Development of a master model concept for DEMO vacuum vessel

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HIGHLIGHTS

- The present work concerns the development of a first master concept model for DEMO vacuum vessel.
- A parametric-associative CAD master model concept of a DEMO VV sector has been developed in accordance with DEMO design guidelines.
- A proper CAD design methodology has been implemented in view of the later FEM analyses based on "shell elements".

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ABSTRACT

This paper describes the development of a master model concept of the DEMO vacuum vessel (VV) conducted within the framework of the EUROfusion Consortium. Starting from the VV space envelope defined in the DEMO baseline design 2014, the layout of the VV structure was preliminarily defined according to the design criteria provided in RCC-MRx. A surface modelling technique was adopted and efficiently linked to the finite element (FE) code to simplify future FE analyses. In view of possible changes to shape and structure during the conceptual design activities, a parametric design approach allows incorporating modifications to the model efficiently.

1. Introduction

One important objective of the EU fusion roadmap Horizon 2020 is to develop a conceptual design of a *demonstration fusion power reactor* (DEMO) to follow ITER, capable of generating several hundreds of MW of net electricity to the grid and operating with a closed fuel-cycle by 2050. Most nations involved in the construction of ITER view DEMO as the last step towards the actual exploitation of fusion power [1].

Indeed, with the construction of ITER well underway, attention is now turning to DEMO that should pave the way to future fusion-based commercial reactors. At the time of the present work, no conceptual design exists of DEMO reactor; being the work carried out in Europe till now mainly focused on assessment of safety, environmental and socioeconomic aspects of fusion power [2].

The present work concerns the development of a first master concept model for DEMO vacuum vessel (VV). One of the objectives was to develop efficiently to manage CAD model in view of the likely changes in VV structure required during the conceptual design activities on DEMO.

The definition of a conceptual model for small structural elements, as well as for large assemblies, is a step-by-step path that starts from a sketch and ends with a preliminary assessment of different possible design solutions. The complexity of projects such as DEMO obliges to use computer-aided applications for both modelling and structural assessments. In particular, the correct set-up of the CAD environment and adoption of proper modelling methodologies are very important points to consider when approaching a new project, especially during a conceptual design phase, when changes to CAD models are likely to be very frequent. In other words, the digital model has to be easy to maintain and to be changed. Moreover, it would be better if the CAD environment could keep a strong connection (so-called *associativity*) with FEM analysis environment, even after major CAD changes. In this way, the same load and the same boundary conditions can be applied to different variants, without have to rebuild the entire FEM sim-

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ulation model. It is clear that this potentially allows saving a considerable amount of time.

Based on these considerations, the authors have prepared a first 3D conceptual model of DEMO VV to be used for later structural analyses under the main loads (i.e. dead weight and VDE).

2. General background

As mentioned, the conceptual design phase is an iterative creation process aimed at developing different *concepts* that potentially meet the "mission need", but have yet to be further analysed and evaluated [3]. In this phase, more than in the others, major changes occur constantly, thus a tight link between CAD and FEA models is crucial to speed-up the whole design process [4,5]. Currently, there are two main approaches to generate computer-aided concepts: CAD-centric and FEM-centric [6,7]. The first approach is widely adopted: the main design activity is conducted on CAD systems where the concepts are improved and refined time by time through an iterative process involving periodic design review and consequent geometrical changes. Unfortunately, CAD models are often unsuitable for FEA needs [7], and therefore an *idealization process*, involving details suppression as well as geometrical adaptations, is necessary. Moreover also other simulation codes, such as MCNP [9] used for complex facilities like the ITER, rely on their own geometry description and the data conversion need external tools [10]. This means that, whenever a change occurs, the CAD to "simulation environment" adaptation must be carried out again.

In a FEM-centric process [8], idealized models are used as actual design concepts before developing a reference CAD model. This approach is used especially in a conceptual design stage, but it makes it more difficult to implement major geometrical design modifications.

In any case, both approaches require to maintain two different models for the same product, with consequent wasting of time and efforts.

Modern CAE systems, like CATIA V5, do provide integrated FEM tools inside the same CAD modelling platform, but these integrated tools mostly do not have the same functionalities as stand-alone FEA systems and thus cannot be suitable for complex designs that involve different physical aspects (e.g. non-linear effects, electromagnetic interactions, dynamic effects, elasto-plastic models, etc.). Therefore, several authors are focusing on the down-stream connection between CAD models and FE analysis tools. Most of their approaches are based on neutral exchange data formats (STEP, XLM, etc.) that yet cause an "interruption" between the CAD model and FEM model.

Other authors are addressing CAD-CAE integration. In particular, Lee [7] proposed an integrated approach that involved a multi-abstraction non-manifold topological (NMT) modelling system. According to this methodology, for each modelling operation, multiple geometric features would be embedded into a single NMT master model. In other words, different types of geometric entities (the ones suitable for design, the other ones for analysis) would be concurrently created and modified. Then, the needed CAD or CAE model would be "extracted" as and when required.

