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Thermal and structural analysis in support of the conceptual design of the DEMO helium cooled pebble bed breeding blanket intermediate heat transfer system

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ABSTRACT

The EU DEMO fusion reactor has been conceived to deliver net electricity. Therefore, it must be equipped with a Balance of Plant (BoP) system aimed at removing fusion-generated heat power and transforming it into electrical power. Different BoP architectures are currently under investigation within the EUROfusion consortium, in order to meet needs and constraints of the in-vessel systems, with particular regard to the different blanket concepts.

In this regard, the BoP system involving helium cooled in-vessel components has been considered in this work, meaning that the design of the DEMO reactor endowed with the Helium Cooled Pebble Bed breeding blanket (HCPB BB) has been taken into account. Specifically, the Indirect Coupling Design (ICD) HCPB BoP variant has been assessed in this work, as it is the reference configuration selected for the Conceptual Design Phase of DEMO. It foresees an Intermediate Heat Transfer system (IHTS), aimed at decoupling the BB- Primary Heat Transfer System (PHTS) from the power conversion system. Therefore, in the present work thermal and structural numerical analysis in support of the design of the DEMO HCPB IHTS are presented. In particular, the pipe stress analysis of the whole IHTS is first presented, highlighting pros and cons of the current piping layout and proposing potential design improvements. Then, a detailed 3D thermal and structural assessment of the IHTS main Heat Exchanger is reported, in order to show its soundness against the prescribed thermal and structural design criteria.

The models, the assumptions, the considered loading scenarios as well as the obtained results are reported and critically discussed, highlighting their impact on the advancements of the DEMO HCPB BoP system design towards the conceptual stage.

1. Introduction

The European DEMOnstration (EU DEMO) fusion reactor will be the first in charge of producing electricity and, for this reason, it must be equipped with a proper Balance of Plant (BoP) capable of withstanding the challenging operating conditions that are typical of this type of system. In fact, DEMO will be characterized by a pulsed operating phase and the BoP has to cope with these oscillating loads. In this context, the development of a robust architecture for the BoP is a crucial aspect of the research activities on DEMO fusion reactor [1–3]. Therefore, in this paper, research activities aimed at advancing the design of a BoP system for DEMO are presented.

Different concepts of DEMO Balance of Plant (BoP) system have been assessed during the DEMO pre-conceptual design phase, identifying criticalities and issues that must be addressed in the next project phases. This necessity arises from the different features of the served in-vessel

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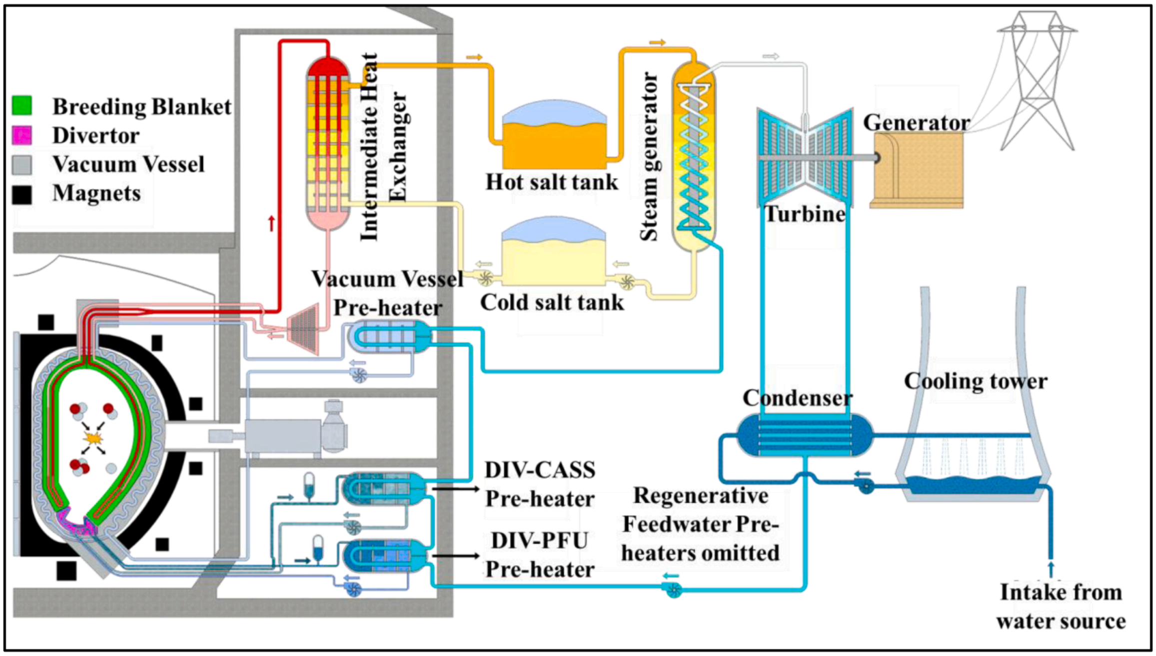

Fig. 1. HCPB BoP ICD simplified scheme [1].

components, essentially consisting in the two different Breeding Blanket (BB) concepts, namely the Helium-Cooled Pebble Bed (HCPB) BB and the Water-Cooled Lithium Lead (WCLL) BB [1]. Together with the main tokamak cooling systems the DEMO BoP also includes a Power Conversion System (PCS), devoted to convert heat power into electricity. In the case of HCPB BB, several BoP variants have been conceived and, at the end of the DEMO pre-conceptual design phase, the Indirect Coupling Design (ICD) has been selected as reference due to its optimal performance to provide steady electricity production. Moreover, this design solution permits the mitigation of the potential negative impact of the cyclic thermal loads on the turbine and other PCS equipment. It comprises three different loops: the Primary Heat Transport System (PHTS), the Intermediate Heat Transfer system (IHTS) and the PCS, as schematically shown in Fig. 1.

In this scheme, the role of the IHTS is to decouple the PCS from the BB PHTS to cope with the challenges posed by the intrinsic pulsed nature of the DEMO machine [1], thanks to the adoption of a molten salt as intermediate fluid and a dedicated Energy Storage System (ESS). In particular, during pulse, 90 % of the thermal power from BB PHTS is delivered to the PCS to supply the grid, while 10 % is stored in the ESS. During dwell time, the ESS releases energy to the PCS, so that the steam turbine is running constantly over the pulsed operation of the plasma. Further details on the IHTS architecture and operation can be found in [1,2,4].

In the present work, its reference layout is assessed from thermal and structural standpoint in order to investigate its performances in view of the prescribed design requirement, with the aim of contributing to the advancement of the IHTS design from its pre-conceptual to the conceptual stage. To this purpose, in Section 2, the structural analysis of the whole IHTS loop is presented. To this scope, the pipe stress analysis technique has been adopted. It is an assessment approach for the calculation of stress and displacement in large pipelines under selected loading conditions to ensure the structural integrity, according to a selected standard, while maintaining the system operability. In this study, the assessment has been carried out according to the ASME BPCV Section III standard, adopting the well validated Rohr2 v33.0 commercial code [5]. Then, in Section 3, the 3D thermal and structural analysis of the DEMO HCPB BoP ICD Intermediate Heat Exchanger (IHX) is reported. The latter study has been performed adopting a numerical

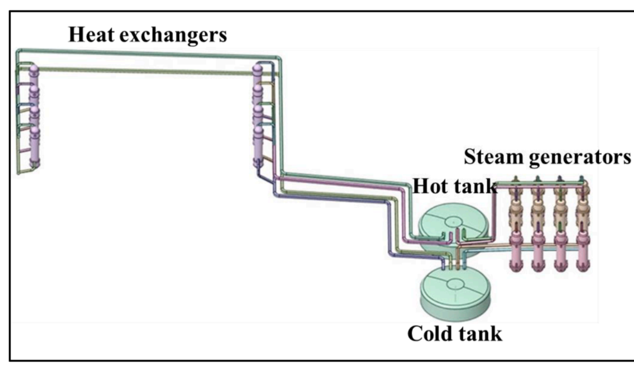

Fig. 2. The geometric layout of the DEMO HCPB BB IHTS.

approach based on the Finite Element Method (FEM) and using the Ansys Workbench v. 2023 R2 calculation suite [6]. The ultimate scope has been the verification of the fulfilment of the structural design criteria prescribed by the ASME BPVC - Section VIII DIV I code under the envisaged nominal loading conditions. The main achievements of the study have been the design review of the IHTS loop and the verification of the compliance with the structural design criteria of the proposed Heat Exchanger layout.

2. The pipe stress analysis of the HCPB IHTS

The structural performances of the DEMO HCPB BB IHTS under the foreseen nominal thermal and mechanical loading conditions have been investigated. To this purpose, the pipe stress analysis technique has been adopted and the Rohr2 v33.0 commercial code has been used. In this regard, the load combinations representing most significant time step of a typical DEMO pulse-dwell operational cycle have been considered, properly performing pipe stress analyses. Then, once selected the most critical, a supporting system and the necessary geometry modifications to the pipeline have been determined, adopting a try-and-fail approach to find a configuration able to obtain acceptable displacement and stress fulfilling the provisions of the ASME BPCV Section III standard.

2.1. The DEMO HCPB BB IHTS

The reference geometric layout of the DEMO HCPB BB IHTS is shown in Fig. 2. Here, the ESS is realized as a classical two-tank (hot and cold tank) solution. The HITEC molten salt [1] is adopted as intermediate fluid. The current design of the DEMO HCPB BB IHTS envisages Intermediate Heat Exchangers (IHXs), thermally coupled to the PHTS, two molten salt tanks (storing hot and cold HITEC), Steam Generators (SGs) delivering power to the BoP system, the pumps, and the necessary piping [1,2].

