3.GEN*, in case of austenitic stainless steels the value of the allowable stress, S_m , is equal to the minimum of the following quantities:

- 2/3 for the yield strength at 20°C;
- 0.9 for the yield strength at temperature;
- 1/3 for the tensile strength at 20°C;
- 1/3 for the tensile strength at operating temperature.

For other than austenitic stainless steel materials the value of the allowable stress, S_m , is equal to the minimum of the following quantities:

- 2/3 for the yield strength at 20°C;
- 2/3 for the yield strength at temperature;
- 1/3 for the tensile strength at 20°C;
- 1/2.7 for the tensile strength at operating temperature.

2.1.4.2 Level A limits

ASME

Considering the ASME code, in case of service level A no specific limits are set for primary stresses. Instead, the limit prescribed is for the sum of primary and secondary stresses which is equal to $3S_m$. The following relationship applies:

$$\overline{P_L + P_b + Q} \leq 3S_m \quad (2.1)$$

RCC-MRx

The RCC-MRx prescribes limits on the primary stresses which are equal to S_m for the general membrane stress and $1.5S_m$ for the local membrane and for the membrane plus bending stresses. The following relationship apply:

- General primary membrane intensity

$$\overline{P_m} \leq S_m(\theta_m) \quad (2.2)$$

- Local primary membrane intensity

$$\overline{P_L} \leq 1.5S_m(\theta_m) \quad (2.3)$$

- Primary membrane plus bending stress intensity

$$\overline{P_L + P_b} \leq 1.5S_m(\theta_m) \quad (2.4)$$

where $S_m(\theta_m)$ designates the value of the allowable stress for the mean temperature in the thickness during the loading in question.

2.1.4.3 Level C limits

Even with respect to service level C, the limits proposed by the two regulations differ from each other, although both provide specific limits on primary stresses.

However, no limits are provided for secondary stresses.

ASME

- General membrane

$$\overline{P}_m \leq \max(1.2S_m, S_y) \quad (2.5)$$

- Local membrane

$$\overline{P}_L \leq \max(1.8S_m, 1.5S_y) \quad (2.6)$$

- Bending

$$\overline{P}_b \leq \max(1.8S_m, 1.5S_y) \quad (2.7)$$

RCC-MRx

- General primary membrane intensity

$$\overline{P}_m \leq \min(1.35S_m, (R_m)_{\min}) \quad (2.8)$$

- Local primary membrane intensity

$$\overline{P}_L \leq 1.5 \cdot \min(1.35S_m, (R_m)_{\min}) \quad (2.9)$$

- Primary membrane plus bending stress intensity

$$\overline{P}_L + \overline{P}_b \leq 1.5 \cdot \min(1.35S_m, (R_m)_{\min}) \quad (2.10)$$

2.1.5 Stress limits for threaded fasteners

The RCC-MRx limits presented below refer to preloaded bolted assemblies not maintaining the leaktightness, as the bolted joints analyzed in this work. The ASME *Section III, Subsection NG*, as reported here, is not dedicated to prevent coolant leakage. Thus one can assume such ASME limits applicable to the same category of bolted assemblies mentioned above as for the RCC-MRx. The limits for threaded fasteners are also function of the minimum yielding strength S_y (ASME), $R_{p0.2}^t$ (RCC-MRx), and the minimum tensile strength S_u (ASME), $(R_m)_{\min}$ (RCC-MRx).

2.1.5.1 Allowable stresses

ASME

According to the ASME *Section II - Part D*, the allowable value, S_m , for bolting with

strenght enhanced by heat treatment or strain hardening is equal to the minimum between the two quantities:

- 1/3 for the yield strength at room temperature;
- 1/3 for the yield strength above room temperature.

RCC-MRx

As for RCC-MRx *Appendix A3.GEN*, the value of the allowable stress, S_{mB} , is equal to the smallest of the following quantities:

- 1/3 for the yield strength at 20°C;
- 1/3 for the yield strength at operating temperature.

2.1.5.2 Level A limits

ASME

- **Shank or threads**

- Average primary membrane including stress from preload

$$\overline{P_m} \leq S_m \quad (2.11)$$

- Average primary plus secondary membrane including stress from preload

$$\overline{P_m + Q_m} \leq \min(0.9S_y, 2/3S_u) \quad (2.12)$$

- Primary plus secondary membrane and bending

$$\overline{P_m + Q_m + P_b + Q_b} \leq 1.33 \cdot \min(0.9S_y, 2/3S_u) \quad (2.13)$$

- **Thread shear**

- Average primary plus secondary membrane including stress from preload

$$\overline{P_m + Q_m} \leq 0.6S_y \quad (2.14)$$

- **Bearing under head**

- Average primary plus secondary membrane including stress from preload

$$\overline{P_m + Q_m} \leq 2.7S_y \quad (2.15)$$

For high-strength structural fasteners, $S_u \geq 690 \text{ MPa}$, the limits of Service Level A also apply.

RCC-MRx

- **Rules for the screw core**

- Limitation of the mean stress due only to mechanical loads

$$\overline{\sigma_m} \leq S_{mB} \quad (2.16)$$

- Limitation of the mean stress

$$\overline{\sigma_m} \leq \min(0.9(R_{p0.2}^t)_{\min}, 0.67(R_m)_{\min}) \quad (2.17)$$

- Limitation of maximum stress

$$\overline{\sigma_m + \sigma_b} \leq 1.33 \cdot \min(0.9(R_{p0.2}^t)_{\min}, 0.67(R_m)_{\min}) \quad (2.18)$$

- **Rules for screw threads and head**

- Limitation of the shear stress due only to mechanical loads

$$\tau_{fv} \leq 0.6 S_{mB} \quad (2.19)$$

$$\tau_t \leq 0.6 S_{mB} \quad (2.20)$$

- Limitation of the shear stress

$$\tau_{fv} \leq 0.6 (R_{p0.2}^t)_{\min} \quad (2.21)$$

$$\tau_t \leq 0.6 (R_{p0.2}^t)_{\min} \quad (2.22)$$

- Limitation of the bearing stress

$$p_t \leq 2.7 (R_{p0.2}^t)_{\min} \quad (2.23)$$

2.1.5.3 Level C limits

ASME

- Primary membrane

$$\overline{P_m} \leq 1.5 S_m \quad (2.24)$$

- Primary membrane plus bending

$$\overline{P_m + Q_m} \leq 2.25S_m \quad (2.25)$$

RCC-MRx

- Rules for the screw core
 - Limitation of the mean stress due only to mechanical loads

$$\overline{\sigma_m} \leq 1.5S_{mB} \quad (2.26)$$

- Limitation of maximum stress due only to mechanical loads

$$\overline{\sigma_m + \sigma_b} \leq 2.25S_{mB} \quad (2.27)$$

For high strength screws, $(R_m)_{min} \geq 700MPa$, rules for Level A, in case of all loadings, shall be satisfied.

- Rules for screw threads and head

For high strength screws, $(R_m)_{min} \geq 700MPa$, rules for Level A, in case of all loadings, shall be satisfied.

2.2 Stress linearization

General guidelines for the application of the stress linearization procedure are given in [22] and a detailed summary can be found in [29]. The stresses obtained from the finite element analyses represent the total stresses. The stress linearization allows to obtain from the total stress the membrane, bending and peak components to make the comparison with the allowable values from the standards. Stress classification is necessary for membrane and bending stress calculations. This can be done according to three different methods, in ascending order of accuracy: stress at a point, stress along a line, and stress on a plane [30]. Point stresses can be used for comparison with allowable stresses only for very simple geometries and loads. Only the second two approaches are reported in the [22]. In the case of stress along a line, when considering an axisymmetric revolution shell, the line through the shell thickness is a plane. In the case of three-dimensional models, however, the association of a line with a plane is not immediate. This aspect is directly related to the difficulty of identifying a bending plane and thus predicting failure in such a case. Classification of stresses along a plane is the most general and potentially the most accurate. In contrast, it is very complex to generalize the location,

orientation, and extent of the plane, especially for articulated geometries and loading conditions. Regions of structural discontinuity where the highest stresses are located are preferred for placement of SCLs or SCPs. In general, SCLs should be normal to the contour lines of the highest stress component. Since this is difficult to achieve, particularly in complex geometries, a common practice is to orient SCLs normally to the mid-surface of the cross section. The choice of the stress components to be linearized is an open issue. In the present work, it was decided to use all 6 components of the stress tensor. Given a cross section in the $x_1 - x_3$ plane with normal in the x_2 direction and considering a SCL oriented along x_3 , the membrane and bending stresses for each component of the stress tensor are calculated as follows:

$$\sigma_{ij}^m = \frac{1}{t} \int_{-t/2}^{t/2} \sigma_{ij} dx_3 \quad (2.28)$$

$$\sigma_{ij}^b = \frac{x_3}{I} \int_{-t/2}^{t/2} \sigma_{ij} x_3 dx_3 \quad (2.29)$$

where t is the length of the SCL.

When linearized on the plane, the following relations apply:

$$\sigma_{ij}^m = \frac{1}{A} \int_A \sigma_{ij} dA \quad (2.30)$$

$$\sigma_{ij}^b = \frac{x_3}{I} \int_A \sigma_{ij} x_3 dA \quad (2.31)$$

Once the membrane and bending tensors are obtained, it is possible to calculate the equivalent stresses associated with them and compare these values with the allowable limits. In the ASME code Section III, the stress intensity is calculated according to the maximum shear method (Tresca) as per Equation (2.32), while in the RCC-MRx, the octahedral shear method (von Mises), as per Equation (2.33), is also allowed.

$$\bar{\sigma}_T = \max(|\sigma_1 - \sigma_2|, |\sigma_2 - \sigma_3|, |\sigma_3 - \sigma_1|) \quad (2.32)$$

$$\bar{\sigma}_{VM} = \frac{1}{\sqrt{2}} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} \quad (2.33)$$

Chapter 3

Seismic analysis and seismic isolation design of the DTT torus complex

Aside from disruption events, whose effects on the DTT vacuum vessel will be presented in Chapter 4, seismic events pose some of the most severe loads for fusion machines. The literature contains numerous examples of seismic analysis conducted on fusion machines and their components. Seismic analysis of the CFTR and EAST vacuum vessel are reported in [31] and [32], respectively. In [33], a detailed model of the KSTAR tokamak is analyzed, including the cryostat and ports in addition to the vacuum vessel. Seismic analysis can be carried out using an equivalent static approach or dynamic methods. The results comparison of these methods in their application on fusion machines is given in [31] and [33]. The approach outlined in [31] employs an equivalent static method, wherein the peak acceleration extracted from the response spectra at the system's natural frequency is amplified by a pre-determined dynamic amplification factor. This procedure is slightly different from the one foreseen by the Italian standard used for the analysis presented here, *Norme tecniche per le costruzioni (NTC2018)* [26]. The dynamic method used in the aforementioned works relies on response spectrum analysis, combining the results of modal analysis, considering that these are capable to describe the total dynamic behavior of the structures. Nonlinear dynamic analysis considering the whole time evolution of the spectra are also frequently employed, as done in [34], for the ITER tokamak support system, giving the best accuracy but a greater computational burden. In this thesis both the equivalent static approach and the response spectrum analysis were used to evaluate the seismic effects on the DTT torus complex, with a particular focus on the cryostat base which is the main component analyzed in this chapter. The identification of potential structural vulnerabilities prompted the consideration of integrating a seismic isolation system at the base. Seismic isolation is a technique widely used in the field of civil engineering, aiming at decoupling

the structures from the ground by interposing a system with a low horizontal stiffness between them [35]. The variety of seismic isolation bearings, and their main characteristics, is well discussed in [36]. The main focus of the work presented in the chapter was on the advantageous effects of integrating an isolation system, particularly in terms of stress reduction. Seismic isolation is not new in the fusion field. The ITER tokamak features an isolation system integrated in the tokamak pit [37]. In this chapter a different perspective on seismic isolation, prioritizing stress reduction over examining relative displacements between components is presented.

3.1 System description and seismic site details

The cryostat base is shown in Figure 3.1. It consists of a horizontal plate to which the vacuum vessel is fixed. It also houses a reinforcing ring to which the magnets gravity supports are fixed. Beneath the plate are six radial beams, each one supported by two pillars anchored to the ground. A bracing system connects each pair of pillars. Two different seismic isolation solutions were analyzed. The first one (green box in Figure 3.2) by placing the isolation plane at the base of the pillars (BISO) and the second one (red box in Figure 3.2) by placing it at the top of the pillars (TISO), below the radial beams. The isolation plane divides the machine into two distinct parts: the substructure, the part below the isolation plane, which experiences the ground seismic spectra, and the superstructure, the part above, which is actually isolated from the ground. To ensure the correct functioning of the isolation system, the pillars are rigidly connected by a frame structure where the isolators are placed. Such structure was considered just in the FE model and it is not already integrated in the design of the cryostat base. From here on, we will refer to the non-isolated configuration as NISO. The DTT machine has been classified of importance class IV, which indicates structures with an important strategic function [26], thus site-specific spectra are needed for the design [25]. The location of the DTT machine can be deduced by Figure 3.3 (red arrow), showing also the values of the maximum ground acceleration of the site. Considering the importance class IV and a peak acceleration between 0.15 g and 0.2 g the seismic hazard level assigned to this project is high. These considerations were the input for the calculation of the seismic local response. The horizontal and vertical spectra used in the analysis are that of the so-called “*Stato limite di collasso (SLC)*”, whose meaning is collapse limit state, the most severe case in terms of seismic action with a return period of 1950 years, and are shown in Figure 3.6.

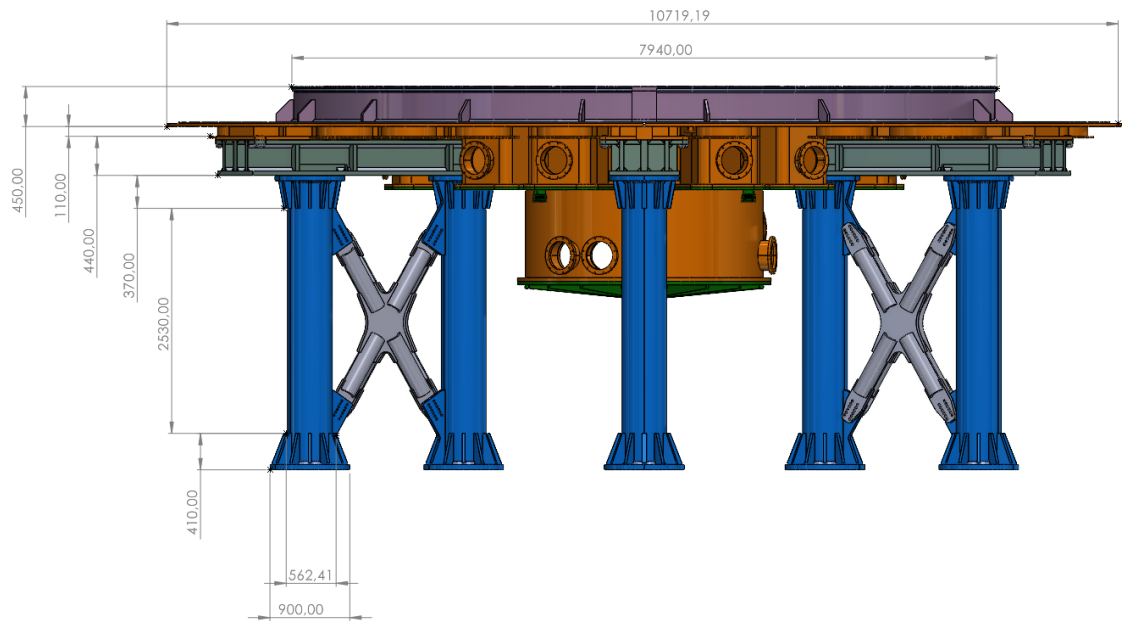


Figure 3.1: DTT Cryostat base. The geometric dimensions are in mm.

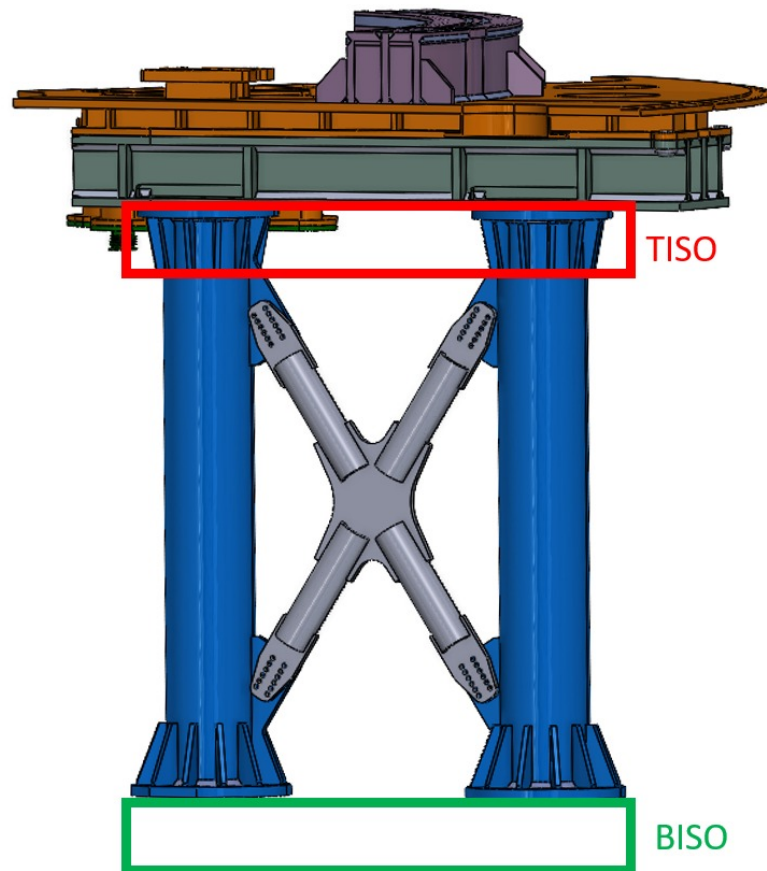


Figure 3.2: 60° radial-vertical cross-section of the cryostat base with the considered positions for the isolators: BISO and TISO.

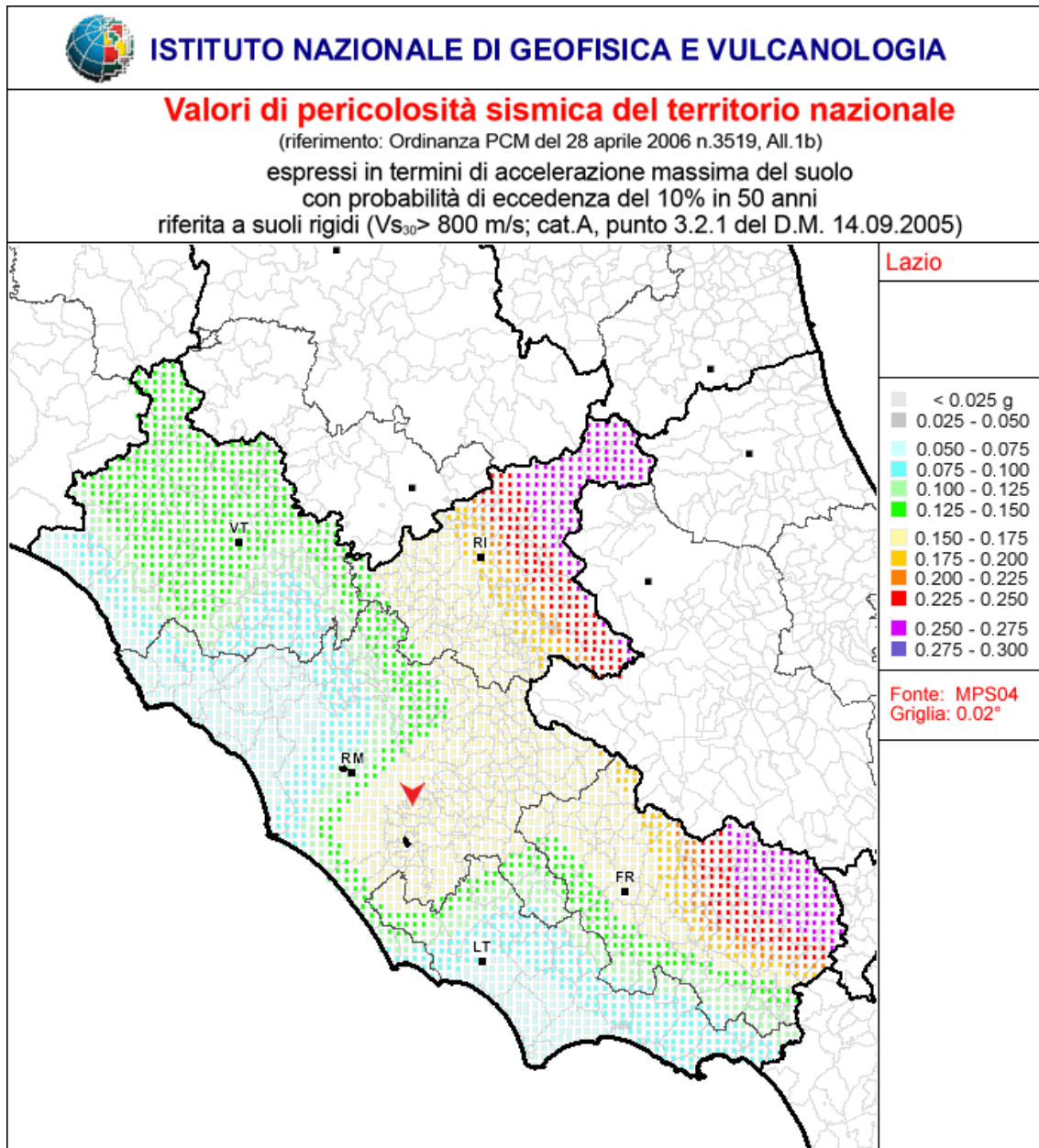


Figure 3.3: Map of the seismic hazard taken from the website of the “*Istituto nazionale di geofisica e vulcanologia (INGV)*” [38]. The DTT site is identified by the red arrow.