However, this approach has some evident limits highlighted by the author himself and in facts does not help the creation of concept variants. Regardless, modern CAD systems, do not implement such a multi-abstraction modelling core, even though NMT modelling is fully integrated in most of them, being especially used in conceptual design phases.

Thus, the present work does not keep insisting on CAD-CAE *integration*, but instead focuses on a design methodology that uses the already-available functionalities of modern CAD/CAE tools, such as CATIA V5 and ANSYS, to simplify variants generation during the conceptual design phase and also to keep *associativity* between CAD and CAE environments.

More specifically, the authors propose a CAD-centric design approach improved with a proper Parametric Associative (PA) model. **This methodology is than applied to the conceptual design of DEMO VV.**

A PA model is a computer-based description of a geometrical model that depends on non-geometrical entities called *parameters* [11]. Parametric systems solve constraints by applying sequentially assignments to model variants [12]. Moreover, any parameter-related modification can be automatically propagated to down-stream applications and geometries, keeping the relationship among geometrical objects and features in diverse design process steps [11]. In particular, ANSYS provides a direct interface with the most common CAD systems that help to keep data consistence with the geometrical models even after design changes. Moreover CAD parameters can be recognized and changed inside the same CAE environment, without have to re-build the reference model.

But, to take advantage of these characteristics, greater attention should be paid on **how a PA master model has to be structured** and handled to be efficiently linked with FEM environments.

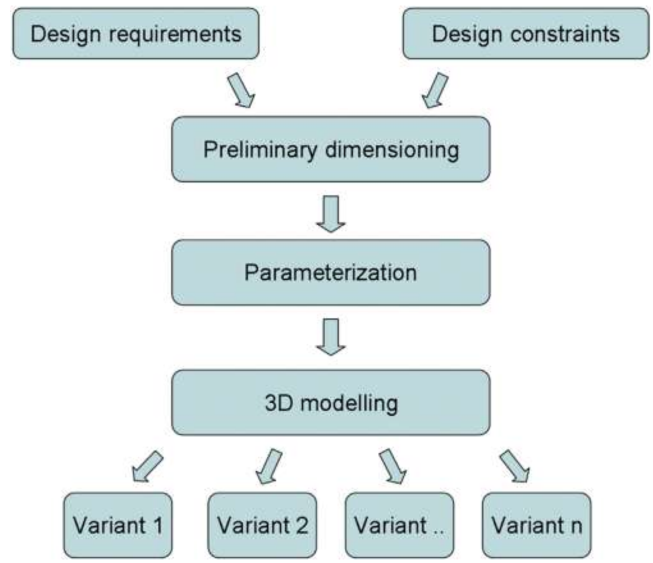

Fig. 1. Master model concept definition workflow.

3. Design workflow for conceptual design of the vacuum vessel

The development of a master model concept for a large assembly should follow the design workflow shown in Fig. 1. Such a workflow is made of several phases:

- Collection of the design requirements (loads, applicable standards, materials, temperatures, etc.)
- Identification of the main design constraints (overall dimensions, cost, interference issues, maintainability, main technological aspects, etc.)
- Preliminary dimensioning
- Identification of the main design *parameters* (e.g. thickness of plates, distance between structural ribs, etc.)
- Development of a *parametric* 3D master model
- Generation of geometrical variants for later assessments (structural as well as cost analyses, technological feasibility studies, etc.)

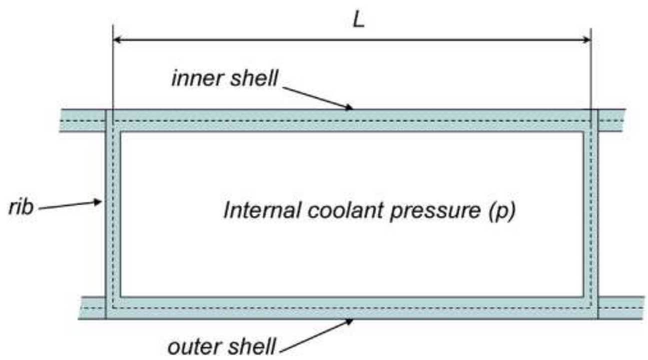

Fig. 2. Shells and ribs structure of VV.

Table 1
Main requirements and constraints for VV.

Material	AISI 316 L(N)
Operating Temperature	200 °C
Coolant Pressure (p)	3.15 MPa
Applicable standards	RCC-MRx RB 3251.112, A3.1S.43
Vessel design	All-welded double-shell structure
Shell Thickness (H)	60 mm
Standard Ribs Thickness	40 mm
Nr. of toroidal sectors	16 (22,5° each)

In particular, the identification of a properly small set of parameters driving the 3D geometry (namely, dimensions or properties that are most likely to be changed during the design process) is a key point, especially in a conceptual design stage. A well-conceived parametric model can indeed be updated by changing a small set of values/properties rather than by deleting existing geometries and creating new ones. In this context, the term "*parametric*" has a broad sense, because *Boolean parameters* can be also used to switch among different configurations belonging to the same master model [13]. Parametric 3D models already have well-known advantages over other conceptual 3D sketching techniques [14] but here it is worth emphasizing that this methodology also improves the *associativity* between CAD and FEM models, even when a design variant implies significant changes in terms of shapes and layout.