2.2. The adopted methodology

In order to assess the structural performances of the DEMO HCPB BB IHTS, the pipe stress analysis technique has been adopted. It is based on a simplified 1D approach which allows assessing large hydraulic loops with a reasonable computational burden without a significant loss in results reliability. Numerical models for pipe stress analyses in Rohr2 are set-up as a series of 1D beam elements creating a depiction of the loop geometry. All the served and auxiliary components are not directly included in the 1D model, but they are considered by means of the introduction of proper boundary conditions. Applying the linear elasticity theory, the solver performs a flexibility analysis to calculate the moments and stress components along all the directions (normal and shear stress components) and to ultimately obtain the system displacements. Then, a stress analysis is performed adopting a specific code&-standard, which in its turn depends on the system purpose and classification. In this study, three load cases are assessed:

- Primary Loads (indicated in the following as SSL): which includes the primary (i.e. mechanical) loads;
- Secondary loads (indicated in the following as SE): including axial thermal expansion;
- Total Loads (indicated in the following as STE): which considers all the loads acting during the normal operation loading scenario (primary loads and the effect of the axial thermal expansion).

In order to verify the structural performances of each piping circuit, the calculated stresses need to be checked. In this study, the provisions of the ASME Boiling and Pressure Vessel Code (ASME BPVC), Section III, Subsection NC, Class 2, Article NC-3600 [5] have been adopted. For the given load combinations, stresses are calculated. In this paper, only nominal load combinations (belonging to Level A) are considered. Then, the following set of equation is assumed.

For what concerns the stress due to sustained loads (namely the primary stress) it is given by:

$$S_{SL} = B_1 \frac{2Pd^2}{D_0^2 - d^2} + B_2 \frac{M_A}{Z} \tag{1}$$

where S_{SL} is the primary stress. In this equation, P is the pressure imposed to the pipe element, whereas d and D_0 are the pipe inside and outside nominal diameter, respectively. Concerning B_1 and B_2, they are the component-dependent stress indices defined in Table NC-3673.2(b)-1 [5]. For straight pipes, $B_1 = 0.5$ and $B_2 = 1$ (Table NB-3681(a)-1). For those components not defined within the ASME code, the Rohr2 solver assumes $B_1 = 0.5$ and B_2 as a function of the stress intensification factor, "i". The latter is also defined in Table NC-3673.2(b)-1 for some components and, for the components not reported in the ASME code, can be defined as per equation NC-3673.2(h) stating that:

$$i = \frac{C_2 K_2}{2} \tag{2}$$

where C_2 and K_2 are stress indices for Class 1 piping products or joints (Table NB-3681(a)-1 [2]).

In equation (3.1), Z is the section modulus of pipe, defined in NC-3653.3 [2] as:

$$Z = \frac{2I}{D_0} \tag{3}$$

being I the moment of inertia of the pipe section. Lastly, the term M_A is the resultant moment loading on cross section due to weight and other sustained loads, calculated as:

$$M_A = \sqrt{M_{Ax}^2 + M_{Ay}^2 + M_{Az}^2} \tag{4}$$

where M_{AX}, M_{AY} and M_{AZ} have been previously calculated by the code by means of the linear elasticity theory. As to the thermally induced stress (namely the secondary stress, S_E), it is given by:

$$S_E = \frac{iM_C}{Z} \tag{5}$$

where M_C is the range of resultant moments due to thermal expansion. Hence, the total stress S_{TE} can be easily computed as:

$$S_{TE} = S_{SSL} + S_E \tag{6}$$

Once calculated the stress, the Rohr2 code allows defining the Utilization Factors (UF) as the ratio in between the calculated stress and the imposed stress limit. Specifically, for primary (i.e. self-sustained) stress the criterion to be met is:

$$UF_{S_{SL}} = \frac{S_{SL}}{1.5S_h} \leq 1 \tag{7}$$

where S_h is the maximum allowable stress defined for the given structural material in the ASME code. In a similar way, for the secondary (i.e. thermal-induced) stress, the criterion is defined as:

$$UF_{S_E} = \frac{S_E}{S_A} \leq 1 \tag{8}$$

where S_A is allowable cyclic range of equivalent stress for stresses due to thermal expansion, defined in the ASME code. Lastly, regarding the total stress, the pertinent criterion is defined as:

$$UF_{S_{TE}} = \frac{S_{TE}}{S_h + S_A} \leq 1 \tag{9}$$

2.3. The numerical model

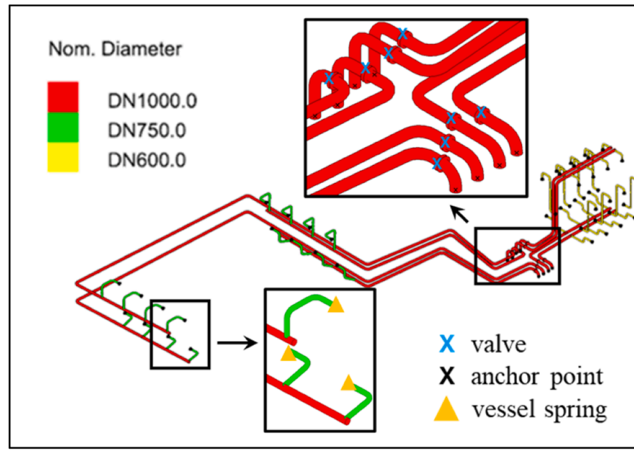

Fig. 3. The 1D model of the HCPB BB IHTS system.

The 1D numerical model has been set-up within the Rohr2 environment, purposely introducing the pipes geometric features such as starting point, end point, wall thickness and diameter. In Fig. 3, the so obtained model is shown. The SS 304 has been considered as structural

Table 1
Selected time steps for the pipe stress analysis.

Apros model time step [s]	Phase
1500	Final phase of the pulse
2200	Dwell
2400	Dwell
2500	Beginning of ramp-up
2600	Ramp-up
2650	Ramp-up
2700	Beginning of pulse
3200	Pulse at full operation

material for the entire piping system, adopting its thermal dependent properties. Regarding the piping insulation, a thickness of 150 mm has been assumed for all the pipes. An insulation material characterized by a density of 50 kg/m^3 has been adopted. As shown in Fig. 3, the presence of the valves, the tanks, the SGs and the IHXs has been simulated by the imposition of boundary conditions. Regarding valves, their weight the length has been estimated and applied to their locations. Instead, the tanks have been represented as “anchor points”, blocking displacement and rotation of the connected pipes because of their greater mass. Lastly, IHXs and SGs have been supposed to act as spring elements, namely flexible elements whose elastic behaviour depends on the geometric quantities (height, diameter, thickness) of the component which they

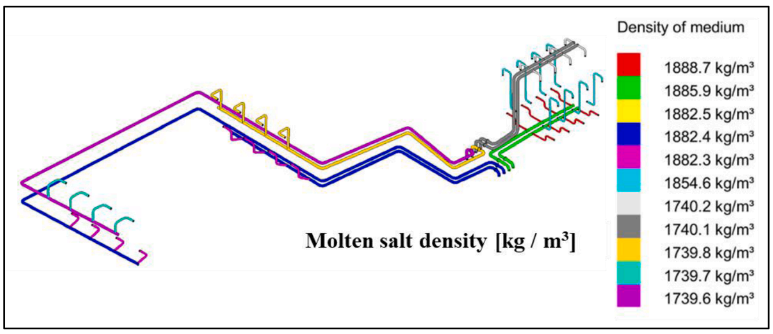

Fig. 4. Molten salt density at $t = 1500$ s.

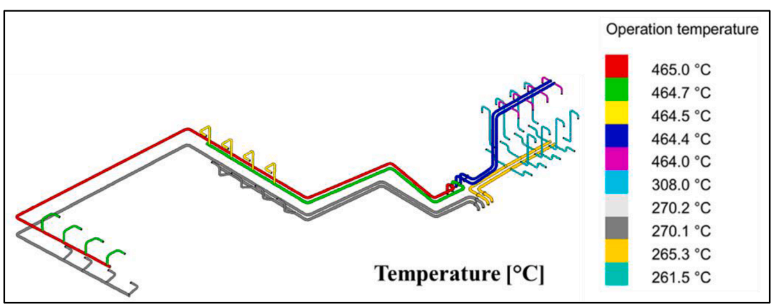

Fig. 5. Temperature at $t = 1500$ s.

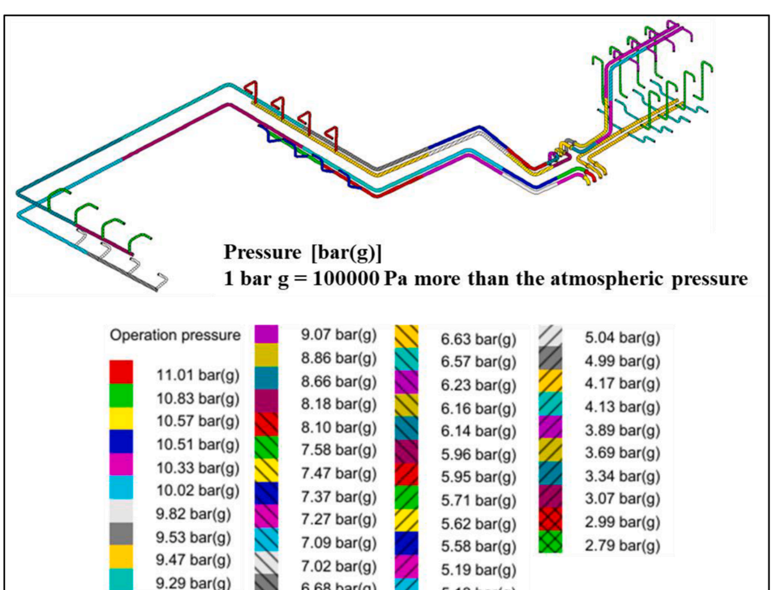

Fig. 6. Pressure at $t = 1500$ s (1 bar(g) = 100,000 Pa more than the atmospheric pressure).