3.2 Methodology

3.2.1 Equivalent static method

The equivalent static method, as per [26], foresees the application of an equivalent static force on each mass of the model, in the three Cartesian directions, calculated as follows:

$$F_i = F_h \cdot z_i \cdot \frac{W_i}{\sum_j z_j W_j} \quad (3.1)$$

where:

- F_i is the force acting on the i_{th} component;
- F_h is a function of the spectrum acceleration evaluated at the natural frequency of the system;
- z_i and z_j are the heights of the centre of mass of the i_{th} and j_{th} components, respectively;
- W_i and W_j are the weights of the i_{th} and j_{th} components, respectively.

Once the analysis have been carried out in the three Cartesian directions, the seismic effects (E_x, E_y, E_z) are combined to calculate the total response of the system according to the following load combinations [26],[25]:

$$LC_1 = E_x + 0.3 \cdot E_y + 0.3 \cdot E_z \quad (3.2)$$

$$LC_2 = 0.3 \cdot E_x + E_y + 0.3 \cdot E_z \quad (3.3)$$

3.2.2 Linear dynamic method

The seismic analyses were also performed using a linear dynamic method, namely a response spectrum approach. This method consists of 3 fundamental steps:

1. Modal analysis
2. Modes expansion
3. Spectrum analysis

Following the modal analysis, the selection of modes to be expanded was carried out according to the Italian code for seismic analysis *Norme tecniche per le costruzioni (NTC2018)* [26]. The code prescribes that only those modes characterised by a mass participation factor of more than 5% should be expanded,

up to a total mass participation factor of more than 85%. The expanded modes are then used to obtain the seismic effects in the three Cartesian directions (E_x, E_y, E_z). This is done using a Complete Quadratic Combination (CQC) method. Finally, as done for the equivalent static approach, E_x , E_y and E_z are combined to calculate the total response of the system using Equation (3.2) and Equation (3.3).

3.2.3 Seismic isolation: analytical model

The preliminary assessment for the isolation system was carried out analytically starting from the 1 DOF model, showed in Figure 3.4.

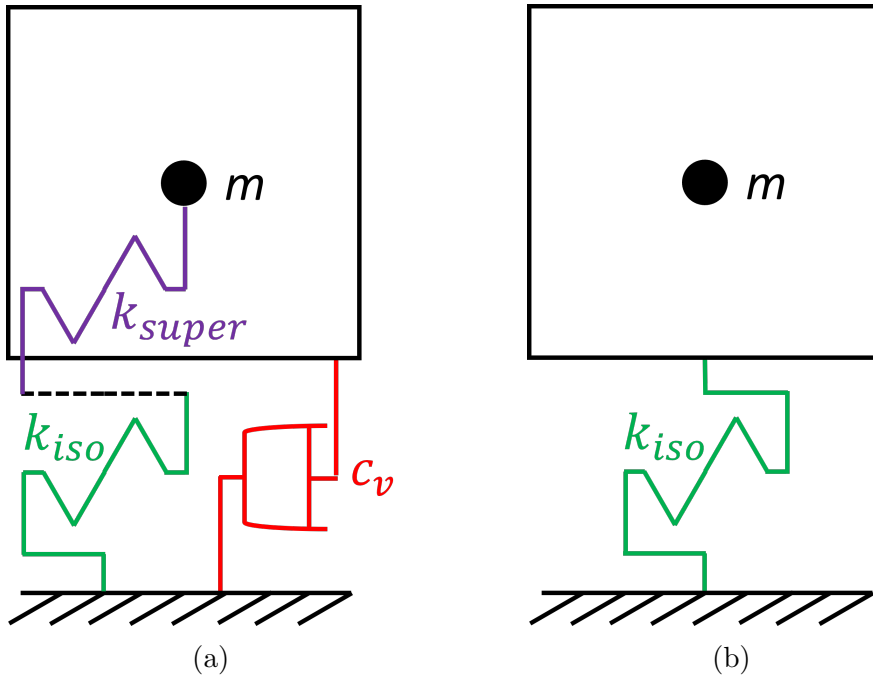


Figure 3.4: 1 DOF model. Reference (a), simplified (b)

The 1 DOF model is described by Equation (3.4):

$$m\ddot{x} + c_v\dot{x} + k_{eq}x = 0 \quad (3.4)$$

where:

$$\frac{1}{k_{eq}} = \frac{1}{k_{iso}} + \frac{1}{k_{super}} \quad (3.5)$$

In Equation (3.5), k_{iso} is the horizontal stiffness of the isolation system, and k_{super} is the stiffness of the superstructure. As a first approximation, the equation can be simplified by setting $k_{eq} = k_{iso}$, assuming that $k_{super} \gg k_{iso}$. Furthermore, damping associated to the isolators can be neglected in this stage. The equation can thus be simplified to:

$$m\ddot{x} + k_{iso}x = 0 \quad (3.6)$$

and the relative model is depicted in Figure 3.4b. The first estimation of k_{iso} was done assuming an isolation period T_{iso} equal to 2.5 s, which corresponds to a frequency of 0.4 Hz. This is the isolation period of the isolated system from which a first acceleration value can be derived directly from the spectrum. Having defined the desired isolation period, the stiffness of the isolating system can be calculated as follows:

$$k_{iso} = m \cdot \left(\frac{2\pi}{T_{iso}} \right)^2 \quad (3.7)$$

The vertical stiffness is assumed to be 800 times the horizontal stiffness [26].

3.3 FE model

The finite element model used for the analysis is showed in Figure 3.5. Being the cryostat and the base essentially made by plates and rolled shapes with small thicknesses with respect to the characteristic dimensions, as can be see in Figure 3.1, they were modelled using shell elements (SHELL181). The connections between the cryostat base pillars and the bracing systems are bolted joints. Rigid elements and radial beam arrays (BEAM188) were used to model the bolted connections in a simplified way. The vacuum vessel, the components inside the vessel, and the magnets were modelled using mass elements (MASS21). Each mass element was placed in correspondence with the centre of mass of the component it represents. The total mass of the model is 1177.7 t, and the mass values of each component are shown in Table 3.1.

Table 3.1: Mass values of the DTT components considered in the FE model.

Component	Mass [t]
Vacuum vessel	156.8
In-vessel components	51.5
First wall	9.5
Thermal shield	27.0
Cryostat	113.0
Poloidal field coils	118.9
Toroidal field coils	448.0
Central solenoid	53.0
Cryostat base	136.0
Gravity supports	64.0

The actual connection between components was modelled by connecting their

respective mass elements together through rigid beam elements (BEAM188). Thus the mass elements were linked to the base using constraint equations (RBE3).



Figure 3.5: FE model of the torus complex.

The seismic isolation was realised considering one isolator for each pillar of the cryostat base, for a total of 12 isolators. For the modelling of the isolators, pairs of coincident nodes were generated on the pillars axis and a triad of orthogonal linear spring elements (COMBIN14) were inserted between them to represent the two horizontal and vertical stiffnesses. In the BISO case, one of the two nodes was constrained, representing the connection between the isolator and the ground. The second node, representing the interface between the isolator and the base of the pillar, was connected by rigid beam elements to the nodes of the base. In this way, the pillars would be decoupled from each other. To avoid this, a rigid frame structure

was inserted into the FE model between the pillar-side isolator nodes. The modelling of the TISO case is similar, with the difference that neither of the two nodes used for representing the sides of the isolator is constrained. Both are connected by rigid beam elements (BEAM188) to the pillar-side and beam-side nodes respectively, after decoupling the mesh of the two components. Also in this case, the pillar-side nodes are connected by means of a rigid beam structure to achieve the same result as illustrated above. The horizontal stiffness used for the spring elements in the x and y directions was calculated according to Equation (3.7), and is equal to 0.617 kN/mm for the configuration with the isolators under the pillars, and 0.614 kN/mm for the configuration with the isolators above the pillars. Due to the small difference in stiffness, an average value of 0.6155 kN/mm was assumed for both configurations, also for ease of results comparison. The vertical stiffness was assumed to be equal to 469 kN/mm, 800 times higher than the horizontal one, as stated in Section 3.2.3. The horizontal and vertical spectra used for the calculations are showed in Figure 3.6. The vertical spectrum was obtained by scaling the horizontal one as specified in the reference standard [26]. The damping factor ξ of the isolators was assumed to be 15% and its effect was considered by scaling the spectrum with a multiplicative coefficient calculated as follows:

$$\eta = \sqrt{\frac{10}{(5 + \xi)}} \quad (3.8)$$

The scaled spectra can be seen in Figure 3.6 on the left side of the figure with respect to the vertical red line, designated as f_{damp} . Also the effect of the seismic isolation on the natural frequency of the system is highlighted in the same figure, shifting from f_{niso} to f_{iso} value.

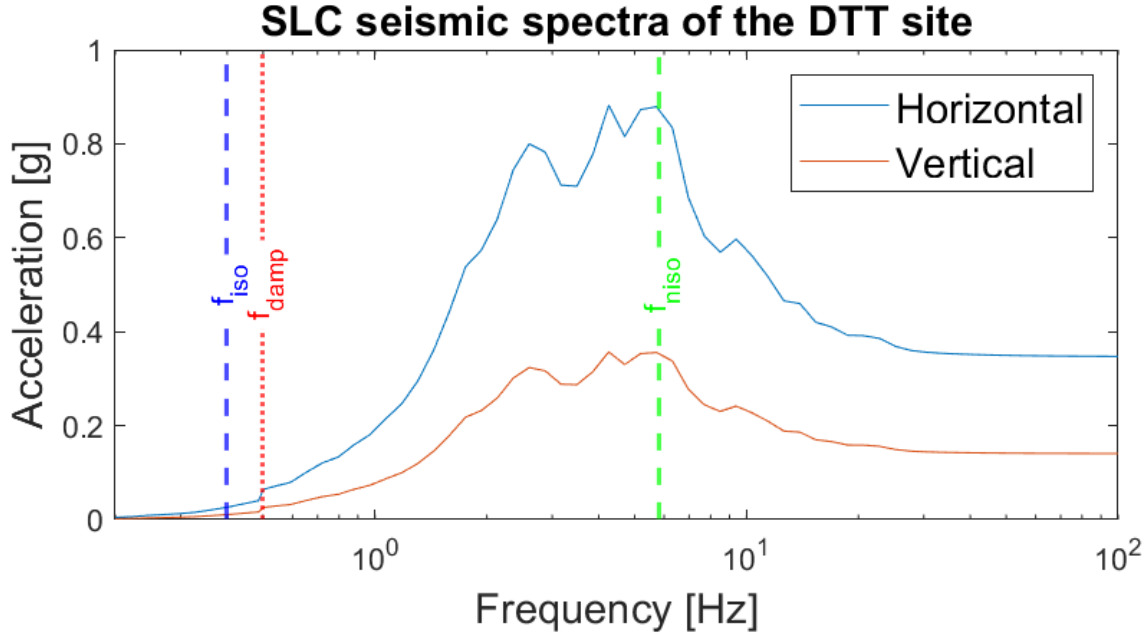


Figure 3.6: Horizontal and vertical spectra (SLC) of the DTT site. The image shows also the changing of the system natural frequency, once the isolation is included, and the effect on the spectra of the damping associated to the isolators.

3.4 Results

3.4.1 Equivalent static vs linear dynamic method

3.4.1.1 Equivalent stress field (von Mises)

In Section 3.4.1.1 a comparison between the two methodologies in terms of the max von Mises stress is presented, while Section 3.4.1.1 points out the differences of the stress field. The allowable stress is assumed to be $\sigma_{yield}/1.05$ according to NTC2018 [26]. Considering the yielding stress of the AISI 304L at room temperature equal to 180 MPa [39], the allowable stress is 171.43 MPa. In general, it can be seen that the equivalent static approach is much more conservative than the dynamic linear approach and the extent of areas exceeding the allowable value is much greater.

Table 3.2: Max von Mises stresses [MPa]. Equivalent static vs linear dynamic approach.

	Equivalent static	Linear dynamic
NISO	630.25	353.21

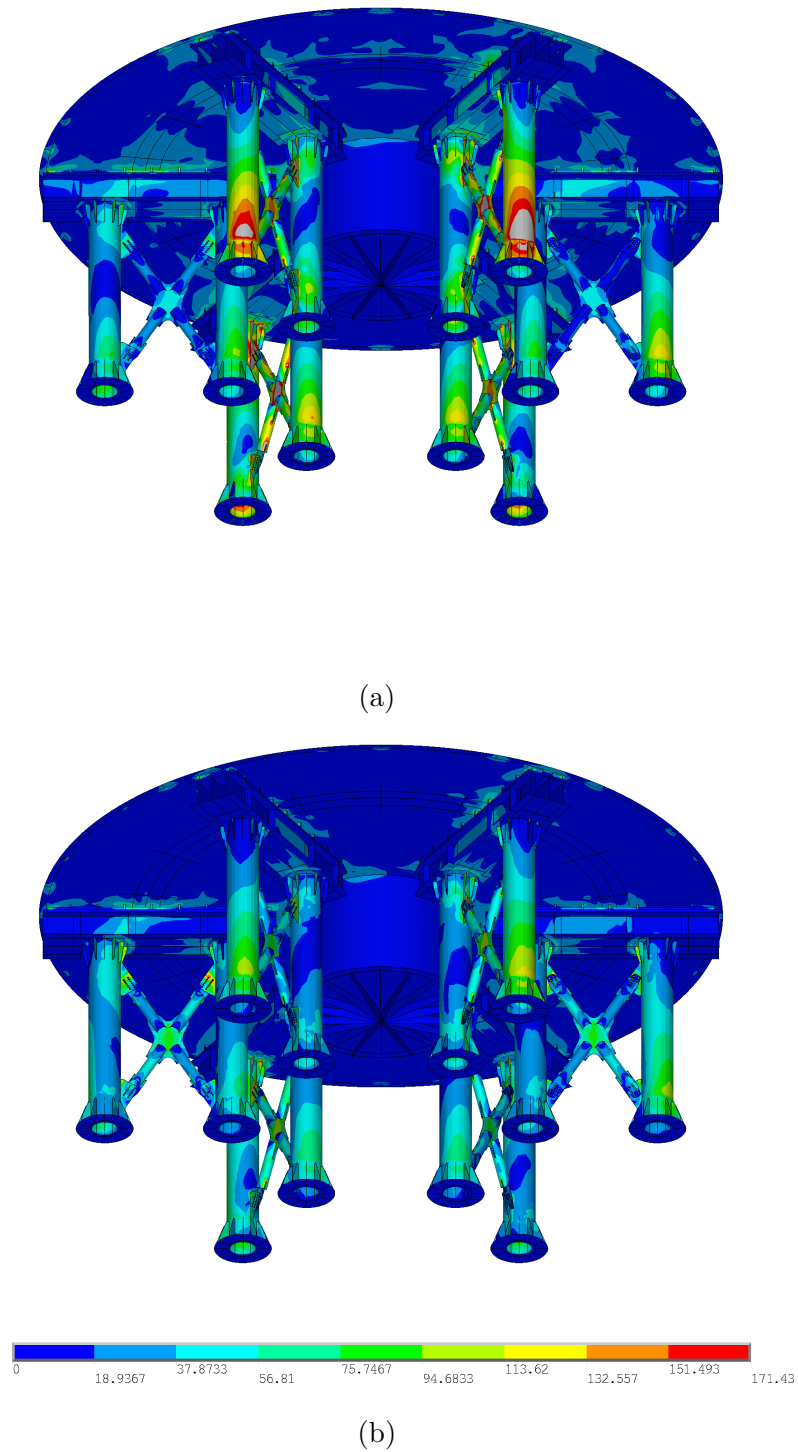


Figure 3.7: Contour maps of the von Mises stress [MPa]: results from equivalent static approach (a) and linear dynamic approach (b).

3.4.2 Not isolated vs isolated configuration

Table 3.3 and Section 3.4.2 show the results of the three configurations analyzed in terms of the system's horizontal eigenfrequencies ν with the highest mass participation factors (MPF) and the associated seismic accelerations a_s . The results for the two isolation cases are quite similar. The eigenfrequencies obtained from the numerical model deviate from the reference frequency of 0.4 Hz used in the analytical model by 8% (BISO) and 5% (TISO), respectively. This confirms the usefulness of the 1 DOF model for evaluating a first-attempt stiffness for the isolation system. Considering the effect of the seismic isolation in terms of maximum horizontal acceleration experienced by the system, this is significantly reduced compared to the non-isolated case, up to approximately 97.5% in the cases analyzed.

Table 3.3: Horizontal natural frequencies with the highest mass participation factor and seismic accelerations: x direction.

	ν_x [Hz]	MPF_x [%]	a_s^x [g]
NISO	5.85014	70.45	0.86865
BISO	0.36934	86.26	0.02162
TISO	0.37815	84.64	0.02291

Table 3.4: Horizontal natural frequencies with the highest mass participation factor and seismic accelerations: y direction.

	ν_y [Hz]	MPF_y [%]	a_s^y [g]
NISO	5.82766	69.93	0.87043
BISO	0.36913	86.27	0.02159
TISO	0.37853	84.52	0.02296

3.4.2.1 Equivalent stress field (von Mises)

Table 3.5 shows the maximum von Mises stress values at cryostat base in the three configurations, considering both the load combinations LC_1 and LC_2 (Section 3.2.2).

Table 3.5: Max von Mises stress [MPa]. Configurations comparison.

	LC_1	LC_2
NISO	351.40	353.21
BISO	19.15	16.03
TISO	14.75	14.24

The reduction of the stress state at the cryostat base is between 95% and 96% in the two isolated cases. Moreover, Figure 3.8 shows a detail of the cryostat base, with the areas exceeding the allowable stress value highlighted. In Figure 3.9 a comparison of the equivalent stress field in the three configurations analyzed is reported, considering the results of the LC_1 combination. As can also be seen from Figure 3.9a, the stress state in the non-isolated cryostat base is quite high. With the use of seismic isolation, the stress levels are dramatically reduced throughout the cryostat base. The BISO configuration, Figure 3.9b, shows higher stresses than the TISO one, Figure 3.9c. This is due to the isolation plane located exactly between the columns and the upper part of the machine, where the largest masses are positioned, which reduces the load transfer from the superstructure towards the pillars.