As mentioned, the authors have followed such a workflow to design a master model concept of the DEMO vacuum vessel. Thus, in the next sections, the main phases of the conceptual design will be discussed step by step.

3.1. Design requirements and design constrains for DEMO vacuum vessel

The VV is a toroidal chamber located inside the magnet system aimed at providing an enclosed vacuum environment for plasma. Also, it acts as a first confinement barrier; thus the nuclear pressure vessel design code RCC-MRx must be considered in its design. The torus shape of VV will be divided in 16 separate sectors of 22.5° each.

The selected material for VV is AISI 316L(N) stainless steel. The heat transferred to the vessel is actively removed by water circulating in-between the double-shell structure. To withstand the coolant pressure the double-shell steel structure of the DEMO VV is internally reinforced by proper stiffeners (so-called *ribs*, see Fig. 2).

The main design requirements and design constraints for the vessel are summarized in Table 1.

The reference load specifications are reported in more detail in [15] and [18]. Moreover, the design activity followed the task guidelines for design and analysis of DEMO vacuum vessel [17].

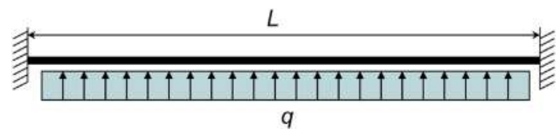

Fig. 3. equivalent static scheme of shell structure.

3.2. Preliminary dimensioning of DEMO vacuum vessel

The first step to preliminarily dimension the VV is the definition of the maximum distance between the reinforcing elements, given the shell thickness.

The maximum admissible spacing of the ribs also sets the minimum number of ribs for each VV sector. Preliminary bending stress calculations were carried out starting from the data in Table 1.

According to RCC-MRx RB 3251.112 [16], the primary membrane plus bending stress shall not exceed$1.5 \cdot S_m$:

$$P_m + P_b \leq 1.5 \cdot S_m \tag{1}$$

where P_m is the primary ***membrane stress*** and P_b is the primary ***bending stress***.

For AISI 316L(N) stainless steel, RCC-MRx A3.1S.43 [16] suggests an yield strength at 200 °C of:

$$S_m = 130 \text{ MPa} \tag{2}$$

Thus, since P_m is negligible in the case at issue, Relation (1) can be written as:

$$P_{b\,\max} = 195\text{MPa} \tag{3}$$

Due to the symmetry of loads and geometry, the structure shown in Fig. 2 can be conceived as an over-constrained beam (see Fig. 3), that is loaded with a distributed load q

$$q = p \cdot B \tag{4}$$

being B the developed poloidal length of a single shell element.

As is known, with reference to the static scheme shown in Fig. 3, given the operating pressure p, L/H ratio of shell and ribs structure can be written just as:

$$\frac{L}{H} = \sqrt{2\frac{P_{b\,\max}}{p}} \tag{5}$$

Finally, using input data in Table 1, we get 667 mm as the maximum width of a single shell on the equatorial plane. However, this value must be increased with the thickness of the ribs themselves which has been neglected till now. Therefore, the maximum allowable distance between two ribs actually is:

$$L_{ribs} = 707\text{mm} \tag{7}$$

3.3. Identification of main design parameters

The most important components of the VV are the main vessel, the port structures and its supporting system.

The study of the layout for ribs allowed defining datum planes and angles on which the ribs had to be placed. It is understood that ribs profiles are given by intersection between ribs reference planes and the mentioned reference surfaces, while shells are the parts of reference surface between two consecutive ribs.

Moreover, the choice of ribs reference planes had to respect the following conditions (see Fig. 4) [17]:

- Ribs should be as close as possible to the centre line of the *breeding blanket segments*
- All ribs must be symmetrical to the centre plane of the sector, where there should be a rib

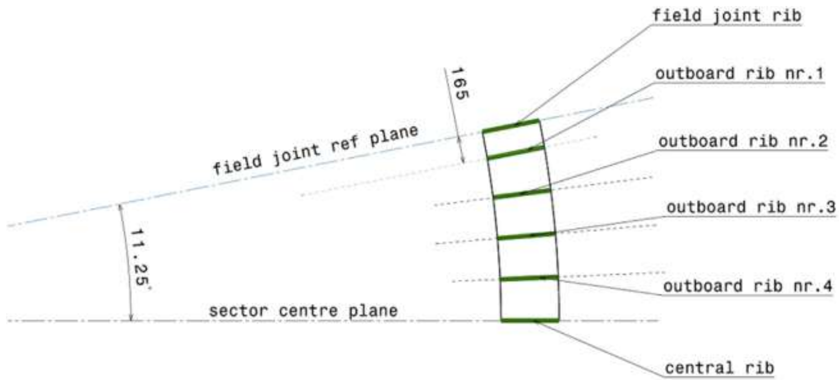

Fig. 4. Outboard ribs references and constraints.