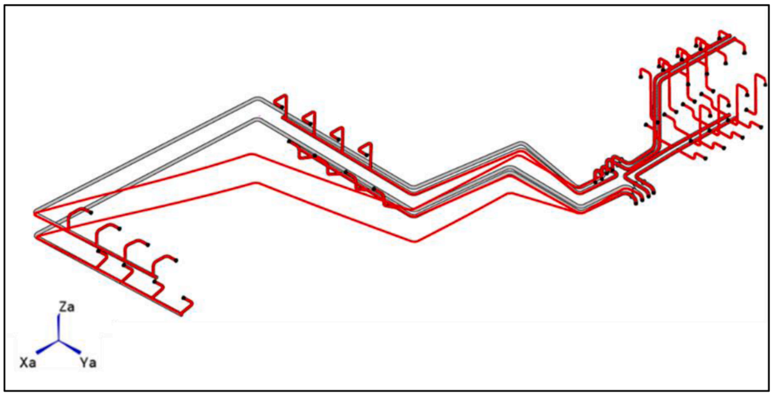

Fig. 7. Deformed (red) vs. undeformed (grey) layout for $t = 1500$ s – STE load case.

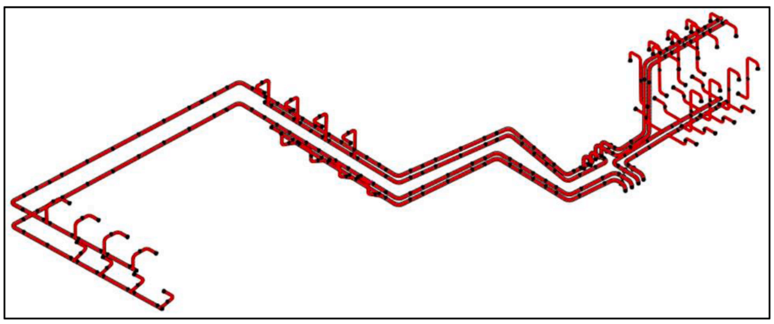

Fig. 8. Deformed (red) vs. undeformed (grey) layout for $t = 1500$ s – SSL load case.

stand for.

Regarding the loading conditions, the outcomes of a transient analysis of the integral HCPB BoP system using the Apros system code, performed in a separate study [7], have been used as input for the pipe stress analysis of the IHTS. This study has assessed a typical pulse-dwell cycle (lasting for 8000 s), allowing to select eight representative time steps (Table 1) for the pipe stress analysis.

For each of them, thermal and mechanical loads have been considered to set-up eight different loading scenarios and the corresponding pipe stress analysis. As an example, the assumed spatial distribution of temperature, pressure and molten salt density at $t = 1500$ s are reported in Fig. 4, Fig. 5 and Fig. 6.

2.4. The pipe stress analysis

From the analyses carried out, the eight scenarios show very similar outcomes both in terms of displacement values and trend of the deformed configuration. The time step characterized by the worst conditions is $t = 1500$ s, representing the end of the pulse phase. As an example, the deformed layout (in red) against the undeformed one (in grey) is reported in Fig. 7 for the STE (i.e. total loads) load case, clearly showing the necessity of a proper supporting system to compensate the predicted deformation while keeping stress under the limits prescribed by the code. To this purpose, a try-and-fail approach has been adopted to find a suitable support system under the loading conditions of the final phase of the pulse (i.e. $t = 1500$ s). In this regard, two types of support

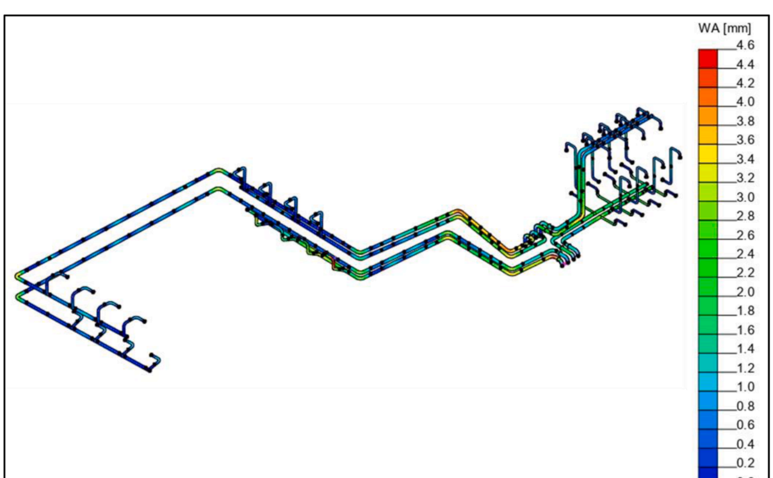

Fig. 9. Displacement for $t = 1500$ s – SSL load case.

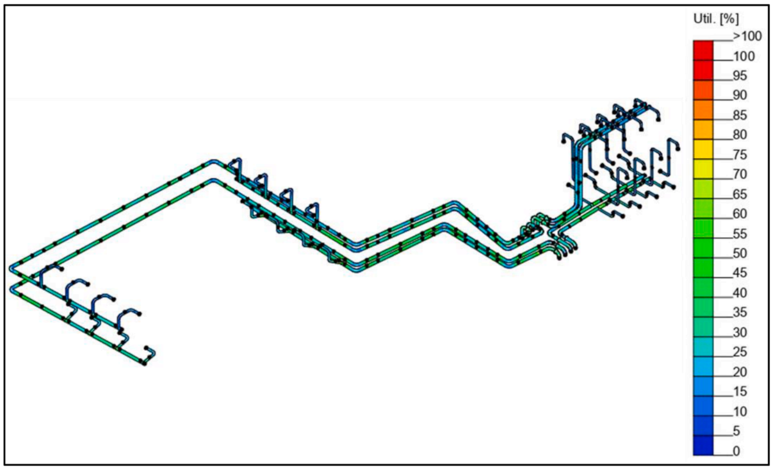

Fig. 10. UF for $t = 1500$ s – SSL load case.

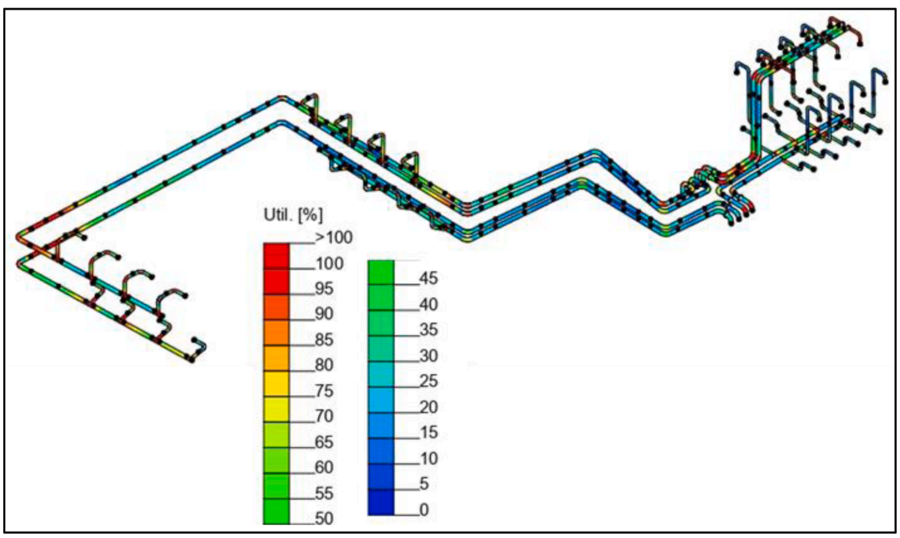

Fig. 11. UF for $t = 1500$ s – STE load case.

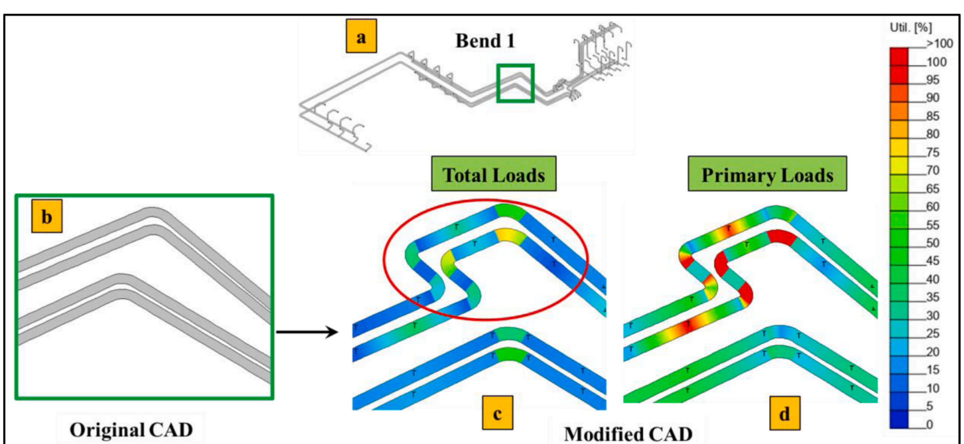

Fig. 12. Geometric modifications to bend region 1: (a) Bend 1 region in the green box of the original IHTS piping, (b) enlargement of the bend 1 region, (c) UF spatial distribution under Total loads of the modified bend 1 region, (d) UF spatial distribution under primary loads of the modified bend 1 region.

have been used: rigid hanger, supporting piping from above and preventing vertical displacement while mitigating those along the other axes, and rigid support, preventing displacement along a specific direction. Then, the try-and-fail approach has allowed determining a supporting system layout made of 53 rigid supports and 157 rigid hangers, reducing displacement (maximum displacement of 4.5 mm)

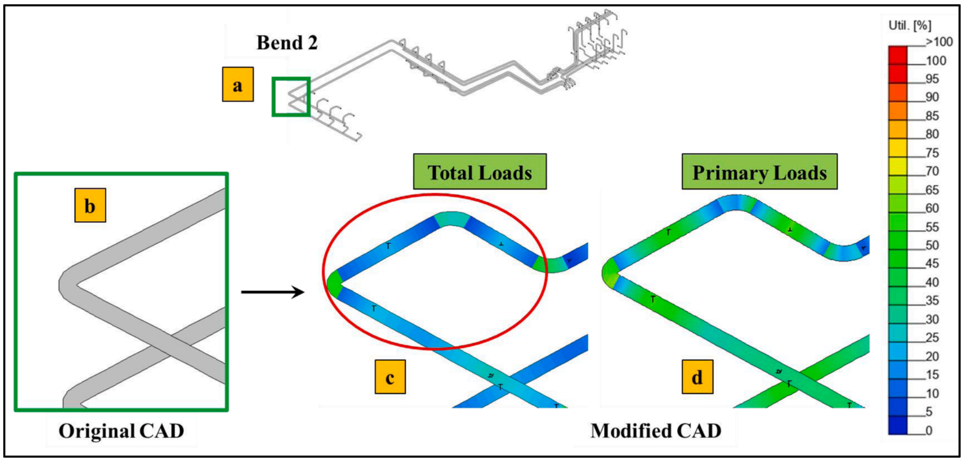

Fig. 13. Geometric modifications to bend region 2: (a) Bend 2 region in the green box of the original IHTS piping, (b) enlargement of the bend 2 region, (c) UF spatial distribution under Total loads of the modified bend 2 region, (d) UF spatial distribution under primary loads of the modified bend 2 region.