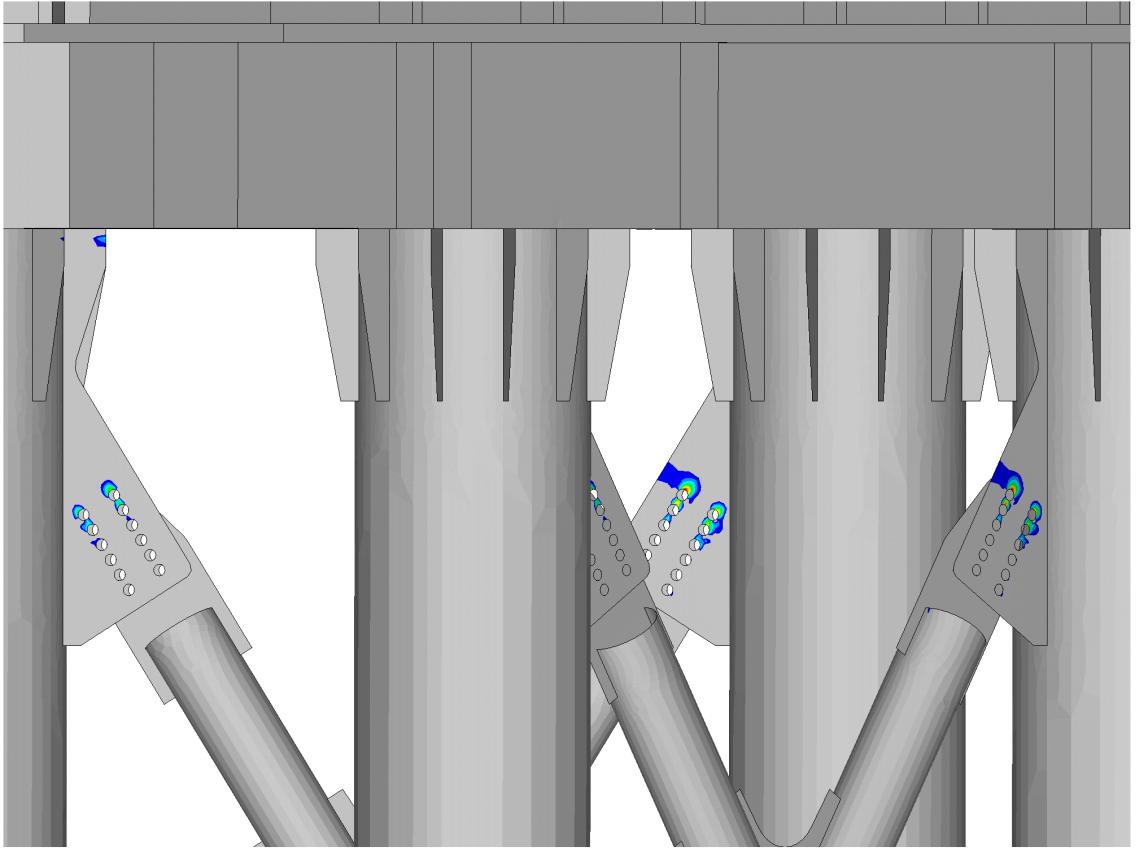
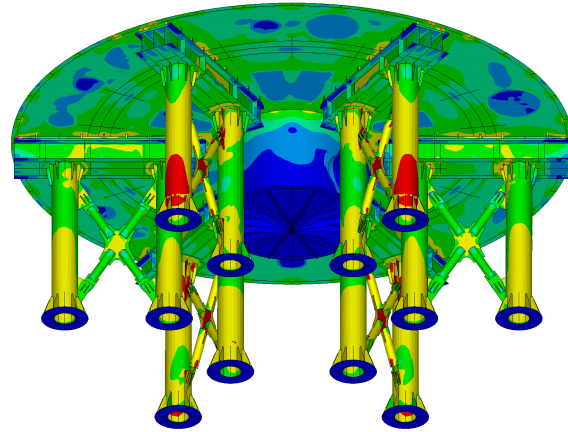
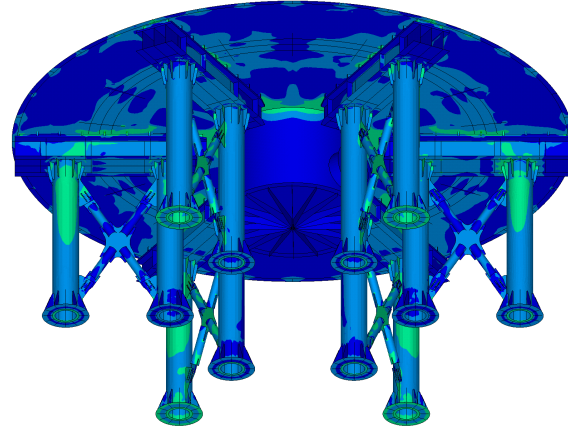


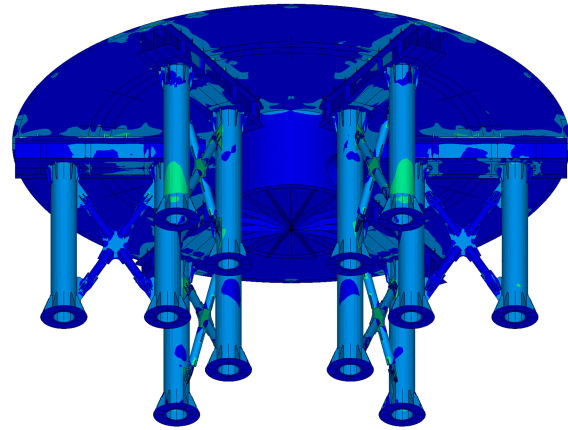
Figure 3.8: Areas exceeding the allowable stress considering the NISO configuration.



(a)



(b)



(c)



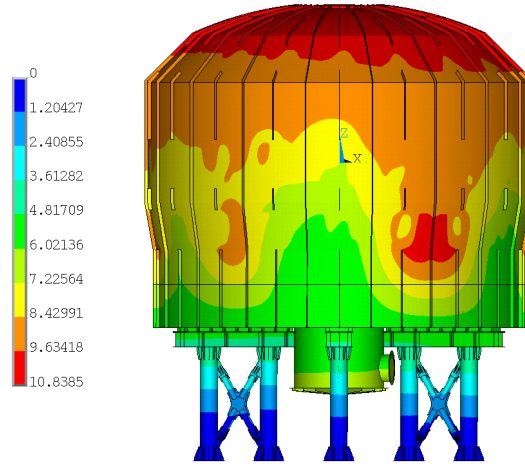
Figure 3.9: Contour maps of the von Mises stress [MPa]: NISO (a), BISO (b), TISO (c).

3.4.2.2 Displacement field

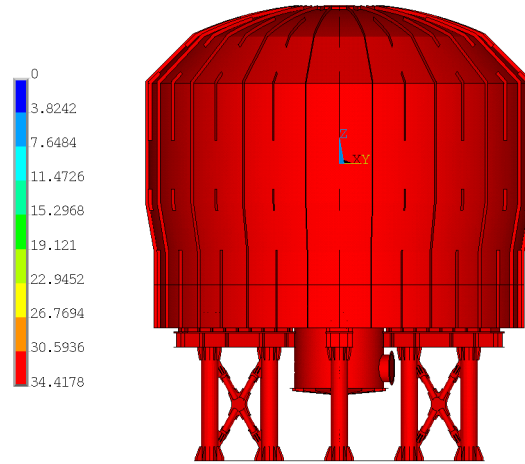
The maximum horizontal displacements are given in Table 3.6. As expected, the maximum horizontal displacements in the two isolated cases are much larger than in the non-isolated case, up to 3.5 times larger for the TISO configuration. The comparison between the three configurations in terms of horizontal displacement is showed in Figure 3.10. The effect of the isolation system can be clearly seen in Figure 3.10a and Figure 3.10b, which works by decoupling the displacements of the substructure and superstructure. The main difference is that in the BISO case the entire toroidal complex is seismically isolated and therefore all its components experience the same displacements. In the TISO case instead only the part above the pillars is isolated, resulting in displacements of the cryostat base comparable to those of the non-isolated configuration.

Table 3.6: Max horizontal displacement [mm].

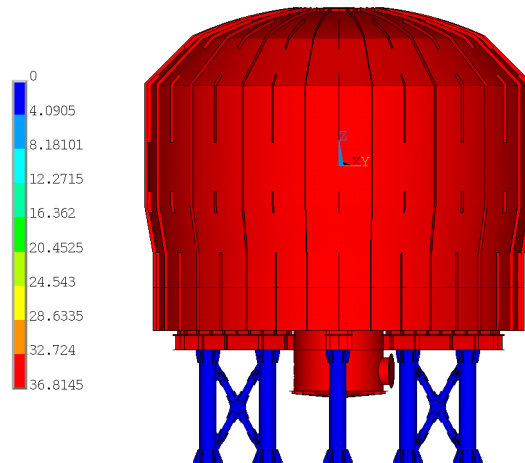
	LC_1	LC_2
NISO	10.54	10.64
BISO	34.42	34.51
TISO	36.81	37.02



(a)



(b)



(c)

Figure 3.10: Horizontal displacements [mm]: NISO (a), BISO (b), TISO (c).

3.4.2.3 1 DOF model vs FE model

A first estimation of the displacement experienced by the isolated system can be derived from the 1 DOF model, using the inverse formula from Equation (3.6). The comparison between the analytical and the numerical maximum horizontal displacements, u_{max} , is showed in Table 3.7.

Table 3.7: Horizontal displacement comparison: analytical vs numerical [mm].

	u_{max} [mm]	Δ [%]
Analytical	40.67	
BISO	34.42	-15.36
TISO	36.81	-9.47

3.5 Final remarks

In this chapter the behaviour of the toroidal complex of the DTT fusion machine in case of a seismic event has been investigated. This was carried out using both static and spectrum analysis based on the most severe local seismic response spectrum of the Frascati site. The analyzed spectrum is the local seismic response corresponding to the worst earthquake with return period of 1950 years expected at the site. This loading condition is coherent with the limit state for the prevention of collapse (SLC) which can be verified with faulted conditions as damage limit (e.g. criterion level D in accordance with ASME). Further verifications will be carried out associating the criterion level to the probability of several seismic events. Given the high stresses on the cryostat base under these conditions, two seismic isolation configurations were analyzed, the first (BISO) with the isolation system below the pillars and the second (TISO) with the isolation system above. Both systems showed significant benefits with respect to the non-isolated configuration, reducing the maximum stresses by an order of magnitude. Of the two isolated configurations, TISO is the one that guarantees lower stresses in the cryostat base, despite the difference is quite low. Taking into account the seismic isolation, a comparison was also made between the analytical results obtained using a simplified 1 DOF model and the numerical results in terms of horizontal displacements of the system. The numerical results did not differ significantly from the analytical results, up to a maximum value of 15% in the BISO configuration, making the 1 DOF model a valid option for first-attempt calculations. The presented work highlights some preliminary results without identifying an integrated solution implementing real seismic isolators. Further design and analysis activities will be performed

concerning the isolation devices, the control of undesirable movements, differential seismic ground motions, the control of displacements relative to surrounding ground and constructions.

Chapter 4

Thermo-structural analysis and structural integrity verification of the DTT vacuum vessel

The vacuum vessel (VV) is part of the plasma confinement system, acting as barrier among the plasma itself and the external environment, which also includes the superconducting magnets. Due to its crucial role in nuclear fusion devices, the vessel is subjected to a wide spectrum of different operating states featured by different types of loads like inertia loads, pressure loads, thermal loads and electromagnetic loads [40]. From a structural standpoint, one of the major concerns for the tokamaks vessels is the capability of withstanding the high loads arising from disruptive events, like Vertical Displacement Events (VDEs) [41]. Being such a phenomena time-dependent, the forces vary with time and the response of the structures is dynamic [42, 43]. It is still possible to utilise less time consuming static analysis in order to obtain information regarding peak stresses. The variability of the loads to which fusion machines are subjected is typically documented in dedicated technical documents, known as Load Specifications. These documents serve as a reference point for designers tasked with the dimensioning of such devices [44], [45]. In the context of DBA procedures, it is possible to rely on either a linear elastic approach or a non-linear elastic-plastic approach. However, an optimised design procedure [46] accounts for both of these approaches. Although the linear-elastic approach is more time-efficient, it is challenging to comprehend the degree of conservatism inherent in the method. Conversely, an elastic-plastic approach ensures the development of more robust design, albeit at the cost of elevated computational demands. In this chapter, the analysis and structural verifications carried out on the last vacuum vessel design of the DTT tokamak are presented. Preliminary assessments on a simplified model of the former DTT vacuum vessel have been already carried out to characterize the response of the vessel against loads from VDEs [47]. Such assessments have been pushed forward in [48], applying the novel

approach of coupling MAXFEA and ANSYS APDL described in [49] to derive EM loads. The same approach has been also used to carry on the VDE analysis on the last design update of DTT VV, which is described later in the chapter. Moreover, such a design has been tested from a thermo-mechanical point of view against load conditions derived from other operational states of the machine. For each analysed load condition structural integrity verifications were carried out in accordance with what is described in Chapter 2. As will be shown following in the chapter, it was necessary to introduce design changes to ensure the compliance with the rules under particularly severe loading conditions, namely when a VDE occurs. To evaluate the effectiveness of such changes the submodelling procedure was applied.

4.1 Methodology

When carrying out this type of analysis, temperature represents a fundamental parameter regarding its influence on the mechanical properties of materials and, consequently, its effect on the calculation of allowable limits during structural verification. Additionally, the presence of a thermal field affects the categorization of stresses when the linear elastic analysis method is pursued. It follows that the first fundamental step is to calculate the thermal field acting on the system. For some parts the temperature is imposed by the existing operating conditions, while for other components the thermal field is the result of the equilibrium established once the system has reached a steady state. To study the vacuum vessel, it was first necessary to characterise the thermal field of the ports and the gravity supports. Due to the heavy computational burden which would be necessary if such simulations were carried out on the global model, six local models, one for each port and one for the gravity support, were realized and analysed separately. This separation was possible considering that these systems are thermally independent of each other, all interfacing with the vacuum vessel and the cryostat, whose temperatures are set fixed depending on the operating conditions. The thermal simulation of such subsystems accounted for both conduction and radiation towards the Thermal Shield (TS). The heat flux along the axis of the ports and the GS was computed and then applied as a boundary condition for the thermal analysis of the global model. Temperature maps along the overall structure were calculated for each thermal condition provided in Table 4.1 and then applied for the subsequent structural analysis following the load-transfer procedure outlined in [50]. In the structural analysis also the mechanical boundaries were applied in terms of dead weight, pressures and electromagnetic loads as again reported in Table 4.1.

4.1.1 Structural integrity verification procedure

Given the temperature dependence of S_m and the fact that different parts of the system are at different temperatures, it is not immediately obvious to identify the most critical area from a structural integrity point of view. Therefore, an internal routine has been developed to calculate, for each finite element, the ratio between the equivalent stress and the allowable value given by the codes, taking into account the temperature of each finite element and thus the variability of the mechanical properties and the allowable stress value with the temperature itself. The total equivalent stress was used to identify the most stressed areas where stress linearization should be carried out, if necessary. In areas where the ratio between the total equivalent stress and the allowable value was greater than one, the total stress state was decomposed into its membrane and bending components according to what is detailed in Section 2.1 and compliance with the allowable limits specified in the regulations was verified according to the service level considered. In this regard results show a comparison of ASME and RCC-MRx.

4.1.2 Submodelling procedure

In order to evaluate the influence of design modifications while maintaining reasonable computational costs, a submodelling approach was employed. As the global model exhibits periodic symmetry, the various submodels under consideration consisted of a single sector. Accordingly, the selection of boundary nodes was a relatively straightforward process. The procedure entails the application of the displacements resulting from the global analysis to the boundary nodes of the submodel, followed by the application of the acting loads on the submodel itself.

4.2 Load specification

Table 4.1 illustrates the full range of load cases to be analysed in accordance with the load specification document for the DTT machine, with the pressure and temperature conditions that are characteristic of the state itself. In case of identical temperature range across the various load cases, the most unfavourable mechanical conditions were selected for analysis. Conversely, with regard to load cases characterised by the presence of EM loads resulting from plasma instability, the only available loads at the time of analysis were those of a Downward Vertical Displacement Event (DVDE) slow. Other sources of EM loads arising from plasma instabilities are Major Disruptions (MD), Upward Vertical Displacement Events (UVDE) and Toroidal Fast Discharge (TFD).

Table 4.1: Load specification for the thermo-structural design of the DTT machine.

ID	Load case	Pressure [MPa]			Temperature [°C]			
		Internal	External	Inter-shells				
L1	Construction	0.1	0.1	0.1	20	20	20	20
L2	Pressure test	0.1	0.1	0.72	20	20	20	20
L3	Pulse-off	0	0	0.4	110	Eq.	20	-193
L4	Baking	0	0	0.4	240	165	20	-193
L5a	Pulse-on	0	0	0.4	60	Eq	20	-193
L5b	MD fast	0	0	0.4	60	Eq	20	-193
L5c	MD slow	0	0	0.4	60	Eq	20	-193
L5d	DVDE fast	0	0	0.4	60	Eq	20	-193
L5e	DVDE slow	0	0	0.4	60	Eq	20	-193
L5f	UVDE fast	0	0	0.4	60	Eq	20	-193
L5g	UVDE slow	0	0	0.4	60	Eq	20	-193
L5h	TFD	0	0	0.4	60	Eq	20	-193
L6	Maintenance	0.1	0.1	0.4	240	Eq	20	20
L7a	Vacuum Vessel & Ports (VVP)	0.15	0	0.4	60	Eq	20	-193
	Loss Of Coolant accident (LOCA)							
L7b	Cryostat (CRS) Loss Of Vacuum Accident (LOVA)	0	0.15	0.4	60	Eq	20	-193
L8	Earthquake	0	0	0.4	60	Eq	20	-193
Vacuum Vessel (VV)					VV	Ports	CRS	THS

A temperature of 20°C was assumed as room temperature. ‘*Eq.*’ stands for equilibrium temperature resulting from the local analysis of the ports and gravity supports. Temperatures and pressures are referred to the fluid flowing in the specific circuit or environment:

- *internal*, refers to the in-vessel space containing the plasma during plasma operating state;
- *external*, refers to the ex-vessel environment faced to the thermal shield and the cryostat;
- *inter-shells*, refers to the volume included between the inner shell and the outer shell of the double walled structure.

4.2.1 Analysed load cases

In light of the aforementioned considerations and assumptions, the cases that have been subjected to analysis are summarized in Table 4.2 described below.

Table 4.2: Load cases analysed in this thesis.

ID	Load case	Pressure [MPa]			Temperature [°C]			
		Internal	External	Inter-shells				
L2	Pressure test	0.1	0.1	0.72	20	20	20	20
L4	Baking	0	0	0.4	240	165	20	-193
L5a	Normal plasma operation	0	0	0.4	60	Eq	20	-193
L5e	DVDE slow	0	0	0.4	60	Eq	20	-193
L6	Maintenance	0.1	0.1	0.4	240	Eq	20	20
L7a	Vacuum Vessel & Ports (VVP)	0.15	0	0.4	60	Eq	20	-193
L7b	Loss Of Coolant Accident (LOCA)	0	0.15	0.4	60	Eq	20	-193
L7b	Cryostat (CRS) Loss Of Vacuum Accident (LOVA)	0	0.15	0.4	60	Eq	20	-193
L8	Earthquake	0	0	0.4	60	Eq	20	-193
Vacuum Vessel (VV)					VV	Ports	CRS	THS

The analysed load cases are describe in the following list:

- **Pressure test**

The pressure test load case refers to the pneumatic pressure test foreseen as final assessment of the pressure equipment. The test will consist of 10 pressure cycles between 0.1 MPa and the required maximum value before carrying out helium leak testing as a propaedeutic measure for detecting major early crack propagation.

- **Baking**

In the case of baking the specified vessel temperature is reached circulating hot nitrogen whilst the port ducts are internally heated with dedicated electrical heaters. The thermal shield is actively cooled and the cryostat inner volume is vacuumed.

- **Normal plasma operation**

The tokamak is operating in pulse-on mode with plasma discharges. The vacuum vessel is thermally conditioned by the fluid flowing through the inner and outer shells. The thermal shield is actively cooled and the cryostat inner volume is vacuumed.