- The ribs at the two sides of each sector must be 165 mm off the symmetry plane between two VV sectors to provide space for an ITER-like splice plate at the field joint.

These requirements have been translated in proper constraints and relations.

Parameters involved the location of the supports (see Section 3.4.4) and the ribs spacing (actually, for symmetry reasons, the number of ribs between ribs nr.1 and central reference plane have been parameterised, see Fig. 4).

3.4. CAD model of the main vessel

The modelling of the main vessel structure started from two reference surfaces corresponding to the inner and the outer side of the vessel respectively (see Fig. 5). This reference model was provided by the EUROfusion Program Management Unit (PMU).

The VV was modelled as a surface geometry, rather than a 3D solid body. In other words, just the profiles of the structure were modelled in CAD environment, while the actual thickness of shell and ribs will be made explicit at the time of later FEM analyses (Fig. 6).

Indeed, given the well-known assumptions of Kirchhoff–Love theory of plates [19], which are suitable for the purposes of this analysis, shell models have two advantages over solid models:

- Meshing of surface models is less time-consuming than the one of solid models,
- Wall thickness can be changed in FEM environment without building up a new 3D model and thus a new 3D mesh.
- The degrees of freedom of the FE model are significantly reduced.

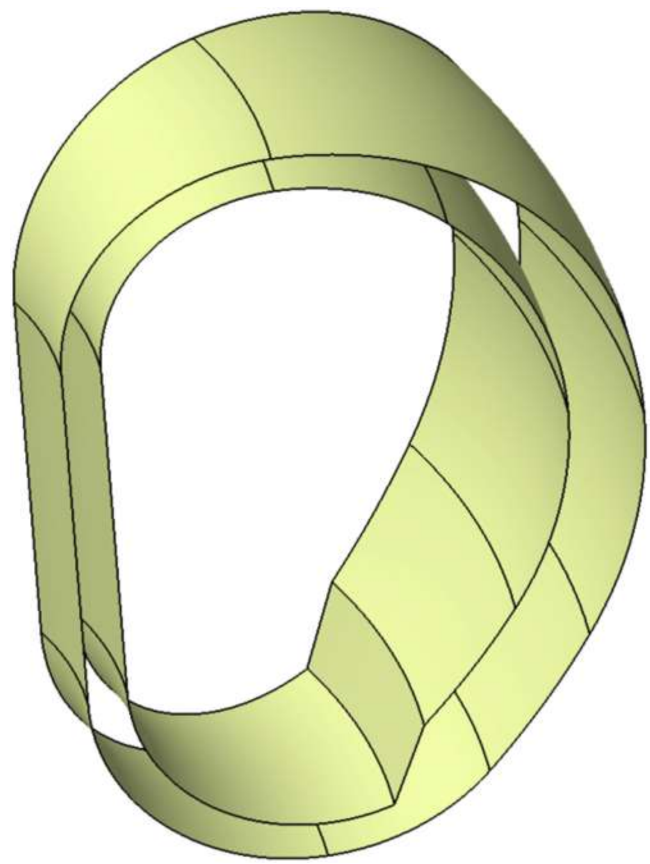

Fig. 5. Inner and outer reference surfaces for main vessel.

All surfaces have been obtained by the revolution of single-curvature profiles drawn on a poloidal plane around the symmetry axis of the torus, except at inboard side, where both inner and outer surfaces are cylindrical and thus have a single curvature on any toroidal plane and no curvature at all on any poloidal plane.

The VV has a torus shape and therefore the arc length of the walls at *inboard* and *outboard* sides of VV torus is different. For this reason, with reference to any equatorial section of vessel sector, while eleven ribs were placed on its outboard side, only five ribs were put on the other side (Fig. 7).

All ribs are 40 mm thick except for the poloidal ribs number 3,4,8,9 on the outboard that are 80 mm thick, which are aligned and joined with the *gusset plates* that support equatorial and lower ports. This choice guarantees the structural continuity in order that loads can be safely exchanged between ports and main vessel [20]. The gusset plates are 100 mm thick and are joined with the side-walls of the ports through machined components that have been modelled as two short ribs (see Fig. 8).

Given the shape of the vessel sector four short poloidal ribs have been added both at top and bottom of the inboard segment (see Fig. 9), in order to keep the ribs spacing less than 707 mm everywhere. These ribs are joined together through one toroidal rib. Also in this case the poloidal ribs at the inboard side are aligned with the poloidal ribs at the outboard side.

Finally, each shell connecting two adjacent ribs was modelled 60 mm thick. The shells do not follow the reference surface exactly. The final surface is in fact mostly faceted because single-curvature shells have been used rather than double-curvature ones mainly for technological feasibility reasons, except with reference to top and bottom surfaces at inboard side, where the double-curvature has been kept.