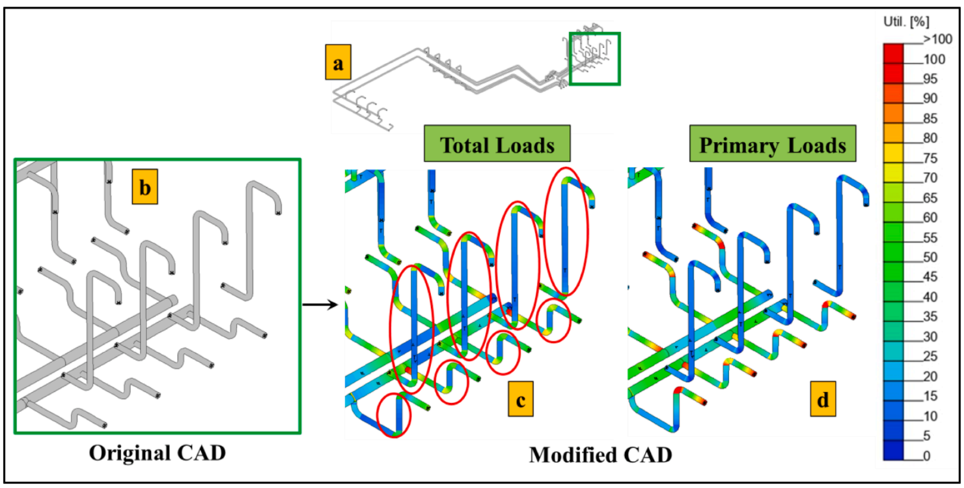

Fig. 14. Geometric modifications to SGs side: (a) SGs region in the green box of the original IHTS piping, (b) enlargement of the SGs region, (c) UF spatial distribution under Total loads of the modified SGs region, (d) UF spatial distribution under primary loads of the modified SGs region.

and keeping stress under the limit (maximum UF of 85.6 %) in the SSL load case (i.e. under primary loads), depicted in Fig. 8, Fig. 9 and Fig. 10, respectively.

Instead, looking at the secondary and total loads (namely the SE and STE load cases), the selected support systems do not allow to achieve UF values lower than 100 % and acceptable displacement. As an example, the UF spatial distribution attained for STE load case is depicted in Fig. 11. Here, one can observe that UF values greater than 100 % are predicted within several regions.

Then, a further campaign of analysis has been launched adopting a try-and-fail approach to modify the supporting system layout and to slightly change the piping geometry, in order to add more flexibility to the pipelines. A geometric modification has been proposed to two bend regions (called “bend 1″ and “bend 2″) of the IHTS circuit, depicted in Fig. 12 and Fig. 13. In particular, in both cases curves have been added to permit the thermal expansion of the pipes. In the “bend 1″ region the addition of the curves with the proper support allows the fulfilment of the structural design criteria under both primary and total loads, showing UF values between 20 % and 60 %. Instead, looking at the “bend 2” region, the design modification brought to an improvement of the structural response under the total loads, significantly worsening the system structural performances under primary loads. The same trend has been observed also performing geometric modifications in the other regions not fulfilling criteria under total loads, namely the IHXs and SGs branches. However, as can be observed in Fig. 14, despite modifications of the branches shape, in which the pipes have been extended by ~1.5 m, issues persist along these pipes, both in primary and total loads.

3. Discussion

The results of the pipe stress analysis reported above leads to the conclusion that a major design review of the whole IHTS is necessary, as it is too rigid to fully withstand the operational loads under while ensuring acceptable displacement. One possible solution could be the change from the current straight configuration of the pipeline to a circular/curved one, potentially contributing to the reduction of the total stress magnitude with simultaneous small displacement associated with the change of the SGs distributor routing that could improve the global structural response. Indeed, despite the introduced design modification and the different supports system layout assessed, a rethinking of the rationale behind the IHTS loop design appears mandatory. Therefore, in the following section, the major design review carried out downstream the presented analysis is presented.

3.1. The 3D model modification proposal

The performed stress analysis has been used as design base for the next steps in the 3D modelling phase; thus, the DEMO HCPB IHTS

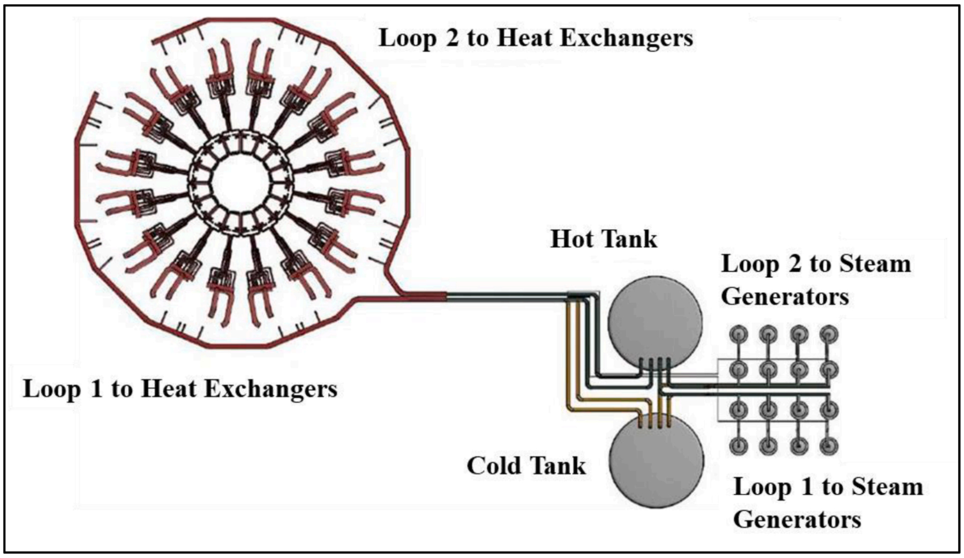

Fig. 15. Modification on internal building sections [8].

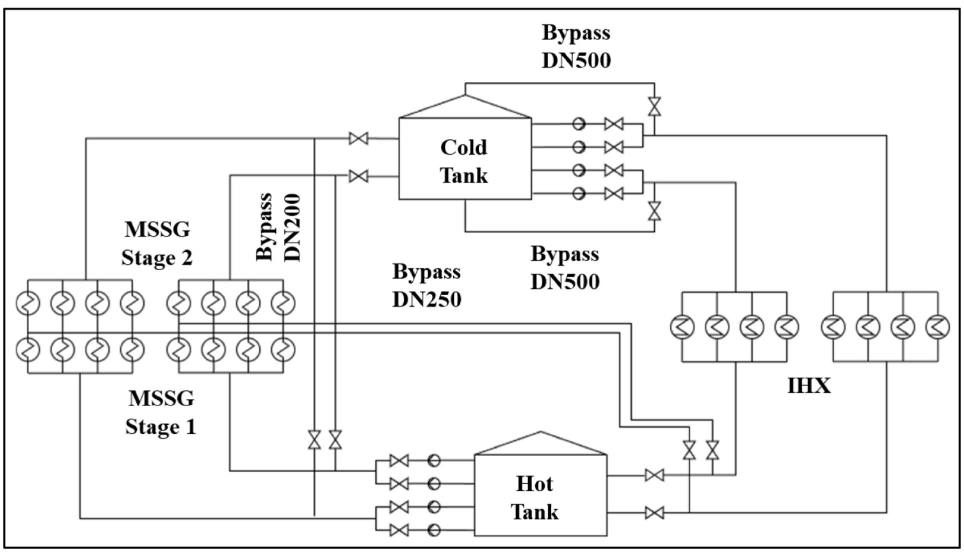

Fig. 16. IHTS and ESS process flow diagram.

geometric model underwent a comprehensive revision process that led to significant changes in the layout configuration. The first set of modifications, illustrated in Fig. 15, were focused on the straight pipe section located inside the building, which has been redesigned with a circular configuration, as suggested in the stress analysis above. Also, the Intermediate Heat Exchangers has been changed with a horizontal design.

In this initial configuration, no valves or bypass lines were included, and the system design was optimized thanks to analytical calculations and dynamic simulations [8]. The updated Process Flow Diagram (PFD) of the new layout is showed in Fig. 16. The main modifications introduced in the revised configuration address the optimisation of mass flow rates during the Pulse and Dwell operation phases.

The new layout foresees three dedicated bypass lines, all to be activated during the Dwell phase:

- a DN500 bypass line, to return the excess mass flow of HITEC molten salt to the cold storage tank;
- a DN250 bypass, to bypass the hot tank and route the molten salt directly to the second stage of the Molten Salt Steam Generators;
- a DN200 bypass line, intended to maintain the temperature level in the cold tank and link directly the hot and cold tanks.