- **DVDE slow**

The three main sources of EM loading on passive structures during plasma perturbations are: poloidal and toroidal field variations (PFV and TFV), which mainly occur during thermal quenching (TQ) and current quenching (CQ), and Halo Currents (HC), which occurs during CQ. The eddy currents coming from the PFV are mainly in the toroidal direction and mostly interact with the poloidal EM field. On the other hand, the currents induced by TFV and HC are mainly in poloidal direction and mostly interact with toroidal EM field. These forces are in-plane forces. The mismatch between the poloidal components of the eddy and halo currents and the poloidal magnetic field could generate local out-of-plane forces within the passive structures, as well

as a net force. The VDE analysed is a downward slow VDE scenario. In order to analyse an even more critical case, a toroidal peak factor (TPF) in the HC distribution is considered. Specifically, the TPF was introduced by applying the following sinusoidal function:

$$F_{HC-TPF} = F_{HC}(1 - \cos\theta) \quad (4.1)$$

where $0 \leq \theta \leq \pi$ is the toroidal angle. As a consequence, in this case the EM halo forces are zero for $\theta = 0$ and assume twice the nominal value for $\theta = \pi$. In the absence of the TPF, one would typically assign a service level A in the case of a VDE. However, the introduction of the TPF has resulted in a reassessment of this level, which has been adjusted to a level C.

- **Maintenance**

Nitrogen is insufflated inside the cooling circuits of the water cooled components under maintenance in order to blow out the cooling water before cutting the cooling headers. The component is blown-out and drained at 20°C. The component is dried at the baking temperature, and this is the case analysed.

- **VVP LOCA**

The design of the ports must be verified considering the load transferred from interfaced systems (e.g. diagnostics and remote handling) and against possible buckling produced by gas pressure in the vacuum vessel possibly caused by a loss of coolant accident in the cooling circuits of the He cryogenic circuit or of the water cooling circuit.

- **CRS LOVA**

The design of the ports must be verified considering the load transferred from interfaced systems (e.g. diagnostics and remote handling) and against possible buckling produced by gas pressure in the cryostat in case of loss of vacuum accident.

- **Earthquake**

The seismic analysis of the vacuum chamber was conducted through an equivalent static approach, the details of which can be found in Section 3.2.1, Chapter 3. The spectrum used for the calculation of the equivalent static forces was the worst-case spectrum provided by the reference standard [26], and is represented in Figure 3.6.

4.3 FE analysis

To carry out the analysis campaign, a full 360° 3D FE model of the vacuum vessel and ports of the DTT VV was realized within the ANSYS suite. The utilization of the global model was essential for simulating the seismic event, which exhibited load components in the three Cartesian directions. The global model, the section of which is shown in Figure 4.1, has been simplified with respect to the real design and exhibits a periodic axial symmetry.

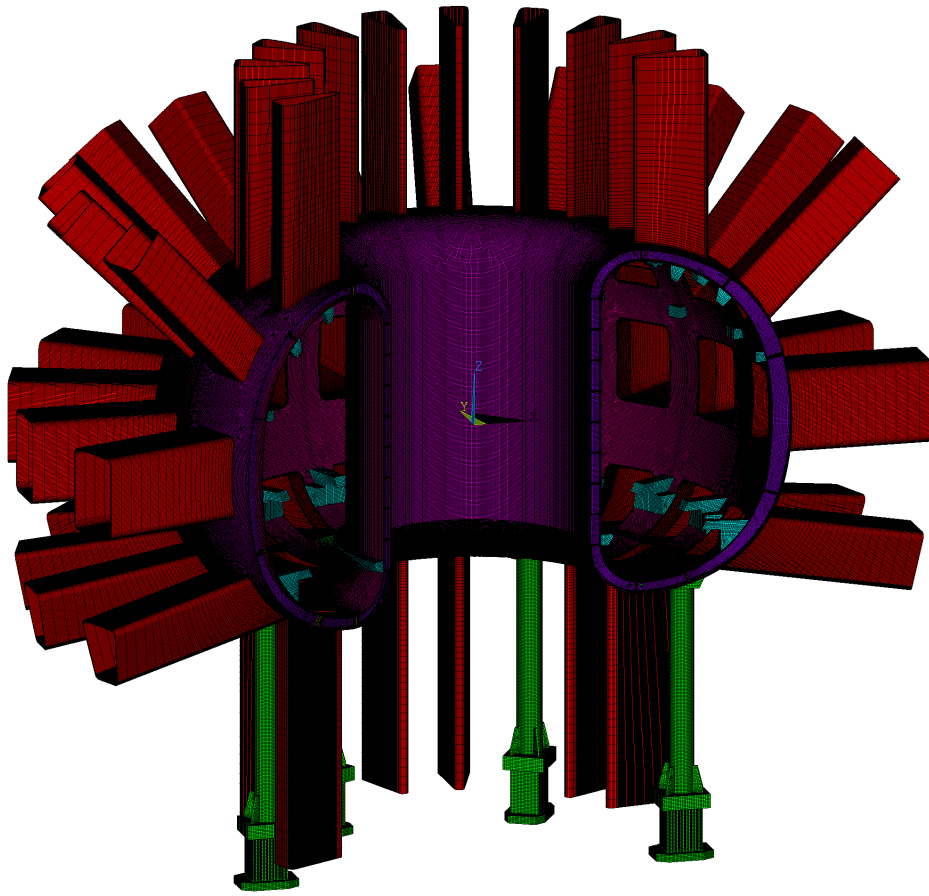


Figure 4.1: Global model of the DTT Vacuum Vessel and Ports.

The module consists of two adjacent half-sectors, the first of which features the gravity support and the second of which incorporates the port 5. The mesh of this module was executed in ANSYS Workbench using linear brick elements and is represented in Figure 4.2.

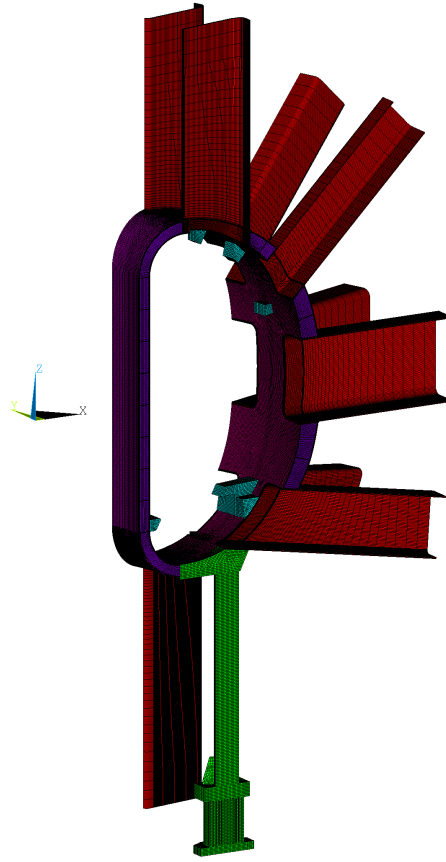


Figure 4.2: Basic module that has been used to build the 360° model.

Given that the model was employed for linear elastic analysis and, where necessary, the structural verification was undertaken through stress linearization, the model mesh was created in a way that ensured at least three elements in thickness. To create the 360°C model from the module, a customized routine was developed within ANSYS APDL. Each part of the system was modelled in detail, with all the fillets included, except for the bellows, whose contribution in terms of stiffness in the axial, poloidal and toroidal directions was considered using spring elements. The springs are fixed at one end and linked to a master node at the other, which in turn is connected to the ports by a set of rigid beam elements. For each port, a local coordinate system was created, oriented along the axis of the port itself, against which the spring elements were inserted. This enabled to catch the thermal expansion to occur within the limits imposed by the presence of the bellows. The in-vessel components are not included in the model so their contribution in terms of stiffness is neglected. The EM loads acting on the divertor and the stabilizing plates were considered through their resultants in terms of forces and moments applied to

the in-vessel components supports, which are included instead . The same global model was used both for global thermal analysis and thermo-mechanical analysis, exploiting the potential of the software, which easily allows, with the same type of elements, to change their degrees of freedom from temperatures to displacements and rotations.

Figure 4.3 illustrates the models of the ports and the gravity support, surrounded by the thermal shield, which have been employed to characterise the thermal field in cases where this was not imposed by the analysed case.

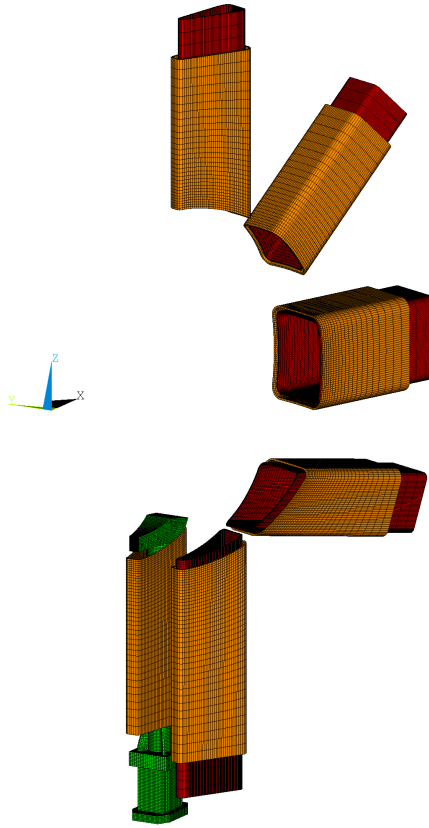


Figure 4.3: Local FE models of the ports and GS used for the thermal equilibrium analysis.

4.4 Results

4.4.1 Pressure test

In this load case, the effects of thermal expansion are not a factor, and thus only the primary stresses are reported. Given that the number of cycles is expected to

be 10, the pressure test is classified with a Service Level C.

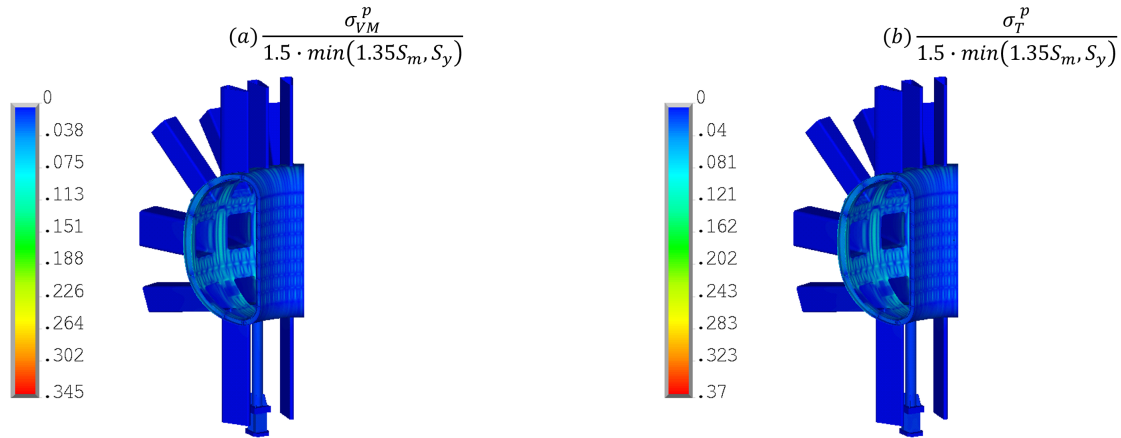


Figure 4.4: Pressure test load case. Ratio between total equivalent stress and membrane limit (service level A): primary (a) von Mises and (b) Tresca stress (according to RCC-MRx).

4.4.2 Baking

The baking load case has been classified as Service Level A. This load case is characterised by the highest temperature of those analysed. The effect of thermal expansion on the stress field is also clearly visible in Figure 4.5(b), where secondary stresses are part of the verification in the ASME code.

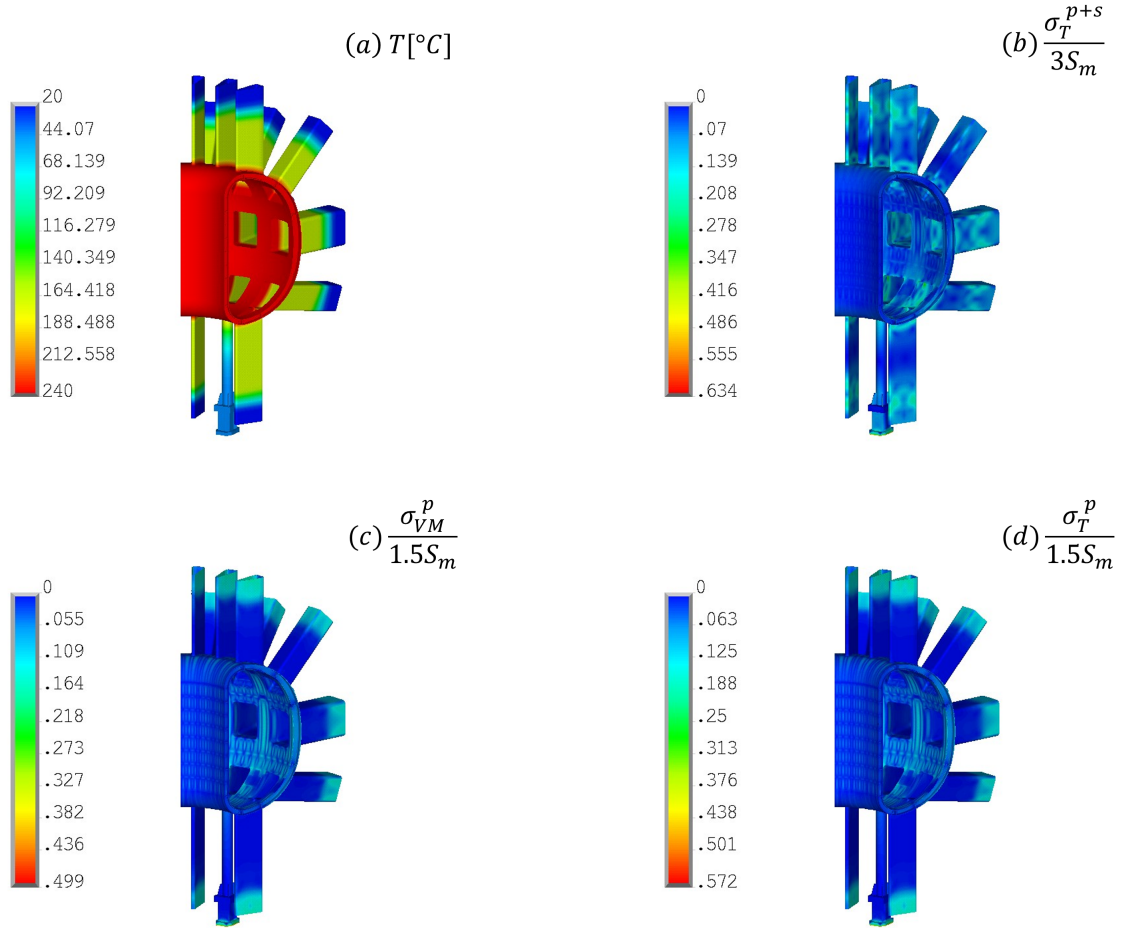


Figure 4.5: Baking load case. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level A): (b) primary plus secondary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.3 Pulse-on

The normal plasma operation has been classified as Service Level A. In this case the ASME code shows more conservatism than the RCC-MRx standard. Compared to the *Baking* load case the effects of thermal expansion are less prominent and the ASME allowable value is greater than the one of the RCC-MRx. In the end, the load case is verified regardless of the code.

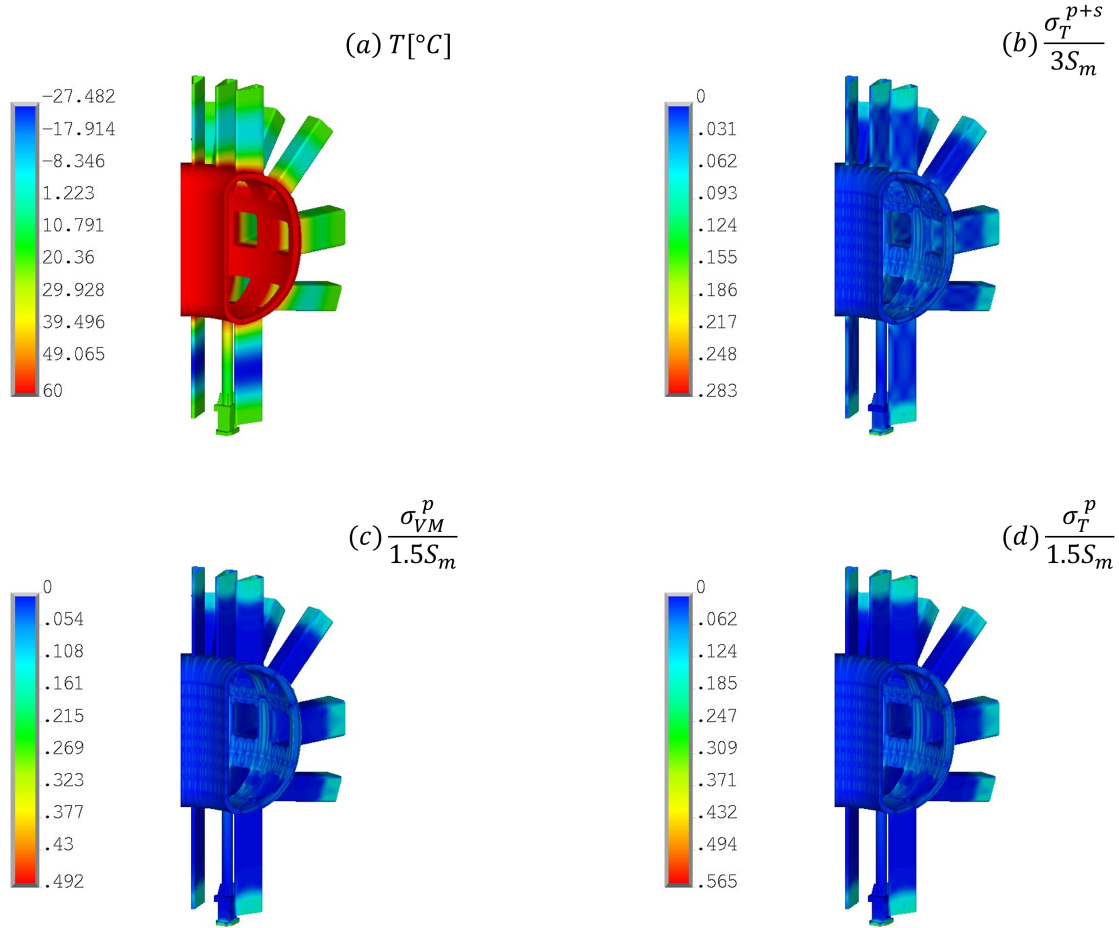


Figure 4.6: Normal plasma operation load case. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level A): (b) primary plus secondary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.4 Maintenance

The maintenance has been considered as Service Level C. In both ASME and RCC-MRx the secondary stresses are omitted. The load case is verified according to both the codes.