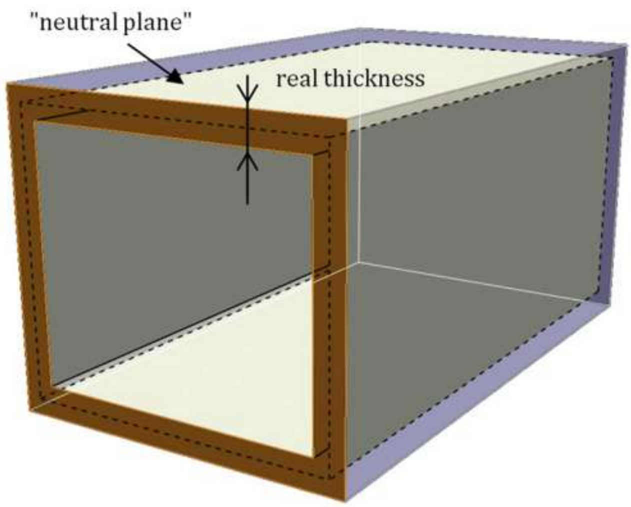

Fig. 6. Surface model correspondent to a thick structure.

3.4.1. Upper port

The upper port sidewalls lying on poloidal planes are single-walled and welded to both inner and outer shells. The cooling concept of these sidewalls will be studied in the near future. Instead, both the walls facing the inboard and the outboard sides of the VV have the same box structure as the main vessel (see Fig. 10). The ribs are aligned to those of the main VV and are parallel to longitudinal axis of the port. This ensures a structural continuity between the upper port and the main vessel.

One toroidal rib has been placed inside the main vessel and aligned with the outer shell of the upper port, as shown in Fig. 11.

3.4.2. Equatorial port

The equatorial port was modelled using a double walled structure with ribs and shells. In particular, three ribs have been provided for each sidewall. The ribs inside the top and bottom walls are aligned to the ribs of the VV, while the ribs inside the other two walls of the port are parallel to the vessel equatorial plane (Fig. 12).

3.4.3. Lower port

The lower port has the same box structure as the equatorial port. The ribs inside the walls of the port shown oblique in the figure are perpendicular to the inner and outer shell and parallel to each other. On the top and bottom the ribs are aligned with VV ribs (Fig. 13).

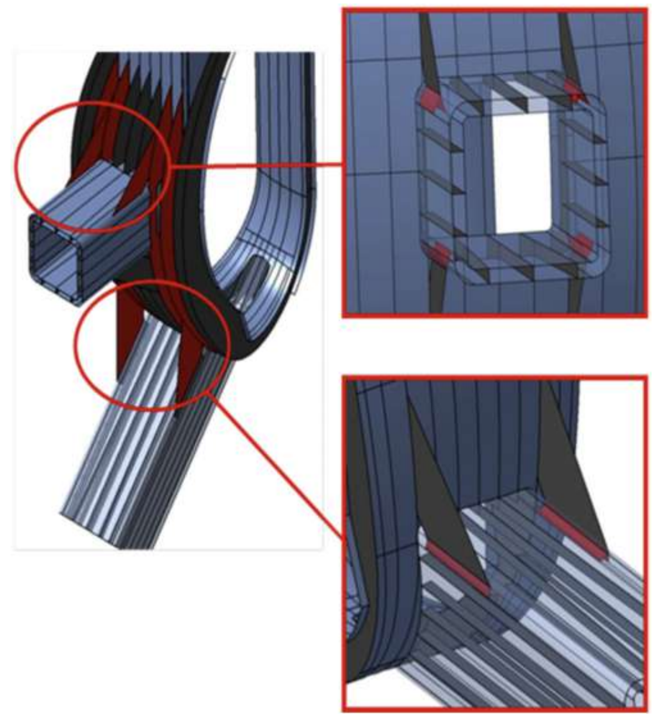

Fig. 8. Gusset plates aligned with the corresponding ribs.

3.4.4. Supports

At the current stage, a simplified design of the DEMO VV supports has been developed to provide a coherent model for structural analyses. Several configurations have been provided, considering supports located either at the lower port or at the equatorial port. Each support is considered welded to the sidewall of the corresponding port (Fig. 14). It is understood that different supporting ports imply different locations for the support plates. Proper Boolean parameter allows switching between two possible configurations. Moreover, also their radial coordinate of the supports has been parameterised.

3.5. Variants generation

The PA approach has been applied to different geometrical aspects of the CAD model. In this way, many different combinations can be generated from the same master model (Fig. 15).