In addition, the pump group was redesigned to enhance system reliability and introduce functional redundancy. The nominal diameter of the main pipeline was increased from DN900 to DN1000. In order to limit the pressure drop in the piping system, bends were modelled using the three times diameter rule. The model takes into account also the need of guarantee a slope for draining of minimum 2 %. To ensure consistency with the updated hydraulic and mechanical configuration, the 3D model was subsequently revised, as shown in Fig. 17 and Fig. 18.

Therefore, to take advantage of the outcomes of the previous thermal stress analyses, the new modelling iteration adopted a more targeted design approach, introducing expansion loops in sections where thermal expansion–induced displacements were found to be most severe. These sections were identified on the longest straight span between two consecutive bends. The total displacement strains caused by a temperature increase were calculated by means of the well-known equation:

$$y = \alpha U \Delta T \quad (10)$$

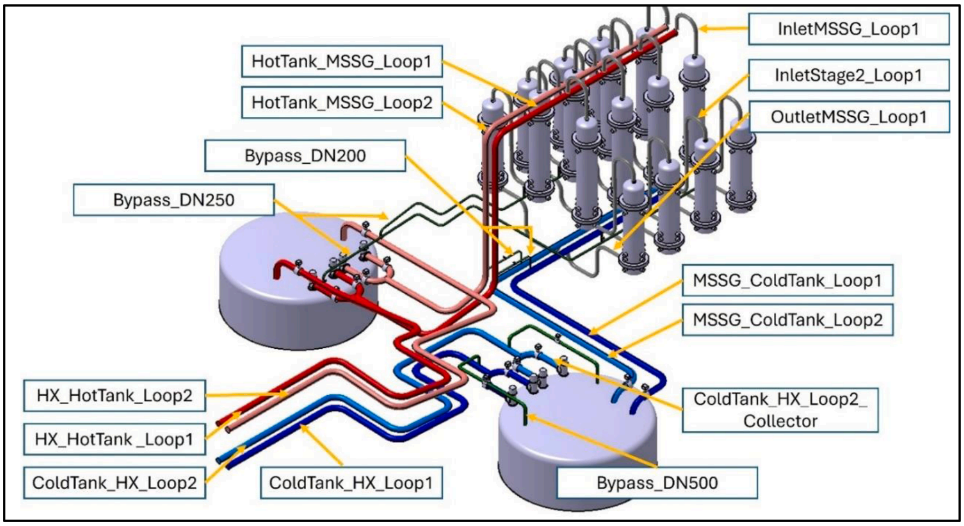

Fig. 17. Latter modification on IHTS and ESS.

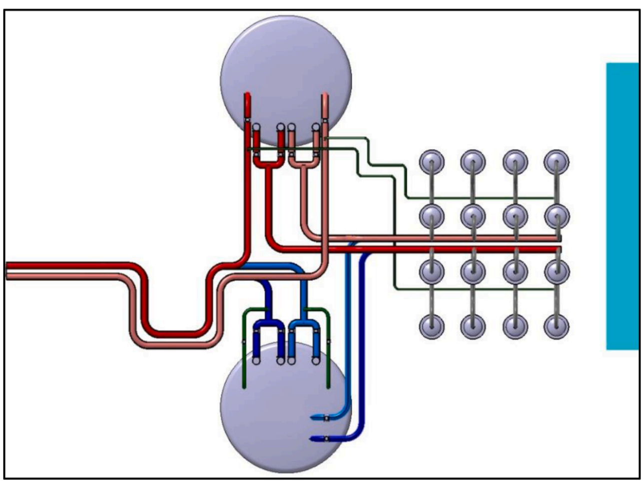

Fig. 18. IHTS and ESS top view.

So, the flexibility analysis was carried out in accordance with ASME B31.3 [9], which provides criteria for the correct sizing of expansion loops, including the determination of their minimum height (H). However, the same standard states that "no formal analysis of adequate flexibility is required for a piping system which is of uniform size, has no more than two points of fixation, no intermediate restraints, and falls within the limitations of empirical equation":

$$\frac{Dy}{(L-U)^2} \leq K_1 \tag{11}$$

$$K_1 = 208000\frac{S_A}{E_A} \tag{12}$$

$$S_A = f(1.25S_C + 0.25S_H) \tag{13}$$

Where:

- D is outside diameter of pipe [mm];
- E_A is reference modulus of elasticity at 21 °C;
- L is the developed length of piping between anchors [m];
- S_A is the allowable displacement stress range [MPa];
- U is the anchor distance, straight line between anchors [m];

Table 2
Expansion-loop main parameters.

	Unit	Value
S_C	MPa	138
S_H	MPa	116
f	-	1
S_a	MPa	201.5
E_a	MPa	195,000
K_1	$(mm/m)^2$	214.93
α	C^{-1}	1.2×10^{-5}
T_{env}	C	25
T_o	C	465
ΔT	K	440
U	mm	26,150
y	mm	138.07
H	m	13
W	m	12

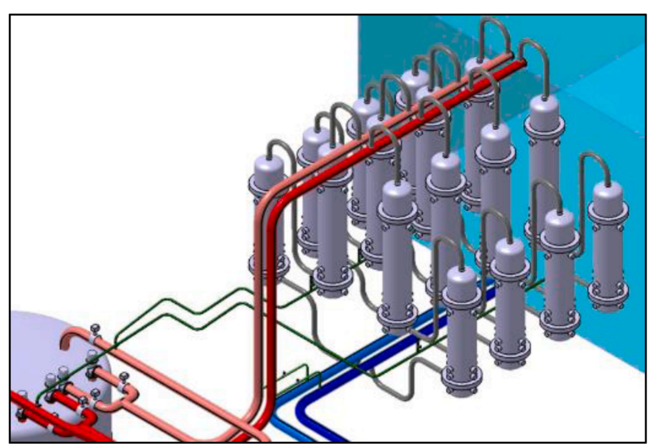

Fig. 19. Modification on MSSG piping.

- y is the resultant of total displacement strains to be absorbed by the piping system [mm];
- S_C is the basic allowable stress at minimum temperature;
- S_H is the basic allowable stress at maximum temperature;

The chosen configuration is the U-shaped expansion loop so that L = U + 2H This equation provides the physical basis for preliminary sizing

Table 3
Molten Salt Tanks main parameters.

	Description	Unit	Value
R_i	inner radius	mm	12,000
h	height	m	9
T	Temperature	°C	465
ρ	HITEC density (142 °C)	kg/m^3	2000
p_h	hydrostatic pressure	MPa	0.177
p_o	N2 blanket	MPa	0.02
P	Operating Pressure	MPa	0.197
P_d	Design Pressure	MPa	0.216
S_m	Design Stress intensity	MPa	130
t_m	minimum thickness	mm	20
CA	Corrosion Allowance	mm	6
t	Minimum Design thickness	mm	26
t_f	Manufacturing thickness	mm	28.575

of an expansion loop. The corresponding minimum height can be defined as:

$$H = \frac{1}{2}\sqrt{\frac{E_A}{S_A}\frac{D\alpha U\Delta T}{208000}} \tag{14}$$

In Table 2 the preliminary minimum sizes of the expansion-loop and input parameters are resumed. For these calculations the reference material has been changed and the chosen one for the piping system is the AISI 347H Stainless Steel, which has a higher resistance to corrosion caused by possible impurities of the molten salt used. The same material has been chosen also for the molten salt tanks.

Other modifications have been done to the inlet and outlet pipes of the Molten Salt Steam Generators (MSSG) (namely the SGs branches), as depicted in Fig. 19, where the pipe stress analysis has shown the need of use bended pipes so as to decrease the UF in those sections (Fig. 14). The modifications adopted to the piping connected with the hot and cold tanks and to the SGs branches could result in an improvement in the overall structural response. Therefore, further investigations with a new iteration of pipe stress analysis on the modified IHTS geometry are necessary. However, a preliminary sensitivity analysis suggested that, especially in the SGs branches regions, although the design modifications implemented may lead to an improvement in the structural response, they are not sufficient to fully resolve the highlighted issues. For this reason, it emerged that a profound revision of the system is necessary, requiring a deep change in the design philosophy in the steam generators regions.

According to the Molten Salt Tanks' dimensions provided in [8], a preliminary assessment of plates' thickness has been carried out. Tanks have been assumed aboveground, all-welded, single-shell, similar to those used for oil storage (API-650). ASME BPVC Sec III Division 1 Subsection NB [10] provides a formula for tentative pressure thickness of cylindrical shells, which consider the Design Pressure, Radius and Design Stress Intensity of the chosen material (AISI 347H). Further considerations have been carried out about the Design Pressure, which takes into account the hydrodynamic pressure of the HITEC salt, and the pressure of the cover gas (N2 at 0.02 MPa) [11] increased by a 10 % amplification factor. Calculations are resumed in Table 3.

4. The thermo-mechanical analysis of the main DEMO HCPB BOP ICD IHX

The thermal and structural performances of the main IHX of the DEMO HCPB BB ICD under the foreseen nominal thermal and mechanical steady state loading conditions have been investigated. To this purpose, 3D thermal and thermo-mechanical analysis of the IHX have been performed adopting the Ansys Workbench calculation suite, checking the obtained stress against the limits prescribed by the ASME code.

4.1. The main DEMO HCPB bop ICD IHX

The role of the IHX of the HCPB BB PHTS is to cool down the hot helium, coming from the BB at a temperature of ~520 °C, and provide it again to the BB sectors at ~300 °C. The IHX envisages parallel cooling concepts, providing redundancy and mitigate failures. The molten salt HITEC is used as coolant fluid within the IHX, whereas SS304, SS316 and SA508 steels have been used as structural materials. In Fig. 20 the geometric layout considered with its main dimension is depicted, whereas in Fig. 21a simplified operation principle of the IHX concept is reported [12].