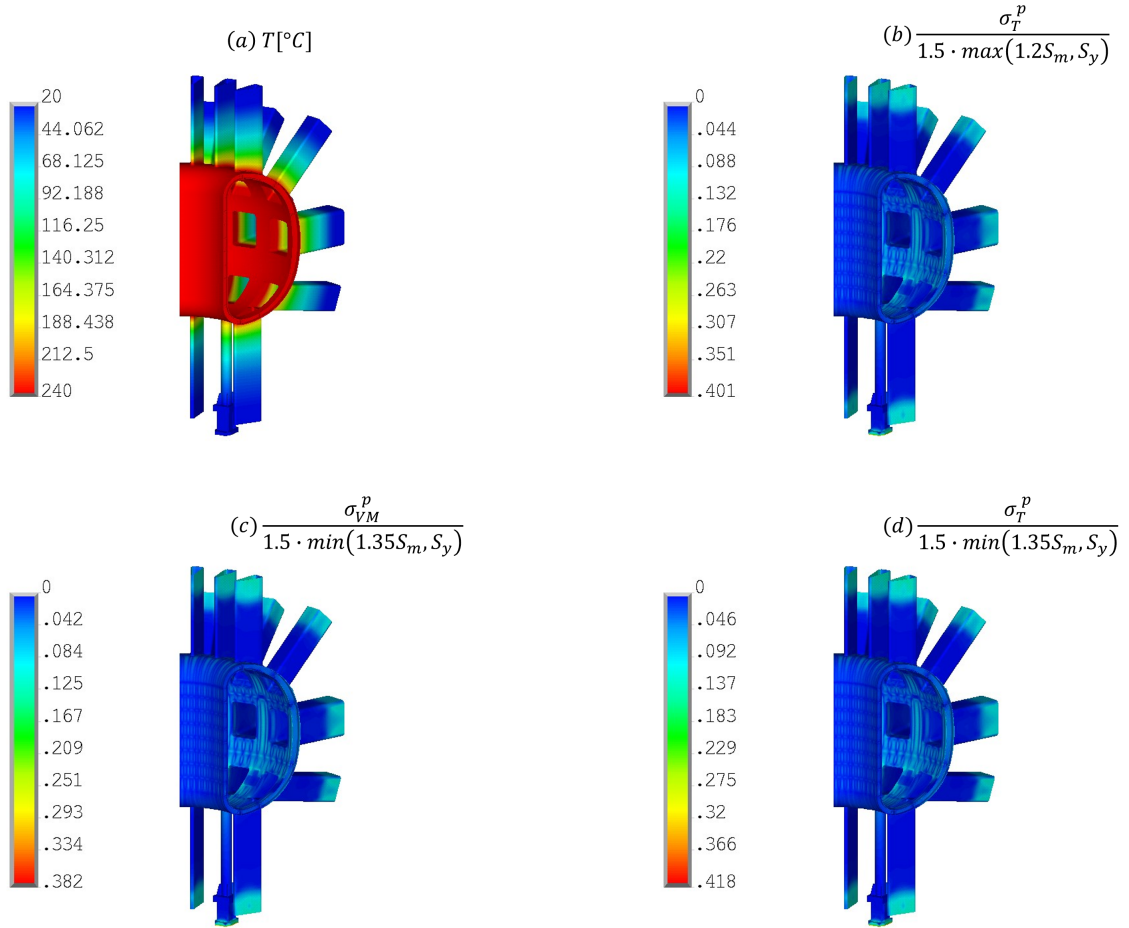


Figure 4.7: Maintenance. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level C): (b) primary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.5 VV LOCA

The vacuum vessel loss of coolant accident has been classified as Service Level A. The highest stresses arise on the ports, however no structural integrity criticalities appear.

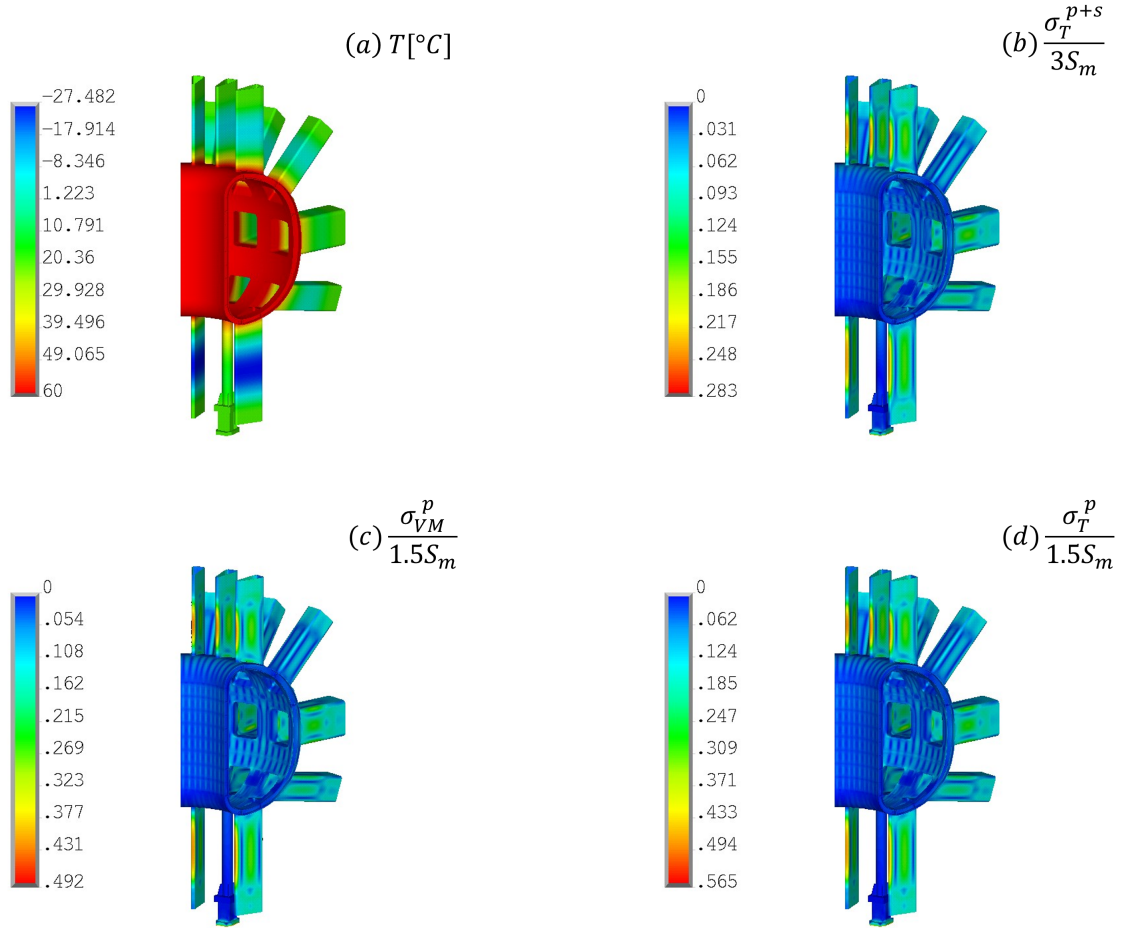


Figure 4.8: VV LOCA. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level A): (b) primary plus secondary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.6 CRS LOCA

In case of cryostat loss of coolant accident the situation is quite similar to the previous one. The level of service is again set as A and the design is safe against the considered loads.

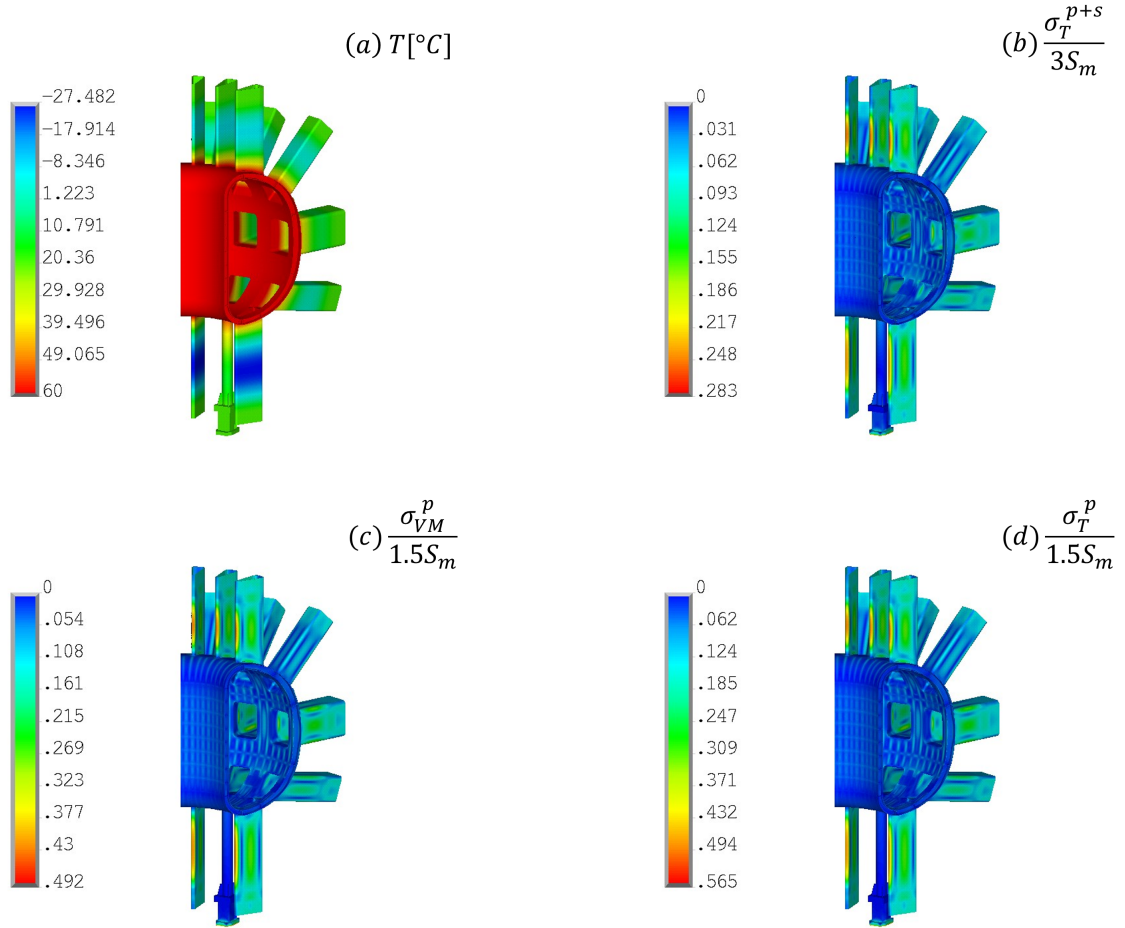


Figure 4.9: CRS LOCA. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level A): (b) primary plus secondary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.7 Earthquake

Given the severity of the event considered, with a return period of 1950 years (Section 3.1), the service level is set to C. In this load case, considering the Tresca formulation for the equivalent stress, both the codes gives the same level of safety.

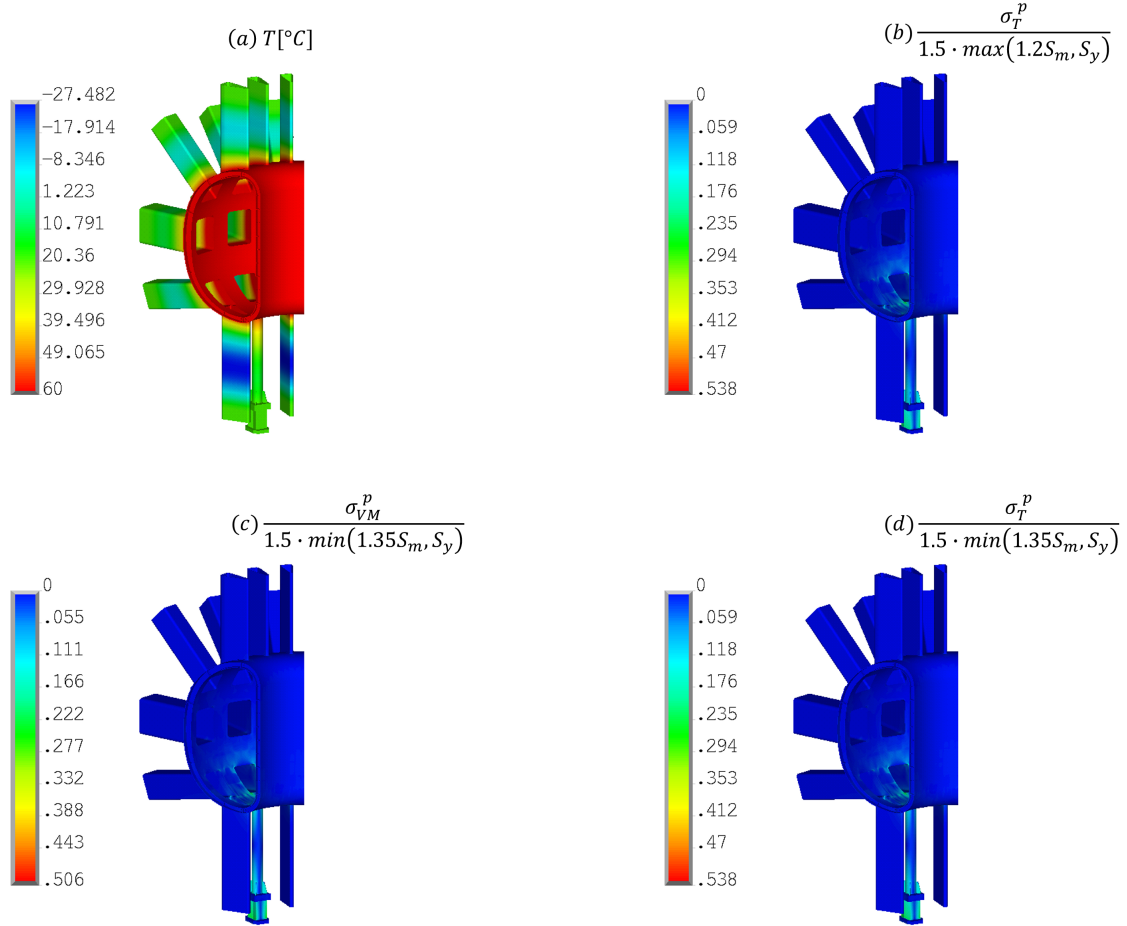


Figure 4.10: Earthquake. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level C): (b) primary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

4.4.8 VDE slow down

The DVDE is the worst load case analysed in terms of structural integrity. Figure 4.11 shows the status of the three sectors between 150° and 210° , which are the most stressed ones considering the application of a TPF to the electromagnetic loads.

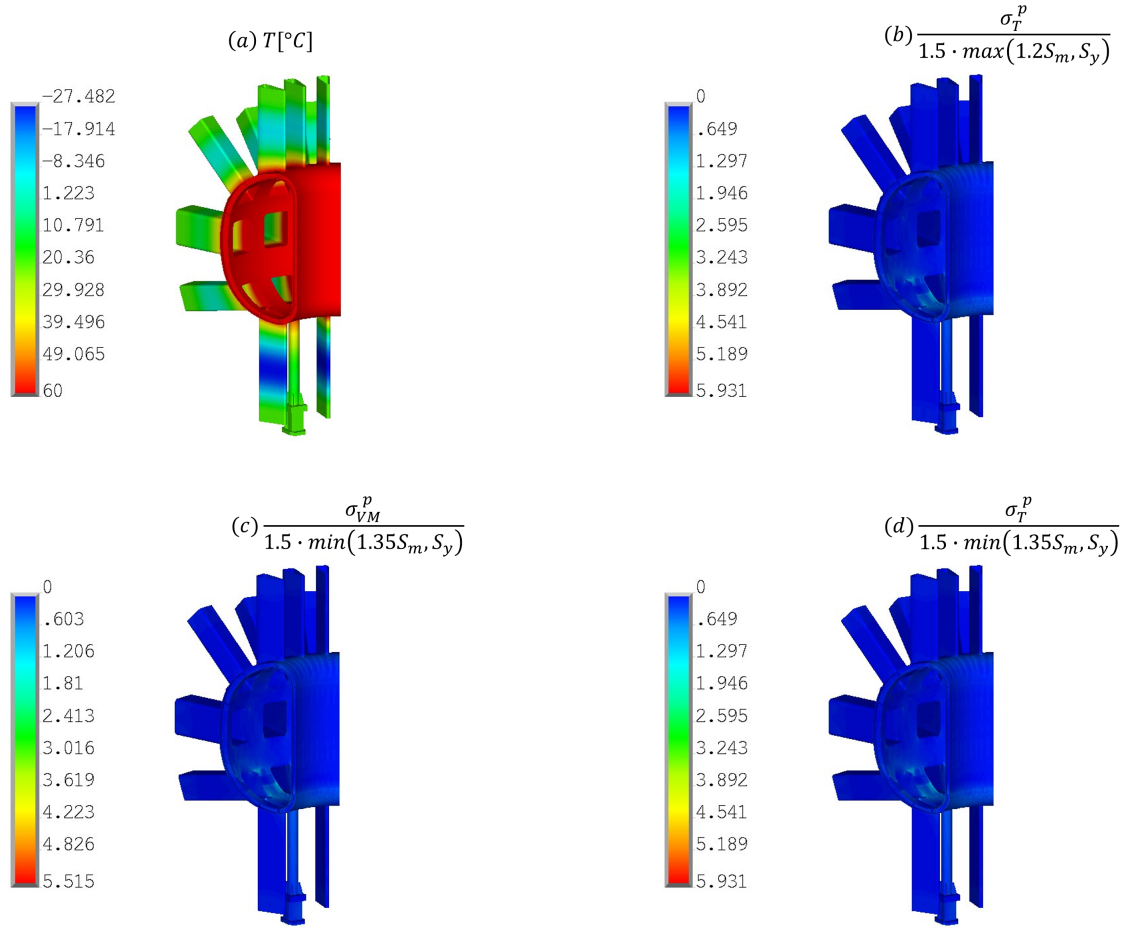


Figure 4.11: VDE. (a) Temperature field. Ratio between total equivalent stress and membrane limit (service level C): (b) primary Tresca stress (according to ASME), primary (c) von Mises and (d) Tresca stress (according to RCC-MRx).

In Figure 4.12 the areas exceeding the limits are shown for each part of the vessel subjected to higher stresses. Moreover the location of each through-thickness path used for the stress linearization is highlighted. In light of the fact that all of the selected paths are situated in close proximity to geometric discontinuities, verification was conducted in accordance with the criteria of local membrane and local membrane plus bending limits. The temperature of the interested areas is 60°C , thus the local membrane limit for the AISI 316LN, which is equal to the local membrane plus bending limit, is 363 MPa. In such a case, the limits calculated in accordance with the ASME and RCC-MRx equations are found to be equal. It can be concluded that the two standards are analogous if Tresca's formulation is assumed for the equivalent stress calculation for RCC-MRx.

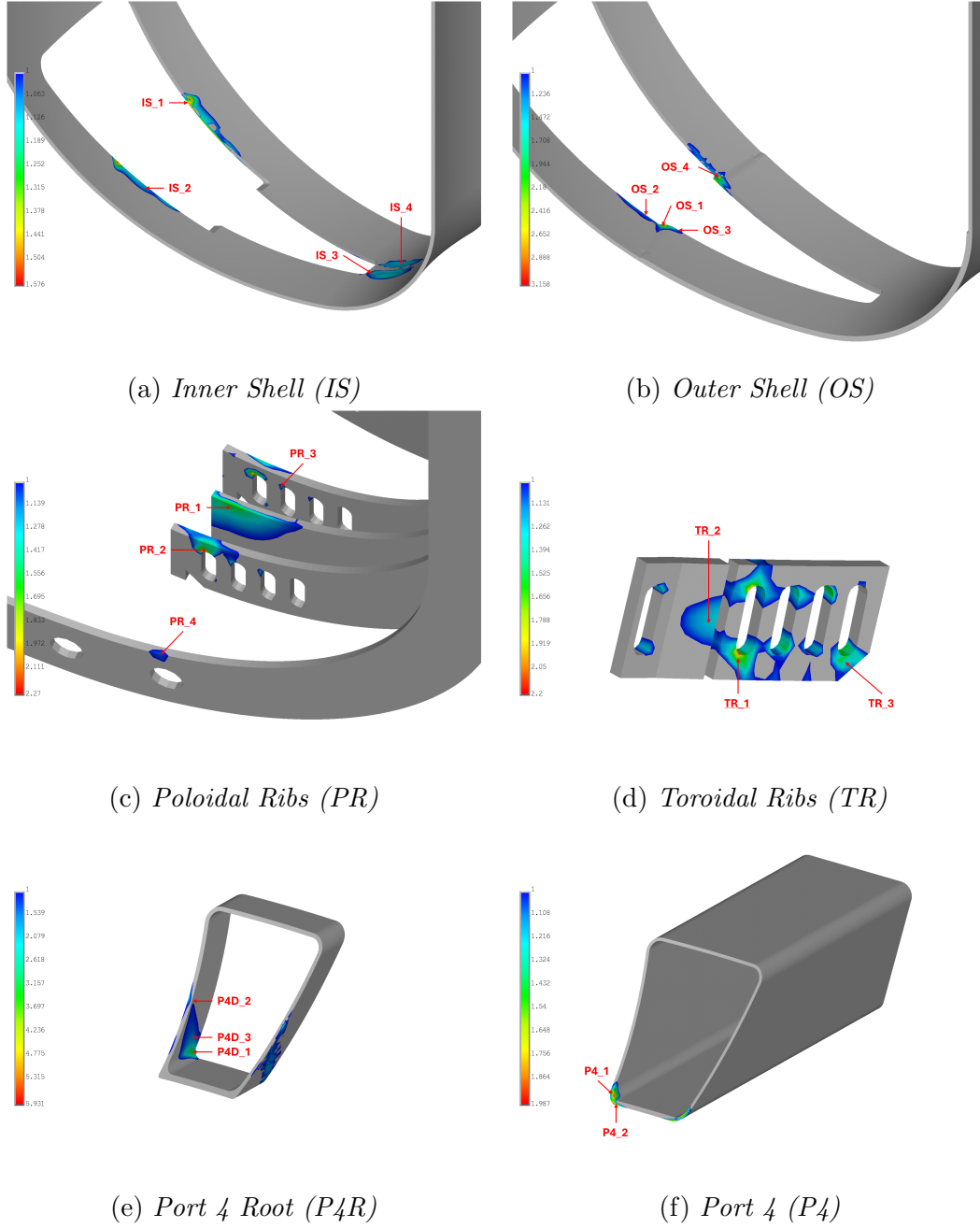


Figure 4.12: Areas exceeding the allowable limit and location of the through-thickness paths for the stress linearization procedure.