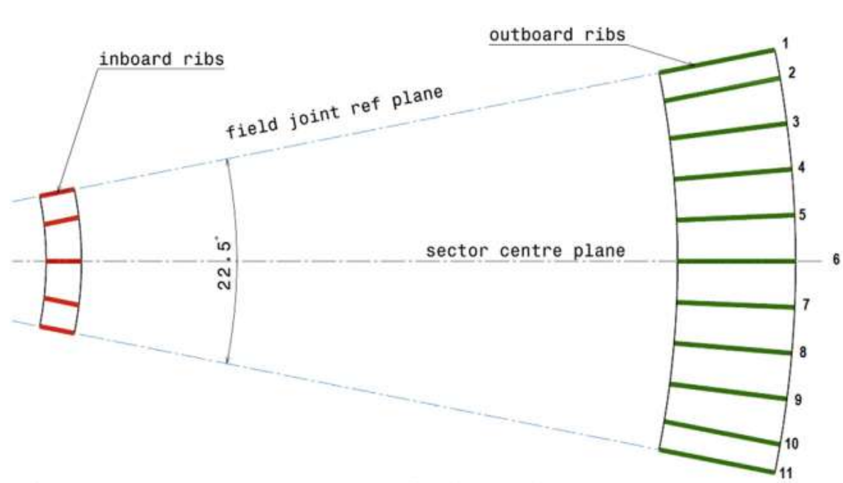

Fig. 7. Layout scheme of ribs at inboard and outboard sides of VV sector (not in scale).

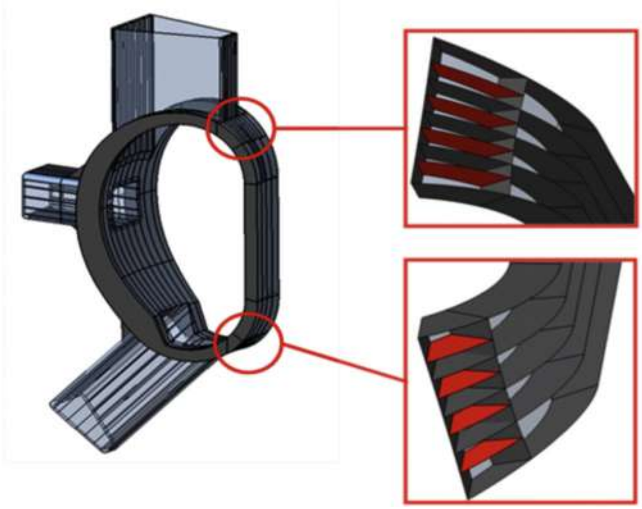

Fig. 9. Poloidal ribs on the inboard segment.

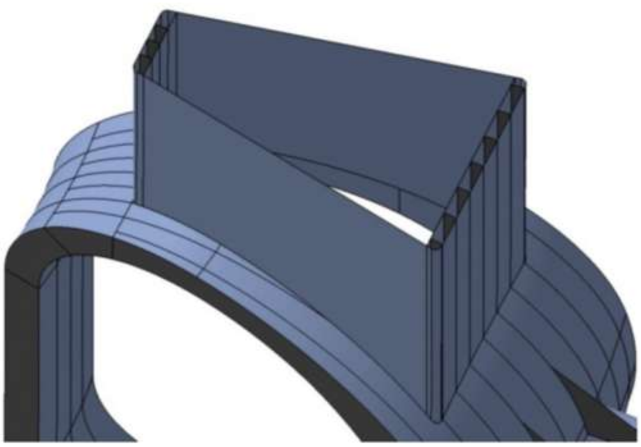

Fig. 10. Upper port.

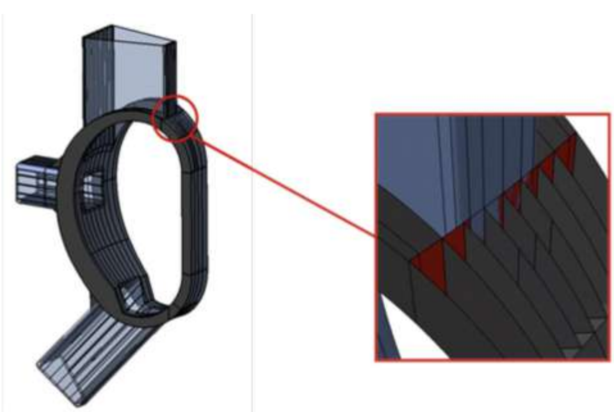

Fig. 11. Alignment of toroidal ribs with the upper port structures.

As mentioned, the distance between the ribs, as well as the length of the ports and the configurations of the supports, have all been defined through proper parameters.

4. Results

The *associativity* between the CATIA master model and the FEA environment (ANSYS) has been then tested. For instance, in Fig. 16

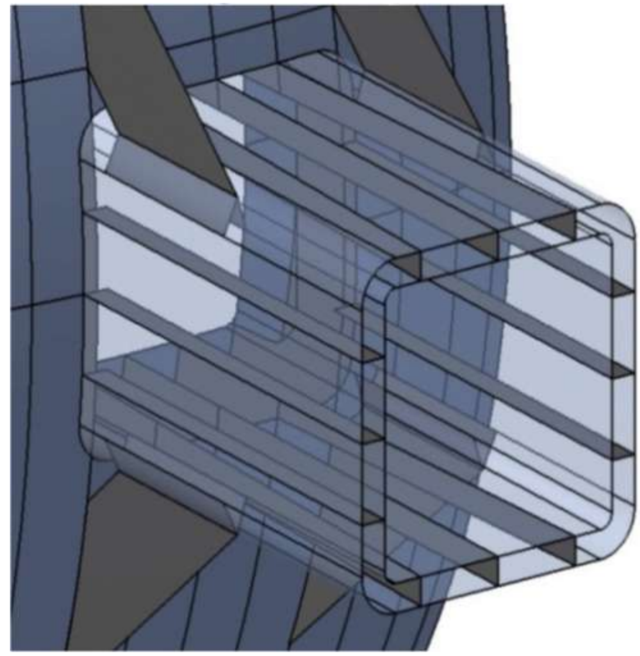

Fig. 12. 3D model of the equatorial port.