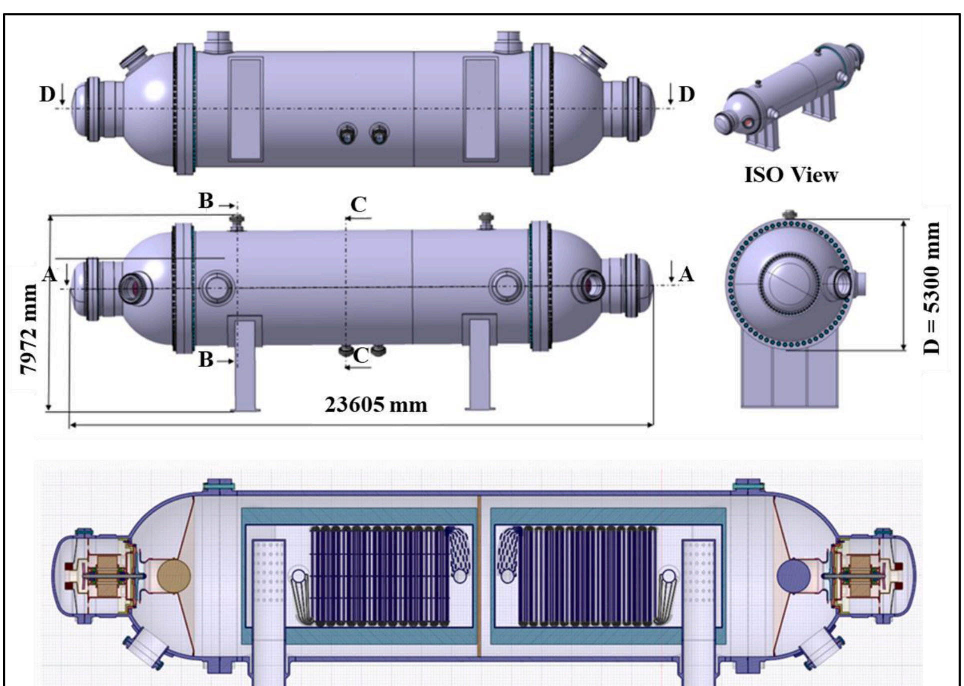

Fig. 20. The geometric model of the main DEMO HCPB BoP ICD IHX [12].

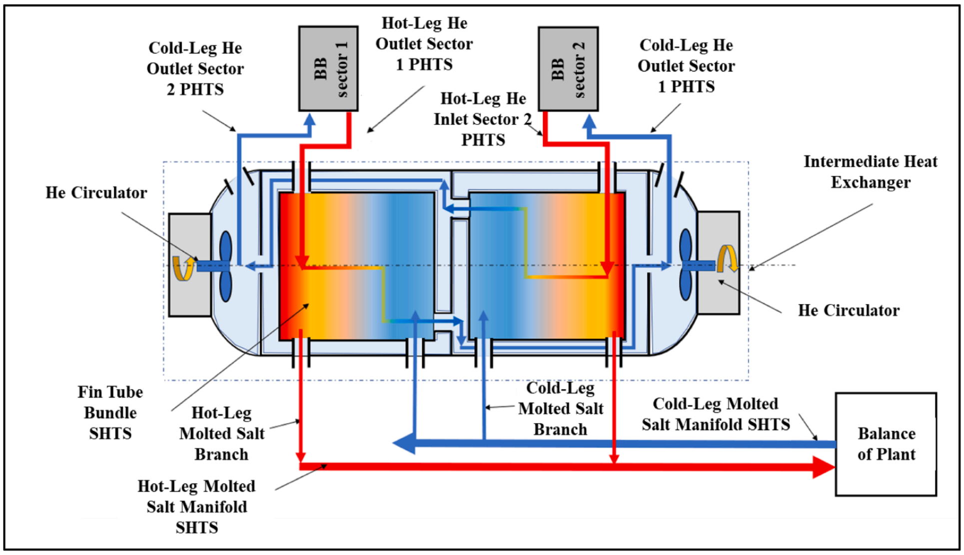

Fig. 21. Functional Flow Block diagram (FFBD) of the IHX [12].

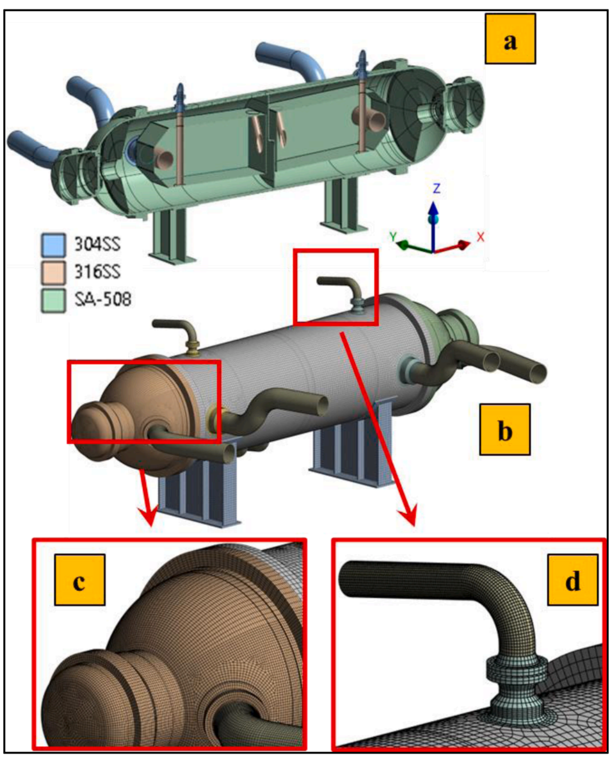

Fig. 22. IHX simplified geometry and mesh details: (a) section of the IHX geometry, (b) mesh grid set up for the IHX, (c) mesh detail of the IHX head, (d) mesh detail of the IHX pipe.

4.2. The adopted methodology

In order to assess the steady state thermal and structural performances of the main DEMO HCPB BoP ICD IHX under nominal loading conditions, a full 3D FEM model has been set-up. Given the complexity of the IHX geometry, some components have not been included in the model and their presence has been simulated by means of dedicated boundary conditions. The thermal analysis has allowed determining the temperature spatial distribution, assumed together with the mechanical loads to perform 3D structural analysis. The effect of realistic IHX supports has been considered too, in order to compute the stress field and to verify the fulfilment of the structural design criteria prescribed by the ASME BPVC - Section VIII DIV I.

4.3. The spatial discretization

With the aim of studying the thermal and structural behaviour of the IHX, the provided geometry (Fig. 20) has been deeply simplified to reduce the modelling effort. Specifically, geometric entities have been removed or simplified in those regions not playing any structural role. Nevertheless, their presence has been simulated by means of appropriate loads and boundary conditions. Then, a mesh composed of ~1.4 M nodes connected into ~460k hexahedral and wedge elements has been set up. A section view of the simplified geometry and some details of the mesh are shown in Fig. 22. With regard to the materials, thermal dependent thermo-physical properties have been assumed.

4.4. Thermal analysis and results

In order to carry out the thermal analysis of the IHX under the steady state nominal scenario, an appropriate set of loads and boundary conditions has been imposed such as:

- imposed temperatures, namely $T = 290$ °C (helium temperature before the compression [13]) onto the internal surfaces of the pressure vessel and the heads, Fig. 23(a), $T = 300$ °C (helium outlet temperature) onto the heads internal surfaces after the circulator and onto the internal surfaces of the helium outlet pipes [13], Fig. 23(b), $T = 520$ °C (helium inlet temperature) onto the helium inlet pipes internal and the external surfaces, Fig. 23(c), $T = 240$ °C onto the molten salt inlet pipes internal surfaces and $T = 470$ °C onto the molten salt outlet ones, Fig. 23(d);
- adiabatic condition onto the external surfaces of the internal liners and all the external surfaces of the IHX, Fig. 23(e) to simulate the presence of the insulation material;
- natural convection and radiation onto the saddle supports externals, with a bulk temperature (T_{bulk}) of 50 °C, a heat transfer coefficient (HTC) of 10 W/m^2 °C and emissivity (ε) of 0.3, Fig. 23(f). A temperature of 50 °C has been imposed to the saddle surfaces in contact with the ground;
- forced convection onto the molten salt pipes external surfaces, both inlet and outlet pipes, since they are in contact with the helium circulating inside the liner. To this purpose, T_{bulk} and HTC values indicated in Fig. 23(g) have been purposely calculated and imposed;

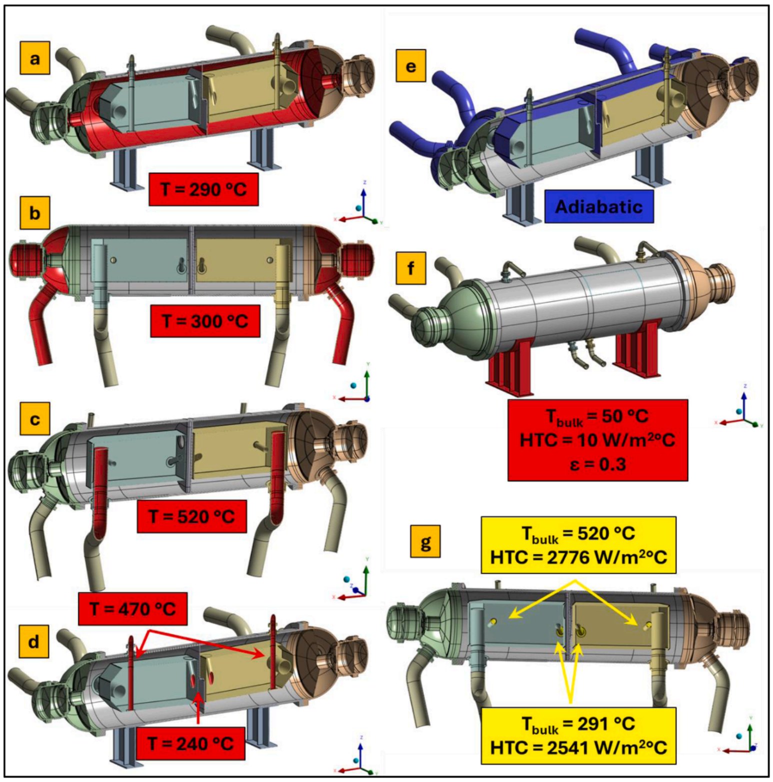

Fig. 23. Thermal loads and boundary conditions: (a) imposed temperature onto the internal IHX surfaces before circulator, (b) imposed temperature onto the internal IHX surfaces after circulator, (c) imposed temperature onto the helium inlet pipes, (d) imposed temperature onto the internal surfaces of the molten salt pipes, (e) adiabatic condition within the IHX external surfaces, (f) convection and radiation onto the external support surfaces, (g) convection onto the external surfaces of the molten salt pipes.