The following tables summarize the results of the stress linearization along the paths detailed in Figure 4.12. The structural integrity verification is faulted in the majority of the analysed areas for both the case of membrane stress and membrane plus bending stress. The most critical areas are located around the *Port 4 root*.

Table 4.3: Results of the stress linearization along the paths on the *Inner Shell*.

Inner Shell		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L+P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
IS_1	0.93	0.97
IS_2	1.00	1.39
IS_3	1.22	1.52
IS_4	1.05	1.23

Table 4.4: Results of the stress linearization along the paths on the *Outer Shell*.

Outer Shell		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L+P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
OS_1	1.51	2.50
OS_2	1.26	1.52
OS_3	0.89	1.93
OS_4	1.14	1.76

Table 4.5: Results of the stress linearization along the paths on the *Poloidal Ribs*.

Poloidal Ribs		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L+P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
PR_1	1.13	1.39
PR_2	1.36	2.16
PR_3	0.70	0.96
PR_4	1.02	1.06

Table 4.6: Results of the stress linearization along the paths on the *Toroidal Ribs*.

Toroidal Ribs		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L + P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
TR_1	1.18	2.35
TR_2	1.42	1.47
TR_3	1.26	1.32

Table 4.7: Results of the stress linearization along the paths on the *Port 4 Root*.

Port 4 Root		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L + P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
P4R_1	2.26	3.40
P4R_2	1.10	1.83
P4R_3	2.02	2.38

Table 4.8: Results of the stress linearization along the paths on the *Port 4*.

Port 4		
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	$\frac{\overline{P_L + P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$
P4_1	1.04	1.79
P4_2	0.95	1.08

As illustrated in Figure 4.13 the vertical displacements of the most heavily loaded portion of the vessel are depicted using a cutaway of the model between 130° and 180°. From the figure it is possible to deduce how the vacuum chamber deforms around the gravity support, which remains essentially undeformed due to its high stiffness. Although an increase in the number of GSs would have resulted in a better redistribution of the load and thus lower stresses on the vacuum vessel, this modification was not considered due to the excessive impact on the design. Local reinforcement of the vacuum vessel was therefore carried out, in particular to increase the cross section of the various shells.

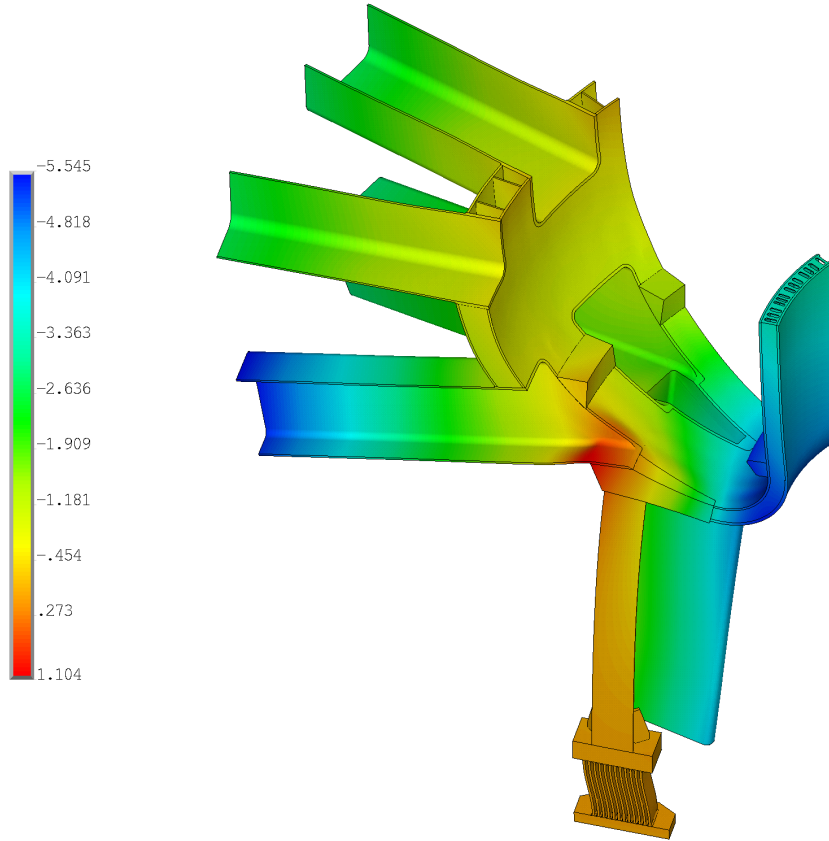


Figure 4.13: Vertical displacements [mm] of the most heavily loaded portion of the vessel between 130° and 180°.

4.4.8.1 Design changes

As seen in the previous section, the structural integrity of the system in the analysed case of VDE with TPF equal to two is not verified. The critical areas were therefore reinforced progressively and iteratively. The evaluation of the modifications made was carried out by means of several submodels of the analysed system, extracting the sector most affected by the TPF effect. A total of five submodels were created and analysed. The overall list of the design changes is collected below and shown in Figure 4.14:

- Inner shell thickening near port 4 and GS from 15mm to 25mm;
- Outer shell thickening near port 4 and GS from 15mm to 22mm;
- Port 4 root extension relative to the outer shell from 20mm to 60mm;
- Port 4 root thickening from 20mm to 30mm;
- Poloidal ribs thickening from 10mm to 15mm;
- Toroidal ribs thickening from 10mm to 15mm;

- Addition of toroidal ribs to support the root of port 4;
- Replacing the GS head previously integrated between the vessel shells with a flanged support holding the chamber;
- Addition of external poloidal ribs to the vessel connecting the GS head and the root of port 4.

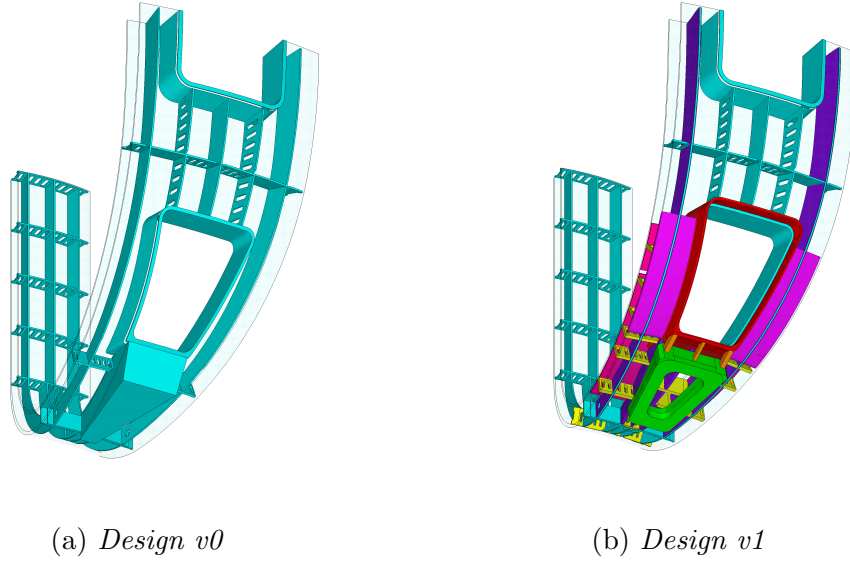


Figure 4.14: Original design(a) compared to the updated design(b).

The modifications carried out were effective in achieving the structural integrity of the vessel, as shown in the next tables which compare the results of the stress linearization before the design changes, namely *Design v0*, and after the implementation of the aforementioned modifications, namely *Design v1*.

Table 4.9: Comparison of the stress linearization on the *Inner Shell* before and after the design modifications.

	Inner Shell			
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$		$\frac{\overline{P_L + P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	
IS_1	0.93	0.36	0.97	0.43
IS_2	1.00	0.32	1.39	0.37
IS_3	1.22	0.39	1.52	0.45
IS_4	1.05	0.27	1.23	0.50
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

Table 4.10: Comparison of the stress linearization on the *Outer Shell* before and after the design modifications.

Outer Shell				
	$\overline{P_L}$ $1.5 \cdot \min(1.35S_m, S_y(T))$		$\overline{P_L+P_b}$ $1.5 \cdot \min(1.35S_m, S_y(T))$	
OS_1	1.51	0.46	2.50	0.58
OS_2	1.26	0.29	1.52	0.38
OS_3	0.89	0.78	1.93	0.82
OS_4	1.14	0.40	1.76	0.51
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

Table 4.11: Comparison of the stress linearization on the *Poloidal Ribs* before and after the design modifications.

Poloidal Ribs				
	$\overline{P_L}$ $1.5 \cdot \min(1.35S_m, S_y(T))$		$\overline{P_L+P_b}$ $1.5 \cdot \min(1.35S_m, S_y(T))$	
PR_1	1.13	0.46	1.39	0.47
PR_2	1.36	0.22	2.16	0.44
PR_3	0.70	0.18	0.96	0.19
PR_4	1.02	0.16	1.06	0.18
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

Table 4.12: Comparison of the stress linearization on the *Toroidal Ribs* before and after the design modifications.

Toroidal Ribs				
	$\overline{P_L}$ $1.5 \cdot \min(1.35S_m, S_y(T))$		$\overline{P_L+P_b}$ $1.5 \cdot \min(1.35S_m, S_y(T))$	
TR_1	1.18	0.93	2.35	0.96
TR_2	1.42	0.71	1.47	0.75
TR_3	1.26	0.63	1.32	0.70
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

Table 4.13: Comparison of the stress linearization on the *Port 4 Root* before and after the design modifications.

Port 4 Root				
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$		$\frac{\overline{P_L+P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	
P4D_1	2.26	0.51	3.40	0.70
P4D_2	1.10	0.48	1.83	0.78
P4D_3	2.02	0.29	2.38	0.37
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

Table 4.14: Comparison of the stress linearization on the *Port 4* before and after the design modifications.

Port 4				
	$\frac{\overline{P_L}}{1.5 \cdot \min(1.35S_m, S_y(T))}$		$\frac{\overline{P_L+P_b}}{1.5 \cdot \min(1.35S_m, S_y(T))}$	
P4_1	1.04	0.37	1.79	0.49
P4_2	0.95	0.33	1.08	0.44
	<i>Design v0</i>	<i>Design v1</i>	<i>Design v0</i>	<i>Design v1</i>

4.5 Final remarks

The results of the analysis conducted for different machine operating states revealed that the *Design v0* is effective in terms of its suitability for normal operation, and also for emergency situations as LOVA and LOCA, within the specified thermo-mechanical conditions. In contrast the design was found to be critical in the case of the DVDE slow with TPF equal to two. The proposed design modifications proved to be effective in ensuring the structural integrity of the vacuum chamber even under extremely severe conditions such as those of the VDE analysed. However, it is evident that such analysis cannot be sufficient to define the design as unequivocally verified. Subsequently, further verifications are required, particularly in regard to all load cases involving high electromagnetic loads on the vacuum vessel.

Chapter 5

Thermo-structural analyses and structural integrity verification of the DTT gravity support bolted-flanged joint

A well-dimensioned bolted joint is one that can ensure throughout its operational span that the flanges remain in contact with each other, thus maintaining the stiffness of the system within which the joint is inserted. In considering a flange joint subject to external loads, it can be observed that the action of a load that tends to separate the flanges represents the most critical condition with regard to bolt sizing. In such a situation, the bolting elements will experience greater tensile stresses due to the fact that the external load is added to the preload force. Conversely, external compressive loads facilitate the maintenance of contact between the flanges, thereby reducing the tensile stress experienced by the bolting elements themselves. This chapter presents the methodology employed for the sizing of the interface bolted joint between the vacuum vessel and the gravity support of the DTT machine. The primary objective is not the design itself, but rather the optimization of it by adopting a multicode approach considering the design constraints existing upstream. At the time of the analyses, no loads were available that would induce a separation between the flanges, which, as previously mentioned, is the most unfavorable situation for sizing bolting elements. Therefore, we proceeded with the use of available loads related to a slow DVDE by operating a sign change on the vertical component representing the sizing load. It is also crucial to note that the geometry of the joint had already been defined based on the spatial constraints introduced by the surrounding components. This limitation significantly reduced the designer's available range of options. The analysed bolted joint is a preloaded structural joint that does not have to comply with leak-tightness functions. The shape of the joint is trapezoidal and it is inclined with respect to the toroidal-radial

plane by 13° . The initial sizing of the system was conducted using a simplified analytical approach, which was subsequently complemented by a detailed sizing through a DBA approach. Additionally, the finite element analysis enabled the assessment of stresses at the interface between the bolt thread the flange threaded area, namely the internal thread. The detected stresses exceeded the allowable limit of the flange material. Thus the possibility to include threaded inserts of stronger materials has been investigated, with the objective of attaining a stress level at the flange internal threads that is consistent with the limit of the flange material. A number of combinations of materials were assessed in the course of the sizing process, with regard to both bolting elements and flanges. In particular, the materials under consideration for use in the bolting elements were ASTM A465-Grade 665 and Inconel 718. The mechanical properties of these materials, used for both the analytical and FE calculations, are set out in the table below.

Table 5.1: Mechanical properties of the materials considered for the bolting elements: ASTM A465-Grade 665 and Inconel 718

	Yield Strength (@RT)	Tensile Strength (@RT)
ASTM A435 - Grade 665	830	1170
Inconel 718	1035	1237

In consideration of the mechanical properties of the materials utilized, the bolting elements are designated as high-strength bolting elements. The scheme and nomenclature employed to refer to a flanged joint are illustrated in Figure 5.1.

The flanged joint that is the subject of this investigation employs screws whose characteristic geometric dimensions are illustrated in Figure 5.2.

5.1 Methodology

The sizing of the flanged joint was conducted in accordance with the SDC-IC code, which was developed on the basis of ASME Section III Subsection NG and RCC-MRx Appendix 6 references section 2.1.5. Instead, preliminary assessments of the diameter and number of bolting elements were made using relationships derived from Eurocode 3 [24]. The maximum diameter of the bolting elements for the application under consideration was calculated from the relation $L_m = 1.2d$. Given the length of the curvilinear abscissa of the flange and based on the minimum distance between two bolting elements, $X_m = 2.2d$, the bolting elements number has been evaluated. The quantities L_m and X_m are depicted in Figure 5.3.

A preliminary sizing and analysis of the joint was conducted analytically in accordance with the simplified procedure outlined in the SDC-IC, which is based on a uniaxial description of the problem. In the present case, the bolted joint is

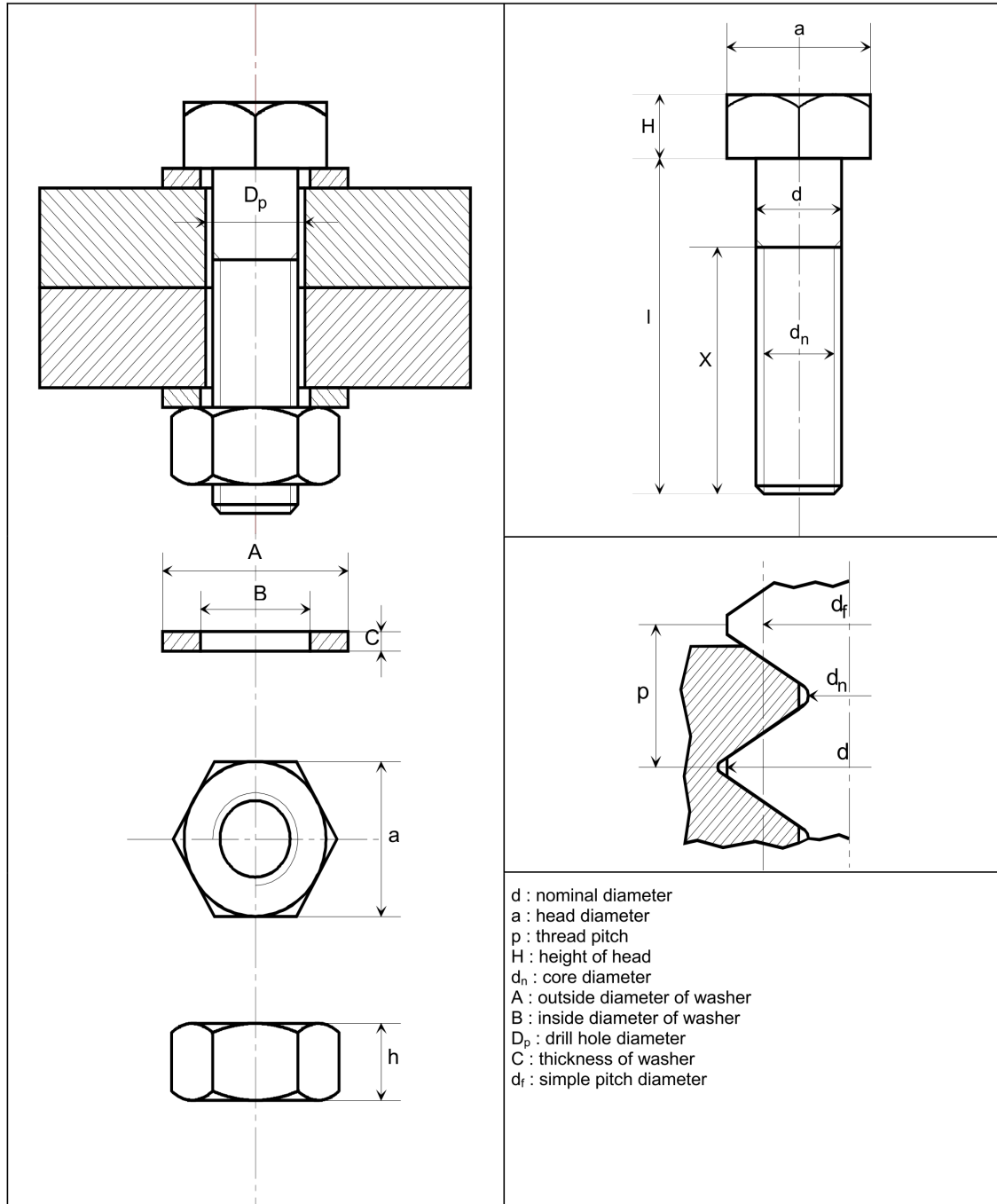


Figure 5.1: General scheme and nomenclature of a bolted flanged joint [23].