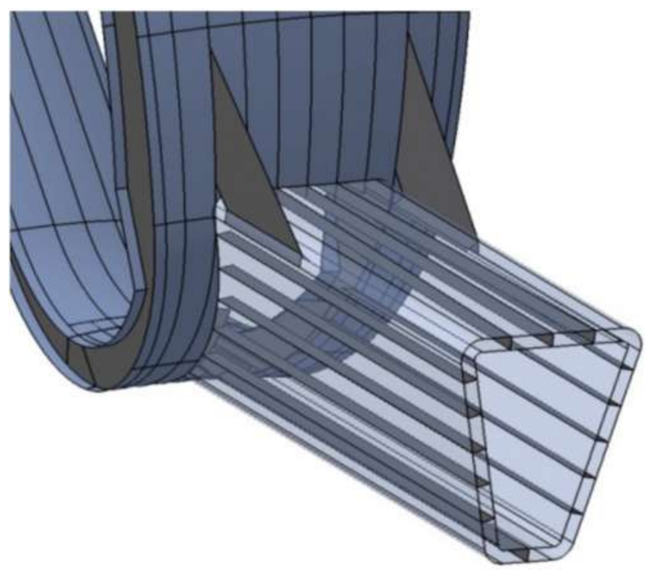

Fig. 13. 3D model of lower port.

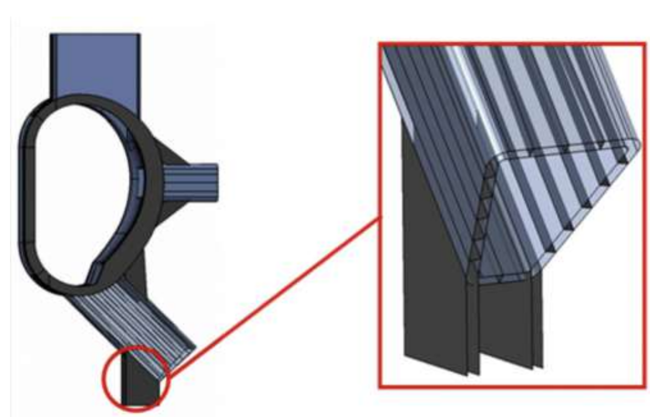

Fig. 14. Supports on the lower port jointed to the four port sidewalls.

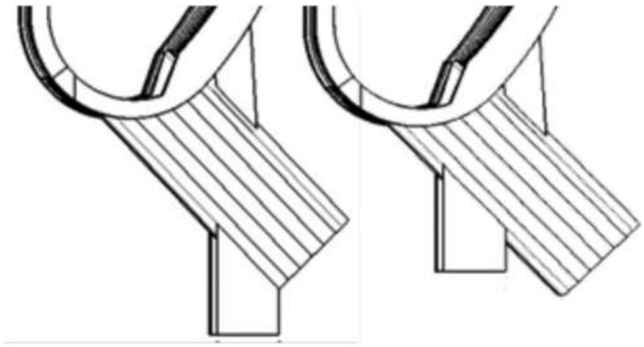

Fig. 15. Different configurations for lower supports.

an example of FEM analysis with reference to two different configurations for lower supports is shown.

Moreover, when needed, the design parameters can be accessed and changed in FEA environment by means of proper software plug-ins. The *design exploration* tools provided by ANSYS 15 allow easily optimising the design depending on defined parameters. In particular, the *response surface* tool was used to finding out the maximum value for the spacing of the ribs inside the vessel by means of a design of experiment chart (Fig. 17).

The results of the optimization process are consistent with the preliminary dimensioning presented in section 3.2 (Fig. 18).

5. Conclusions and future work

A parametric-associative CAD model concept of a DEMO VV sector has been developed in accordance with DEMO design guidelines. Starting from just two reference surfaces, the minimum space between the ribs of VV box structure was preliminarily determined in compliance with RCC-MRx and design requirements.