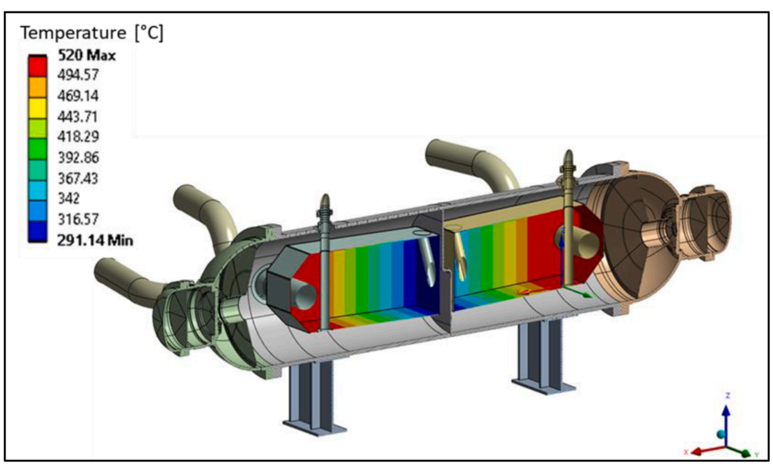

Fig. 24. Imposed temperature onto the liner inner surface.

- purposely calculated thermal field onto the helium wetted liner internal surfaces (Fig. 24);

Then, thermal analysis has been carried out and the arising 3D thermal field is reported in Fig. 25.

4.5. Mechanical analysis and results

In order to carry out the structural analysis of the IHX under the steady state nominal scenario, an appropriate set of loads and boundary conditions has been imposed such as:

- gravity acceleration and 3D temperature spatial distribution (Fig. 25) to all the components;
- a set of pressure loads, namely atmospheric pressure onto all the external surfaces, 9.20 MPa (helium design pressure) onto the helium-wetted surfaces after compression, Fig. 26(a), 8.89 MPa (obtained assuming a compression ratio of the helium circulator of 1.035 [13]) onto the helium-wetted surfaces before the compression, Fig. 26(b), and 0.80 MPa (molten salt nominal pressure) onto the internal surfaces of molten salt pipes, Fig. 26(c);
- the set of mechanical restraints shown in Fig. 26(d) and Fig. 26(e) have been imposed in order to simulate the presence of the piping support systems as well as the pipes continuity. One can observe that

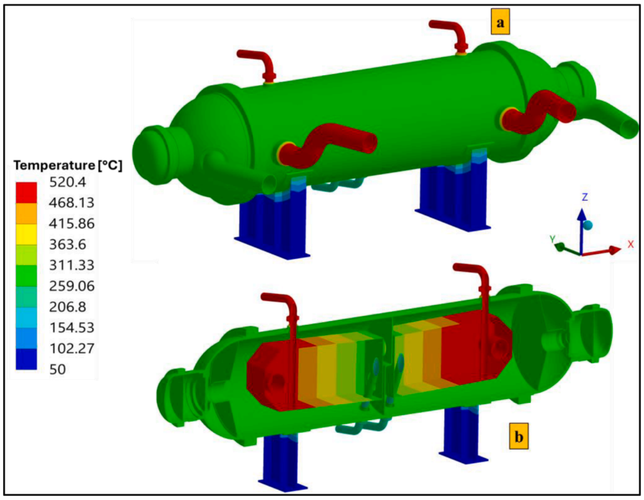

Fig. 25. IHX arising thermal field: (a) thermal field within IHX, (b) thermal field within a IHX section.

one saddle support is anchored to the floor (namely, all displacements are prevented) whereas the other is free to slide along the x direction [14], with an applied friction force calculated assuming a friction factor of 0.45 (lubricated steel to concrete [14,15]). In this regard, a maximum allowable displacement equal to ~ 32 mm has been calculated as a function of the vessel temperature and the distance between the saddles [14,15]. It is equal to the size of the slot allowing the saddle sliding.

Mechanical analysis has been launched and the structural behaviour of the IHX has been assessed. The displacement along the X direction is shown in Fig. 27. Looking at the sliding saddle, the maximum displacement is equal to ~ to 24 mm, well below the maximum allowable value (~ 32 mm).

The Von Mises stress field stress field arising within the IHX is reported in Fig. 28. As it can be observed, in those regions circled in red, stresses are very high due to the thermal field.

Moreover, a set of paths has been selected to check the fulfilment of the selected ASME criteria [16], reported in Table 4. Here, the prescribed criteria are reported as ratios between the stress intensity and the limit. Hence, values lower than one mean that the criterion is fulfilled.

To this purpose, the paths have been selected looking at the Von Mises Stress (VMS) over the maximum allowable stress (S_m), considered for the SA-508 steel [17] at the average temperature of the pressure vessel (290 °C). Paths within the pressure vessel thickness have been selected in those regions where VMS/S_m field is between 1 and 3, as depicted in Fig. 29 and Fig. 30.

The verification of the fulfilment of the ASME criteria is reported in Table 5, where no criticalities are observed.

4.6. Discussion

From the thermal point of view, the proposed design of the DEMO HCPB BoP ICD IHX seems not suffering of any particular issue due to the heat transfer conditions. The only critical regions are four short pipe regions (Fig. 28) that are not insulated. In these areas, a great temperature difference occurs between the external (290 °C) and the internal (470 °C and 520 °C for molten salt and helium pipes) sides. Therefore, extending the insulation could decrease the thermal rise so significantly reducing the associated stress amount. In any case, this has a very limited impact on the overall thermal performances of the assessed component.

From the mechanical standpoint, it is important to recall that the

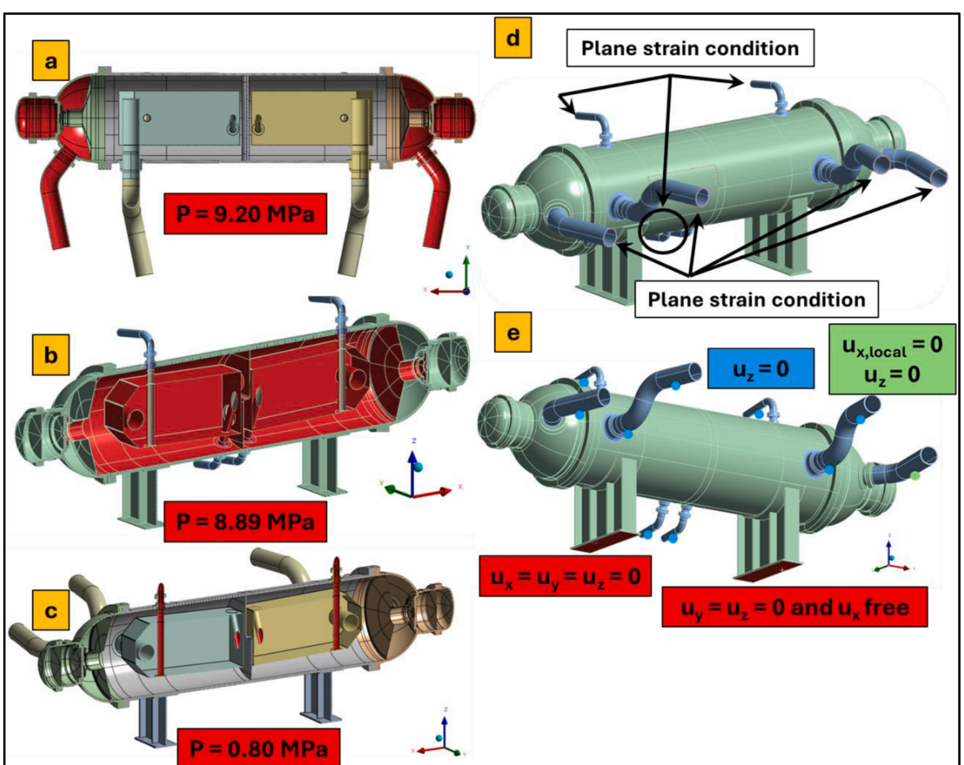

Fig. 26. Mechanical loads and boundary conditions: (a) imposed pressure onto the helium-wetted surfaces after circulator, (b) imposed pressure onto the helium-wetted surfaces before circulator, (c) imposed pressure onto molten salt internal surfaces, (d) plane strain condition within the pipes external surfaces, (e) imposed displacements to supports and selected nodes.

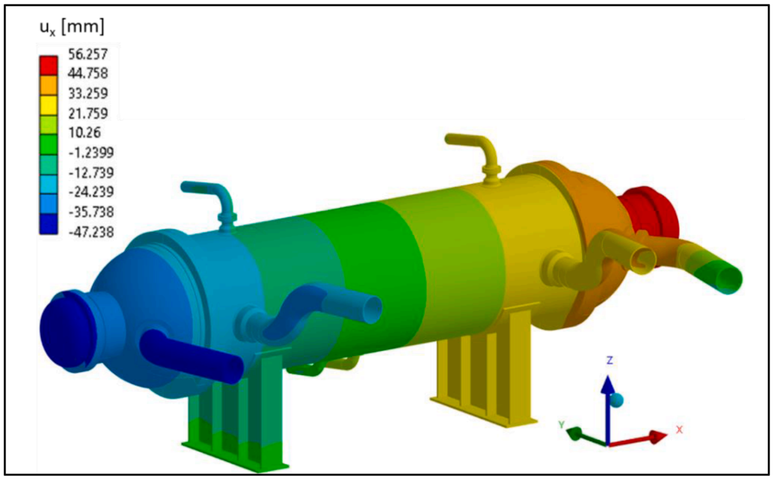

Fig. 27. IHX displacements along X direction.