Screw ($l_{a3} = 0,8d$)

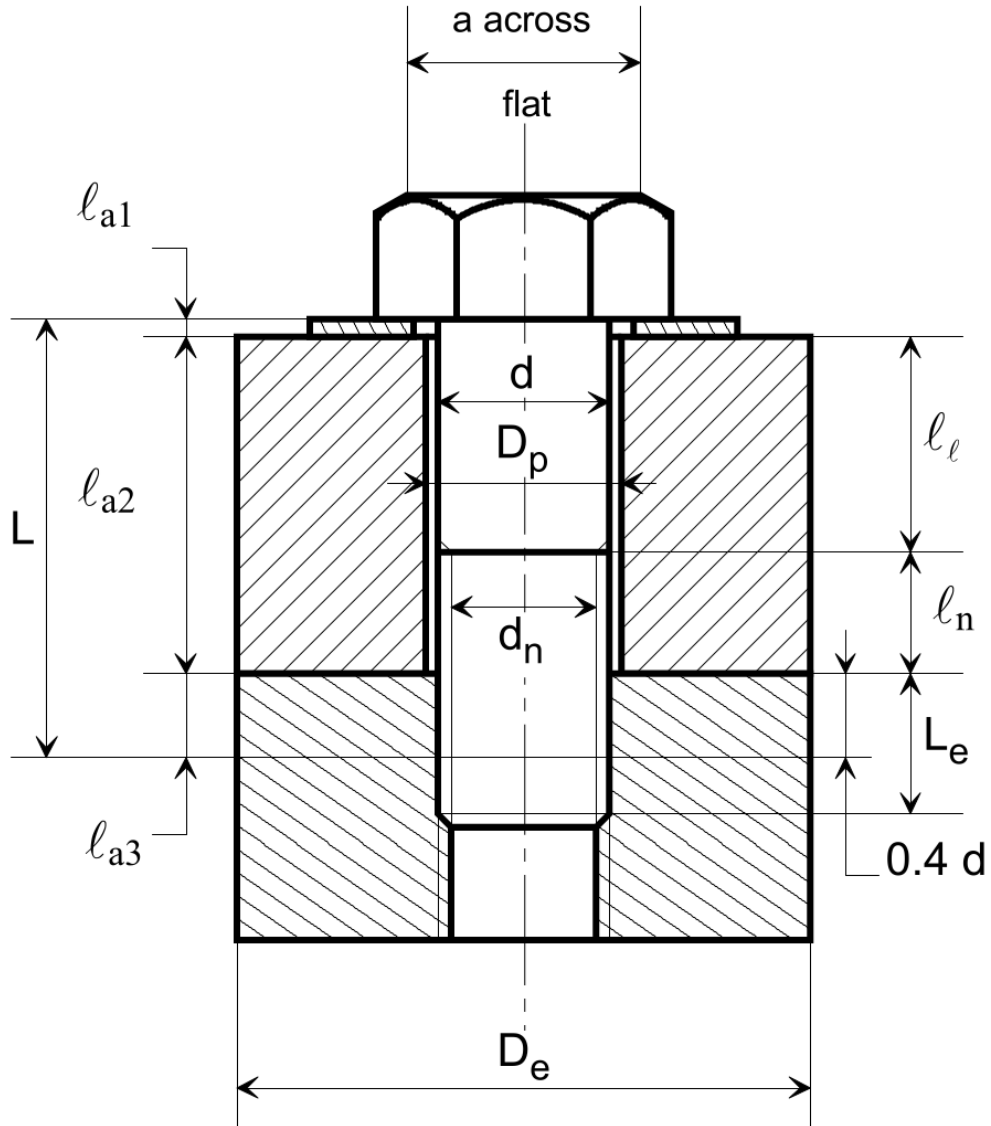


Figure 5.2: Main geometric parameters of a screw in a bolted flanged joint [23].

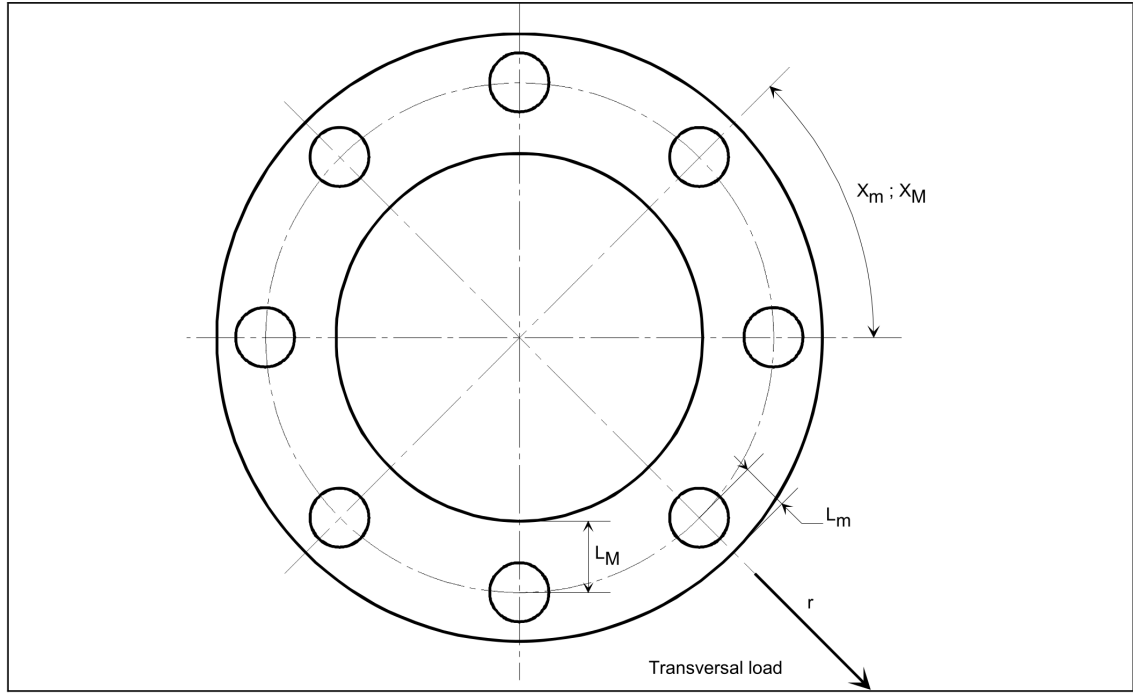


Figure 5.3: Geometric parameters used for the evaluation of bolting elements number and distribution within a bolted flanged joint [23].

represented by two springs in parallel: one with a stiffness equal to that of the assembled parts, and the other with a stiffness equal to that of the bolt. For this approach to be applicable, it is necessary to assume that there is no slip between the flanges, which is a critical aspect determining the efficacy of the design. Additionally, it is essential to assume that the effect of the bending moment on the joint is negligible, such that the bolting elements are subjected to uniform stress. To prevent slip between the flanges, the transverse component of the force must be less than a threshold value that is also a function of preload. This threshold value can be expressed through the following formula [24]:

$$F_{s,Rd} = \frac{k_s n \mu (F_{p,Cd} - F_{t,Ed})}{\gamma_{M3}} \quad (5.1)$$

where,

k_s depends on the bolt holes, in this case is equal to 1,

n is the number of the friction planes,

μ is the slip factor,

$F_{p,Cd}$ is the preload,

$F_{t,Ed}$ is the applied tensile force,

γ_{M3} is a correction factor equal to 1.25.

The selection of the appropriate preload is not a straightforward process and

could be iterative. Nonetheless the Eurocode 3 provides a formula for calculating the design value for preload, which is given by the following expression:

$$F_{p,Cd} = \frac{0.7f_{ub}A_s}{\gamma_{M7}} \quad (5.2)$$

where,

f_{ub} is the ultimate tensile stress of the bolt,

A_s is the tensile area, which is a function of bolt diameter and thread pitch,

γ_{M7} is a correction factor equal to 1,1.

As can be seen from the formula, the value of the preload is a function of not only the size of the bolt but also the material of the bolt through the ultimate tensile stress. Once it was established that the two flanges do not slide between them and that the effect of external moments on the joint is negligible, the analytical procedure was applied. This procedure is comprised of the following steps:

1. Calculation of rigidity of the bolting elements and the assembled parts;
2. Effect of preload;
3. Effect of different thermal expansion;
4. Effects of an external load applied to the bolted assembly;
5. Definition of stresses in the bolt;
6. Elastic analysis verification.

The relationships associated with individual points 1 through 4 can be found within the standard and are not provided here to avoid undue burden on the discussion. The following relationships from Appendix A6 of RCC-MRx are relevant to the definition of stresses on bolting elements. The stresses can be divided as follows:

Stress induced by axial load

- Mean tensile stress on the thread root section

$$\sigma_n^N = \frac{4N}{\pi d_n^2} \quad (5.3)$$

- Mean tensile stress on the smooth shank section

$$\sigma_I^N = \frac{4N}{\pi d_I^2} \quad (5.4)$$

- Shear stress on threads

$$\tau_f^N = \frac{2N}{\pi d_f L_e^I} \quad (5.5)$$

- Shear stress on the screw head

$$\tau_t^N = \frac{N}{\pi d_1 H} \quad (5.6)$$

- Contact pressure under the head

$$p_t^N = \frac{4N}{\pi(a^2 - B^2)} \quad (5.7)$$

where B is the inner diameter of the washer.

Stress induced by a residual twisting torque

- Thread root shear stress

$$\tau_n^{C_r} = \frac{16C_r}{\pi d_n^3} \quad (5.8)$$

- Smooth shank shear stress

$$\tau_I^{C_r} = \frac{16C_r}{\pi d_I^3} \quad (5.9)$$

- Head shear stress

$$\tau_t^{C_r} = \frac{16C_t}{\pi d_I^3} \quad (5.10)$$

Stress induced by the application of a transverse load

- Core shear stress

$$\tau_n^T = \frac{4T}{\pi d_n^3} \quad (5.11)$$

- Smooth shank shear stress

$$\tau_n^T = \frac{4T}{\pi d_I^3} \quad (5.12)$$

The verification process is conducted through a comparison of the mean and maximum equivalent stresses, as expressed through the relations detailed in section 5.1 and section 5.1, with the allowable values.

Mean stress intensity

- Core section

$$(\overline{\sigma_m})_n = \sqrt{(\sigma_n^N)^2 + 3(\tau_n^T)^2} \quad (5.13)$$

- Smooth shank section

$$(\overline{\sigma_m})_I = \sqrt{(\sigma_I^N)^2 + 3(\tau_I^T)^2} \quad (5.14)$$

Maximum stress intensity

- Core section

$$(\overline{\sigma_m + \sigma_b})_n = \sqrt{(\sigma_n^N)^2 + 3(\tau_n^{Cr} + \tau_n^T)^2} \quad (5.15)$$

- Smooth shank section

$$(\overline{\sigma_m + \sigma_b})_I = \sqrt{(\sigma_I^N)^2 + 3(\tau_I^{Cr} + \tau_I^T)^2} \quad (5.16)$$

In the event that the aforementioned verifications are not met, while maintaining the analysed configuration's diameter and number of bolting elements constant, one potential solution is to reduce the stress on the bolting elements by reducing the shear component resulting from the external load. This can be achieved by incorporating elements, such as shear keys, to support the external transverse load into the assembly. The joint analysed in this chapter has four keys. Once the configuration that had been subjected to analytical verification was identified, a finite element calculation was performed, first without considering the external acting moments, then with the moments included. A preliminary comparison was conducted between the results obtained through the analytical approach and those of the finite element analysis without the external moment. Subsequent to this comparison, the effect of the external moments was incorporated, thereby enabling a comprehensive examination of the joint's overall behaviour, including that of the flanges, within the specified load case. The verification of the screws, based on numerical results, was performed using stress linearization, which is based on a surface integral formulation. This formulation is described in detail in Section 2.2. The identification of the most stressed bolt was followed by the extraction of the stress tensors related to the section under consideration. The subsequent derivation of the average and maximum equivalent tensors was then possible. Given the potential complexity of interpolating the stress fields on the section under verification, first a triangular discretization of the domain was performed through Delaunay triangulation [51]. In order to facilitate the calculation of the integrals in Equation (2.30) and Equation (2.31), the mean value of the stress has been considered for each triangle. The resulting equations are as follows:

$$\sigma_{ij}^m = \frac{1}{A} \sum_k (\overline{\sigma_{ij}^k} \int_{A_k} dA_k) \quad (5.17)$$

$$\sigma_{ij}^b = \frac{x_3}{I} \sum_k (\overline{\sigma_{ij}^k} \int_{A_k} x_3^k dA_k) \quad (5.18)$$

Finally, an investigation was conducted into the stresses exerted on the AISI 316LN flange internal screw, with the objective of evaluating the potential incorporation of threaded inserts to achieve stress levels on the flange compatible with the material limits. A radial path has been created starting from the interface between the flange and the screw, along the flange itself. The trend of the equivalent von Mises stress on this path has been compared with the yielding strength of the material in order to size the threaded inserts.

5.2 Load specification

The most loaded gravity support has been analysed. The loads considered for sizing were those resulting from the combination of an upward VDE and seismic loads which refers to the Service Level C. The components in terms of forces and moments, which had already been projected onto the flange interface plane, are given in the table below.

Table 5.2: Forces and moments on the most loaded GS resulting from the combination of an UVDE and a seismic event.

		Radial	Toroidal	Axial
Force	[kN]	542	165	2820
Moment	[kNm]	178	-241	-196

In light of the joint's inclination, the shear component can be calculated as the sum of the external shear force and the projected component of the normal force along the flange plane.

5.3 FE analysis

The finite element model used for the calculations is shown in Figure 5.4. Linear solid elements, designated as SOLID 187, were utilized for the modelling process, which was conducted within the ANSYS Workbench environment. Bonded contact elements were used in the contact area between the threaded part of the screw and the nut, while frictional contact modelling was used on the contact planes between the bolt head and washer and between the washer and flange. Frictional modelling was also employed at the interface between the two flanges, such as on the contact surfaces between flanges and keys. Preload is applied through bolt tensioning by a specific feature inside the software. In comparison with the analytical

approach, in the FE analysis the contribution related to residual torque cannot be taken into account. To do this would have required detailed modelling of the screw and nut threads, and a frictional contact between them. The application of loads was executed through the implementation of a remote point that was connected to the centroid of the radial toroidal cross-section of the flange. The area modelling the flange nut was discretized through a structured mesh in order to facilitate the extraction of the stress trend along a radial path to compare it with the yielding stress of the flange material.

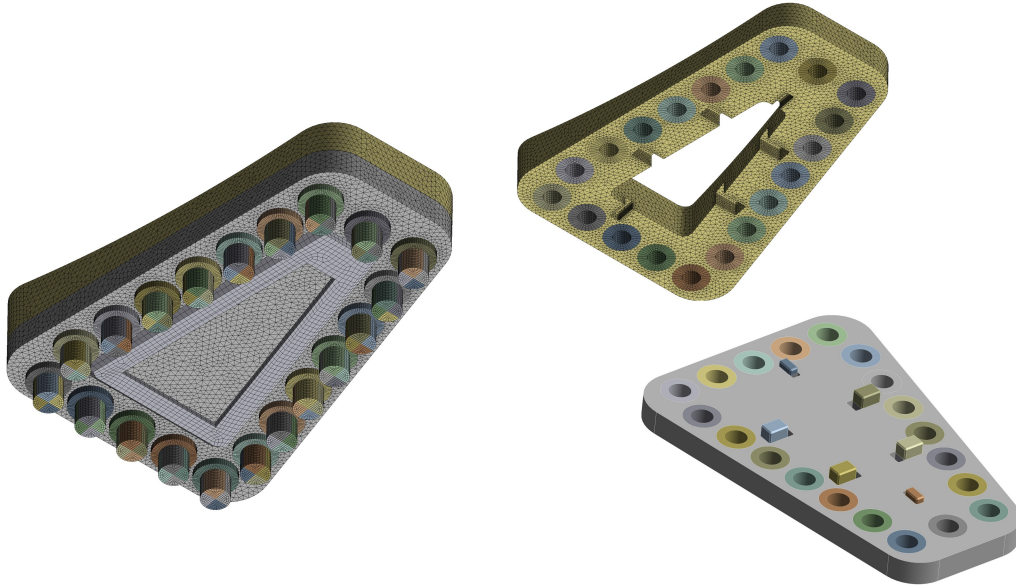


Figure 5.4: FE element model of the bolted joint used in the numerical calculation.

5.4 Results

In accordance with the formulas presented in Section 5.1, the maximum diameter of bolting elements that can be accommodated in the flange is 30mm. This value was then used to estimate the maximum number of bolting elements, which is 20. The values of the preload and slip resistance limit for both materials were estimated respectively using the formulas Equation (5.2) and Equation (5.1). The results are collected in Table 5.3.

Table 5.3: Preload, design slip resistance and transversal force per bolting element.

	ASTM A453-Grade 665	Inconel 718
Preload	394	492
Design slip resistance [kN]	45.1	60.7
Transversal force	28.3	

A comparison of the preload values for the two materials with the design slip resistance reveals that the preload is approximately one order of magnitude higher than the slip force value. Therefore, the preload has been established considering a target value for the slip resistance of 5% higher than the external transverse force. The new value of the preload and the design slip resistance, which are now equal for both the materials, are:

Table 5.4: Tuned preload based on the external transversal force.

		Both materials
Preload		298.4
Design slip resistance	[kN]	29.7
Transversal force		28.3

5.4.1 Analytical procedure

Following the characterisation of the bolted joint in terms of screw diameter and number and after the evaluation of the preload necessary to avoid slippage between the flanges, the analytical procedure has been applied. The joint sizing was conducted with consideration of the system at both the reference temperature, with the objective of evaluating the effects of mechanical loads alone, and at the operating temperature, in order to include the effects of thermal expansion as well. Accordingly, structural integrity verifications of the joint were conducted by employing allowable values of S_{mB} for the two materials under consideration, as specified in Table 5.5.

Table 5.5: Allowable stress for bolting elements, S_{mB} , for both the materials considered.

		ASTM A453-Grade 665	Inconel 718
S_{mB}	[kN]	370.5	412.2

The structural integrity verifications for the mean and maximum stresses have been conducted on the core section, which is the least resistant area and thus experiences higher stresses with respect to the smooth shank section. As a consequence of thermal expansion, the preload value diminishes. Consequently, the most critical scenario from a structural standpoint is represented by the case of only mechanical loads. The results of the verifications conducted on the two materials in question are presented in the table below.

Table 5.6: Structural integrity verification of the screw core section in terms of mean and maximum stresses from analytical procedure.

	Mechanical loads on the screw core section	
	Mean stress	Max stress
	$\frac{(\overline{\sigma_m})_n}{1.5S_{mB}}$	$\frac{(\overline{\sigma_m+\sigma_b})_n}{2.25S_{mB}}$
ASTM A453-Grade 665	1.05	0.73
Inconel 718	0.94	0.80

As evidenced by the tables, only the screws made by Inconel 718 are verified through the analytical procedure.

5.4.2 FE analysis

5.4.2.1 Displacements

The joint displacements, in the case without including the external moments, are displayed in Figure 5.5. As can be seen from the figure the deformation of the flange is localized in the centre. This deformation leads to the uneven elongation of the bolting elements, which are also under moment-loading conditions despite the absence of an external moment in this particular case. This aspect alone draws attention to the limitations of the analytical approach in the case of complex geometries such as the one analysed.

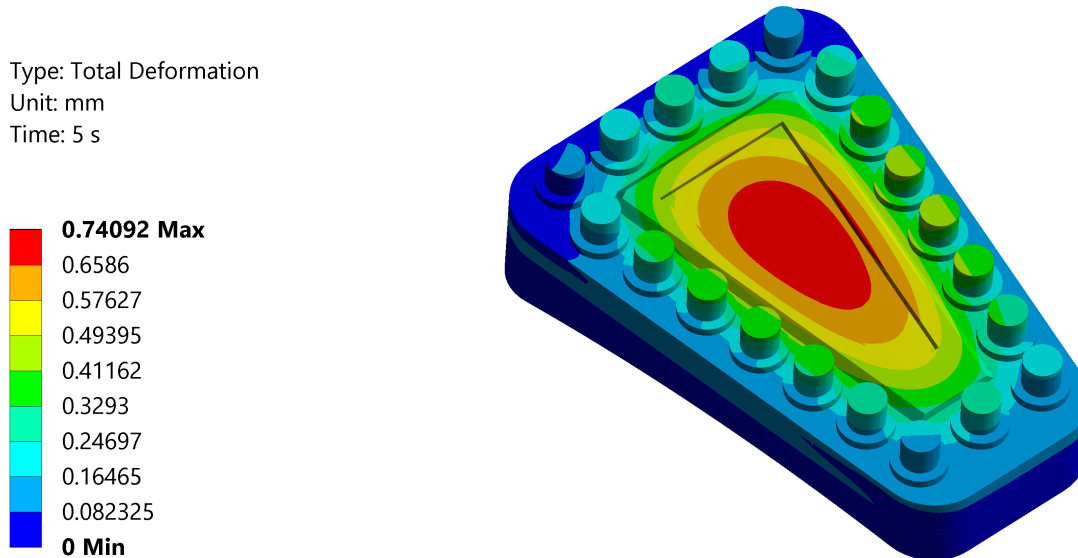


Figure 5.5: Total displacement [mm] without considering the contribution of external moments.