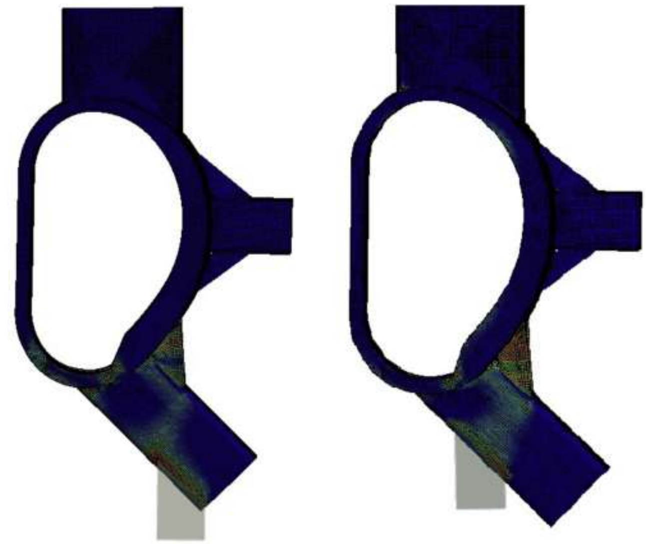

Fig. 16. Structural analyses for different support configurations.

Ports, gusset plates and internal structure of the vacuum vessel were modelled accordingly with CATIA V5.

From this point of view, a proper CAD design methodology has been implemented in view of the later FEM analyses based on "shell elements".

Through this approach the design of the DEMO VV internal structure and ports have been developed.

Currently this model is being effectively used for detailed structural assessment on the vacuum vessel and its supports. Some partial results were also highlighted.

This work showed that a CAD-centric approach can be as effective as a FEM-centric approach for structural analyses also in a conceptual design phase thank to the features of the modern CAE

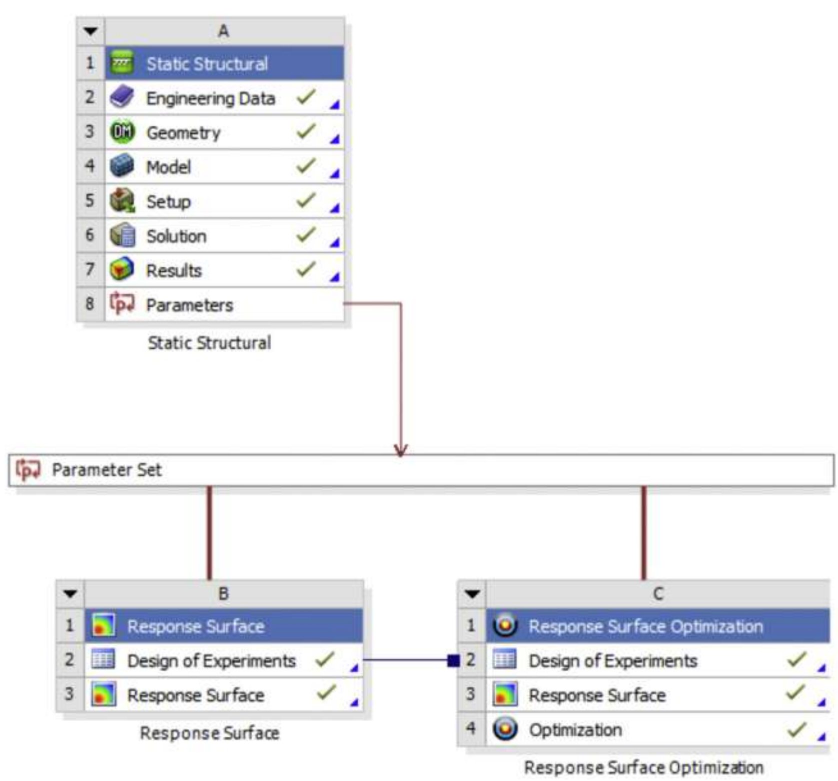

Fig. 17. Optimization process in Ansys Workbench.

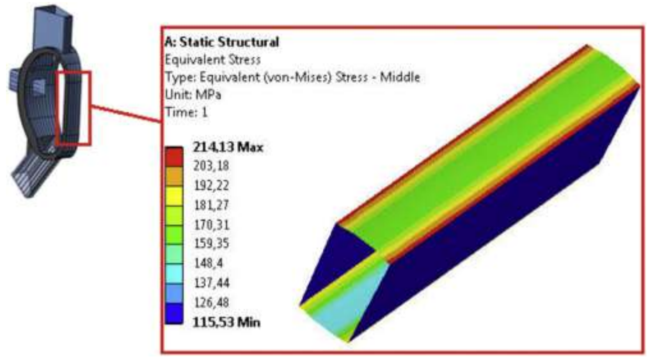

Fig. 18. Detail of stress distribution in a shell element at inboard side of VV.

software tools. The parametric approach used for computer-aided design of the VV makes indeed any change to vessel shape or its internal structure easy to implement. Moreover, the associativity between CAD and CAE models is kept even after major changes. This aspect has a huge impact especially in a conceptual design phase when the number of design changes is expected very high. In particular, Boolean parameters have been used to switch between different support configurations, while dimensional parameters drive the distance between the ribs. It is worth emphasizing that these parameters can be changed directly inside the CAE environment, without have to re-load the original CAD model. This allows conducting some analyses about geometrical variations that have not to be necessarily propagated to the original CAD model.

This will also give the opportunity to compare the methodology proposed with other design methods found in the published literature.

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