The total displacement considering the load case with the external moments

included are shown in Figure 5.6.

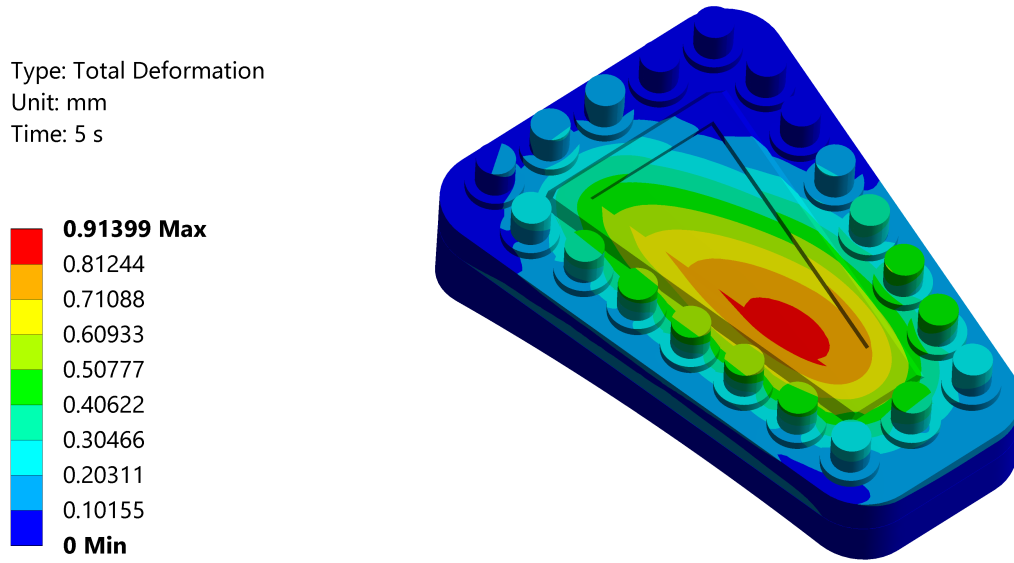


Figure 5.6: Total displacement [mm] considering the contribution of external moments.

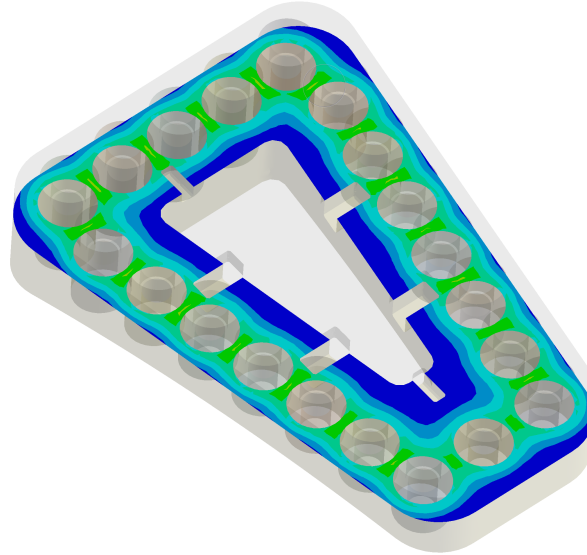
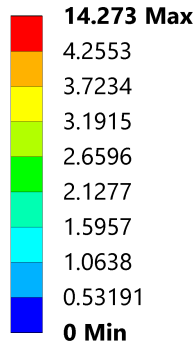
As the moment's effect increases the maximum displacement, it also localizes it at a different point on the flange. Furthermore, the inhomogeneity of the load distribution on the screws is even more pronounced. The ensuing evaluations pertain to this particular load case.

5.4.2.2 Contact pressure

In Figure 5.7 the contact pressure at the interface between the two flanges is shown following pre-tensioning and after the application of external forces and moments. As can be seen in Figure 5.7(b) the pressure between the two flanges has considerably reduced after the application of the external loads. This is mainly due to the choice of lowering the pre-tensioning of the bolting elements in order to limit the stress value associated with the residual twisting torque on them.

C: Static Structural

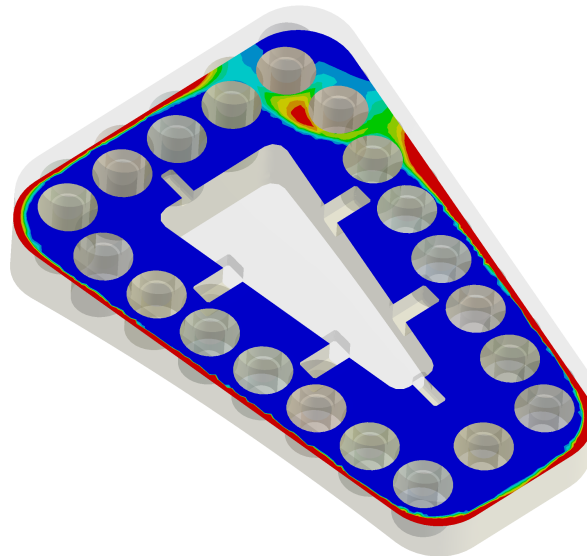
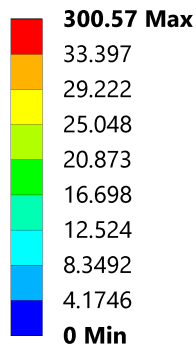
Pressure 3
Type: Pressure
Unit: MPa
Time: 2 s



(a)

C: Static Structural

Pressure
Type: Pressure
Unit: MPa
Time: 5 s

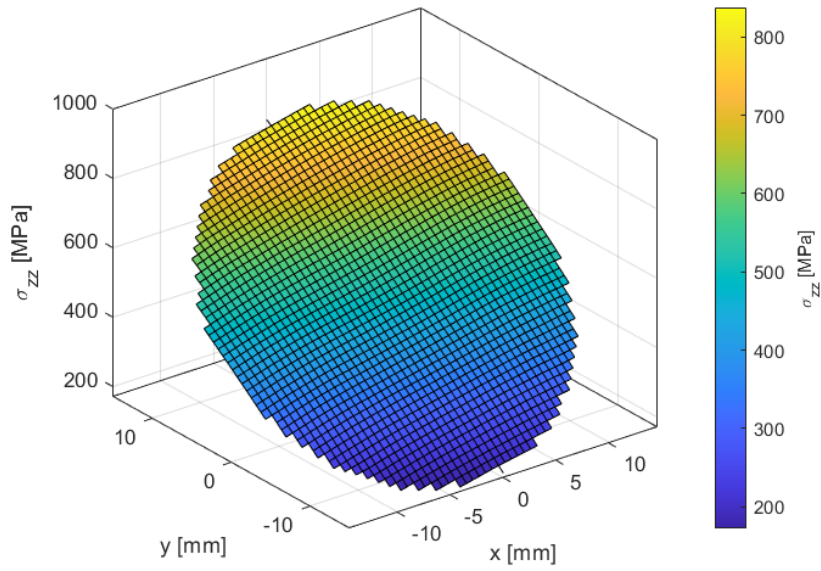


(b)

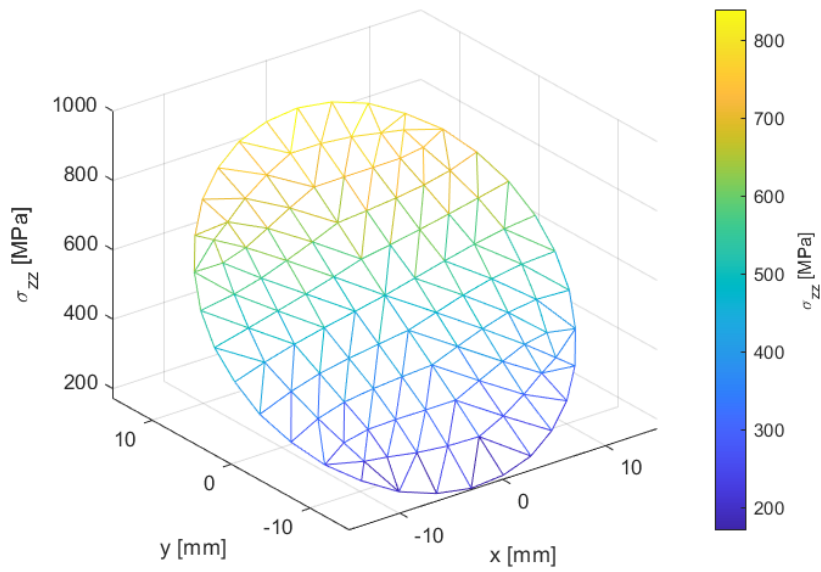
Figure 5.7: Contact pressure after pre-tensioning (a) and following the application of external loads (b).

5.4.2.3 Bolting element

With regard to the screws, in Figure 5.8(a) the normal stress coming from the numerical calculation is depicted, since the other stress components are negligible. While the stress after discretization obtained by the Delaunay procedure is shown in Figure 5.8(b).



(a)



(b)

Figure 5.8: Normal stress distribution from FE calculation (a) and after the application of Delaunay procedure (b).

As illustrated in the figure above, the tension field can be traced back to the case of a biaxial bending plus tension. The mean and maximum stresses were calculated through the procedure detailed in Section 5.1. The structural integrity verification is summarized in the table below.

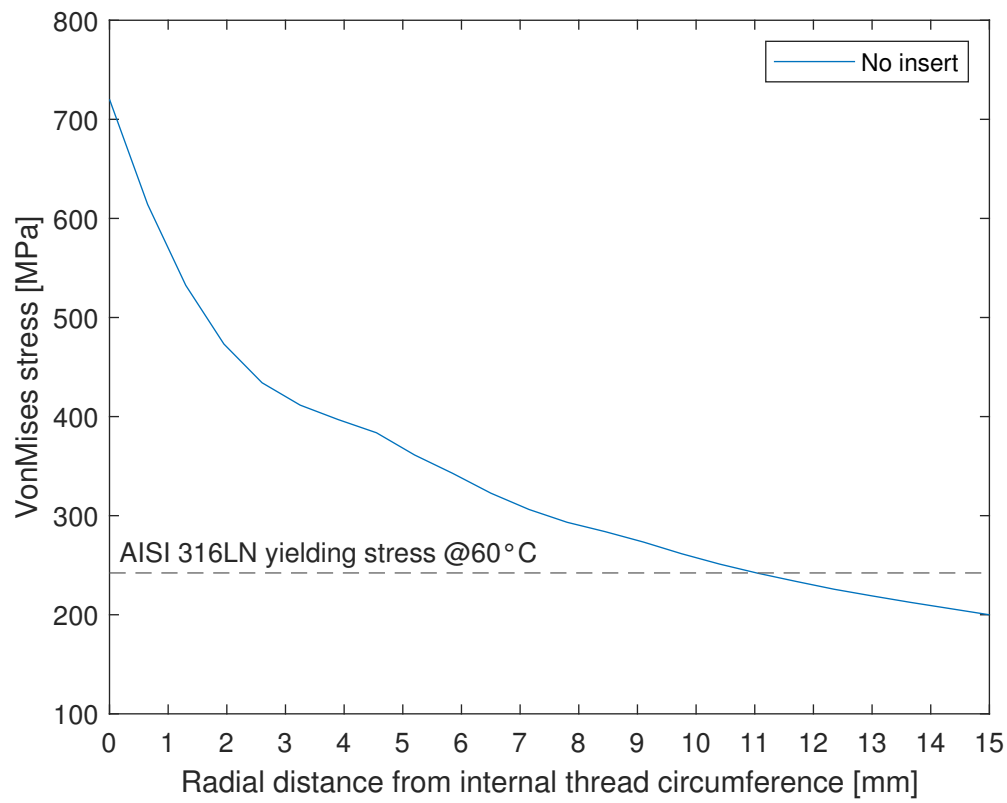
Table 5.7: Structural integrity verification of the screw core section in terms of mean and maximum stresses from FE results.

	Mechanical loads on the screw core section	
	Mean stress	Max stress
	$\frac{(\overline{\sigma_m})_n}{1.5S_{mB}}$	$\frac{(\overline{\sigma_m + \sigma_b})_n}{2.25S_{mB}}$
ASTM A453-Grade 665	0.95	0.98
Inconel 718	0.86	0.88

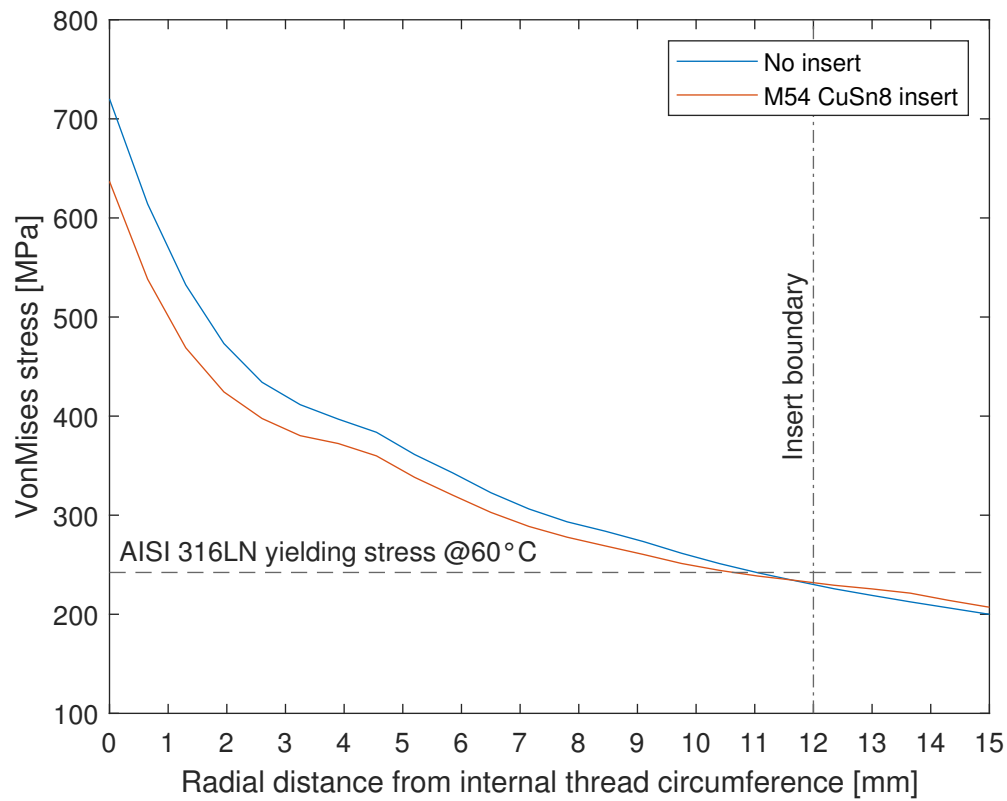
In contrast to the analytical procedure, the FE analysis shows that both the materials would be suitable for bolting elements. However, it should be noted that the safety margins associated with ASTM A453-Grade 665 are rather limited.

5.4.2.4 Internal thread

In Figure 5.9(a) the equivalent von Mises stress along a radial path starting from the interface between the screw and the flange welded to the vessel is shown, in the case that no threaded inserts are included. Moreover the yielding stress of AISI 316LN at operating temperature is reported. The diameter value of the threaded insert to be used was derived from the intersection of the two curves. This value was determined in order to ensure that the equivalent stress on the flange is below the yield strength of the flange material. Given an additional safety margin, the insert diameter was assumed to be 54mm. The material selected for the threaded insert is a copper-tin alloy, specifically *CuSn8*. In Figure 5.9(b) the comparison between the stress trends with and without inserts is depicted.



(a)



(b)

Figure 5.9: Stress inside the flange without considering the inserts (a) and with the inserts included (b).

5.5 Final remarks

As previously stated, the occurrence of a gap in a bolted joint should be prevented in any operational state. Consequently, this design offers the optimal solution related to the geometric constraints imposed by the overall assembly configuration but not an optimum in absolute terms. A different shape and orientation of the flange would have helped the design towards a better solution. The preliminary design was performed using the analytical procedure. The resulting choice of screws number and diameter was then analysed in more detail using finite element modelling. The results of the finite element analysis showed that due to the complexity of the joint being analysed, analytical analysis alone would not be able to capture all aspects related to the behaviour of the joint itself. Indeed, flange deformation alone has been demonstrated to induce inhomogeneity in the load distribution on individual screws, regardless of the presence of the external moment. Therefore, one of the two fundamental assumptions on which the analytical procedure is based is invalid. Conversely, the finite element modelling used does not allow for the capture of the contribution of residual twisting torque, which, in general, would appear to have a role in the sizing of the bolting elements. In this chapter also the use of threaded inserts in the flange that is welded to the vessel has been investigated. This investigation aimed to ascertain whether AISI 316LN could be utilised as the material for the flange, thereby avoiding potential welding incompatibilities with the vessel material. Finally, the decision was taken to utilise bronze threaded inserts.

Chapter 6

Summary and conclusions

The design of fusion machines represents a considerable technological challenge, involving a variety of closely interrelated fields of physics. The contents of the present work dealt purely with problems of thermo-mechanical analysis and verification of structural integrity for several components of the DTT fusion machine, under construction at the ENEA research center in Frascati. The work presented in this thesis involved the study of different components of the machine at different degrees of detail. The scope of the study ranged from the examination of the dynamic behavior of the toroidal complex, with a focus on its seismic response, to structural integrity verifications of the vacuum vessel under various machine operating states. The study concluded with the analysis and optimization of the design of the flanged joint that connects the chamber and its gravity supports. The structural integrity verifications of the analysed systems serves as a link between the various chapters of this thesis. The mechanical design process is deemed validated when the stresses resulting from the most critical loading conditions on the component settle below the prescribed levels outlined by the reference standards and codes. The analysis of these standards, as well as the comparison of them to highlight similarities and differences, played a crucial part in the conduct of the proposed activities. The codes used in the nuclear field, namely ASME NB, NCA, NG and RCC-MRx, were the main reference in the analysis of the vacuum vessel and flanged joint. Conversely, the NTC2018 standard served as the foundation for the study of the seismic behavior of the DTT toroidal complex. The dynamic behavior of the toroidal machine complex was studied through two different numerical approaches: the equivalent static method and the dynamic method, based on the use of spectrum analysis. Preliminary findings indicate critical issues with the cryostat base. Seismic isolation devices are being examined to mitigate the associated stresses and results show that this solution has proven effective to do so. A preliminary study of the vessel was conducted in accordance with the load specification document. This document presents the conditions in terms of temperature and pressure related to each load case deemed essential for the sizing of the component. As for the vessel, the design

loads are mainly due to EM forces and moments which arise in the passive structures when plasma instabilities occur. In this thesis, the case of a downward vertical displacement event was examined, with load distributions that have been modulated along the structure under consideration. This is achieved by implementing a toroidal peaking equal to 2. Consequently, one of the vessel sectors experiences EM loading that is twice as high as it would have been in the absence of the TPF. An analysis of this extremely severe condition highlighted potential critical issues in the vessel design, particularly in the areas around the gravity support. The submodeling technique was adopted to evaluate successive solutions for reinforcing the vacuum chamber until a structurally verified design was obtained. Parallel analyses carried out by independent groups were concerned with verifying the compliance with the codes rules of the vessel design under other loading conditions. Consequently, the contribution presented in this work was supplementary to a more extensive array of activities, all of which were ultimately directed towards the construction of the vacuum vessel. These activities included one related to the analysis of the flanged joint between vessel and gravity support. The analysis of the flanged joint was first carried out through a simplified analytical procedure by which a preliminary configuration of the joint was obtained. The overall joint behavior was therefore analysed through finite element modelling. These analyses revealed the inapplicability of the simplified analytical procedure, failing some of the assumptions underlying the procedure. In particular, the inhomogeneity of mechanical load distribution associated with flange deformation resulted in localization of stress peaks in certain bolts. The most heavily loaded bolt was verified according to the rules. Of the two materials considered for the bolt, Inconel 718 was chosen. One of the objectives in sizing the joint was the possibility of keeping the material of the welded flange the same as that of the vessel. Given the high stresses at the interface between the bolting element and the nut, the possibility of using threaded inserts of a material with superior mechanical performance was explored. This study led to the selection of bronze inserts, the use of which allowed the basic flange material to be retained.

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