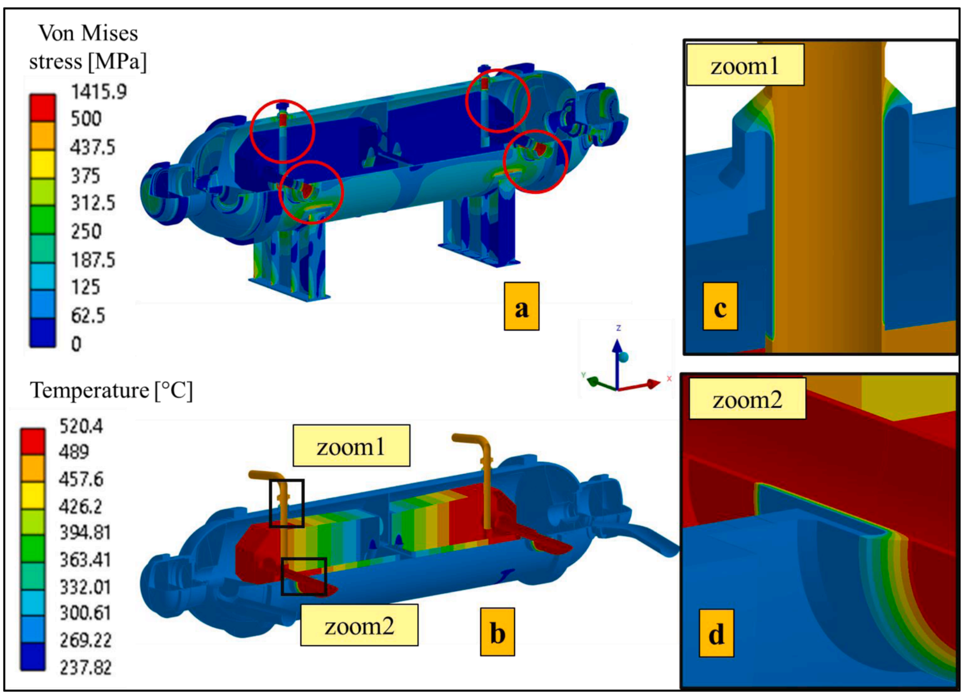

Fig. 28. IHX Von Mises stress field and details of critical regions: (a) Von Mises stress field within a section of the IHX highlighting critical regions, (b) thermal field within a section of the IHX, (c) detail of the thermal field within "zoom 1″, (d) detail of the thermal field within "zoom 2".

Table 4
Selected ASME criteria [16].

ASME Criteria		
$\frac{P_m}{S} \leq 1$	$\frac{P_m + P_b}{1.5S} \leq 1$	$\frac{(P+Q)_{tot}}{3S} \leq 1$

observed results strongly depend on the assumed piping support system, as it must be capable of reducing the induced displacements (and corresponding stress) to the IHX while maintaining reasonably low the stress level arising within the ground supports. However, assuming a reasonable piping support scheme, the obtained results show that all the selected ASME structural design criteria are fulfilled along all the selected paths with a remarkable margin against the stress limit. This demonstrates the soundness of the proposed IHX architecture.

5. Conclusion

The analysis herein presented has allowed the investigation of the structural performances of the DEMO HCPB BB IHTS under nominal loading conditions. Results allowed the determination of a viable supporting system layout capable of ensuring, under primary loads only, the fulfilment of the prescribed structural design criteria. Nevertheless, major design updates are necessary to obtain this result also to thermal induced stress. Mainly, the considered hydraulic loop shows excessive rigidity because of the long straight pipelines. A potential solution could be the adoption of circular or curve pipelines, so to give more intrinsic flexibility to the IHTS piping.

Then, the 3D thermal and structural analysis of the IHX allows showing its robustness under the envisaged nominal loading conditions. Only minor design modifications are necessary to improve its thermal and structural performances in some localised areas, extending the

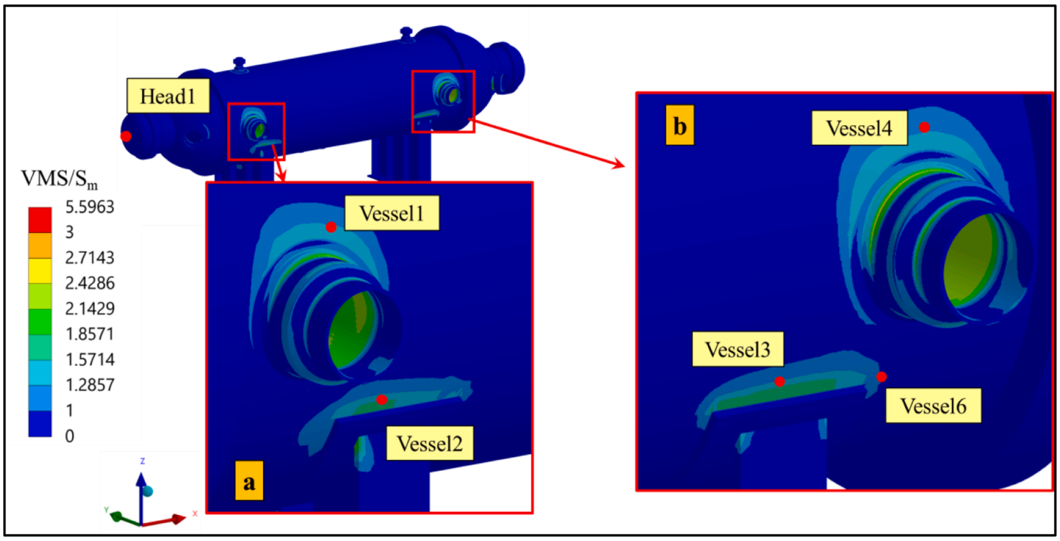

Fig. 29. Selected paths within the pressure vessel: (a) location of paths “Vessel1” and “Vessel2”, (b) location of paths “Vessel3”, “Vessel 4″ and “Vessel6”.

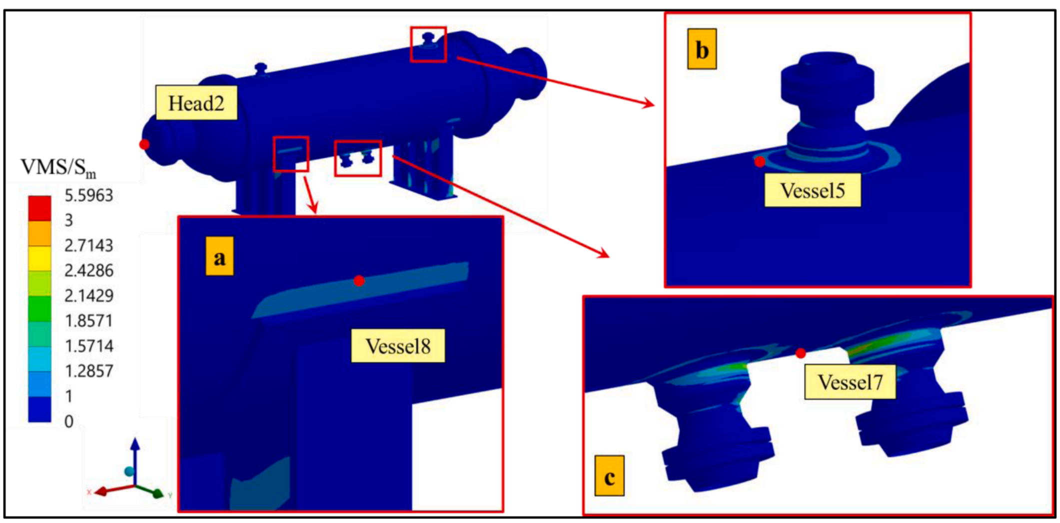

Fig. 30. Selected paths within the pressure vessel: (a) location of path “Vessel8”, (b) location of path “Vessel5”, (c) location of path “Vessel 7”.

Table 5
Verification of the ASME criteria.

ASME criteria verification			
Path	P_m/S	$(P_m+P_b)/1.5S$	$(P+Q)_{tot}/3S$
Vessel1	0.586	0.538	0.376
Vessel2	0.634	0.723	0.553
Vessel3	0.611	0.742	0.584
Vessel4	0.513	0.567	0.387
Vessel5	0.682	0.474	0.287
Vessel6	0.519	0.529	0.325
Vessel7	0.632	0.495	0.308
Vessel8	0.462	0.362	0.364
Head1	0.683	0.610	0.306
Head2	0.679	0.606	0.304

insulation layer to cover some regions presently showing high thermal gradients through small thicknesses.

Globally, this work represents a contribution to the ongoing development of the HCPB BoP design, providing insights that can support the design evolution from the pre-conceptual to the conceptual stage.

CRediT authorship contribution statement

I. Catanzaro: Writing – review & editing, Writing – original draft, Visualization, Validation, Methodology, Investigation, Formal analysis, Conceptualization. **G. Bongiovì:** Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Software, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **E. Vallone:** Writing – review & editing, Writing – original draft, Visualization, Supervision, Methodology, Investigation, Conceptualization. **I. Moscato:** Writing – review & editing, Visualization, Validation, Supervision, Investigation, Conceptualization. **A. Tincani:** Writing – review & editing, Visualization, Methodology, Investigation, Conceptualization. **S. Perez-Martin:** Writing – review & editing, Visualization, Validation, Investigation, Conceptualization. **A. Quartararo:** Writing – review & editing, Visualization, Investigation. **P. Chiovaro:** Writing – review & editing, Validation, Conceptualization. **P.A. Di Maio:** Writing – review & editing, Visualization, Supervision, Project administration, Funding acquisition, Conceptualization. **S. Basile:** Writing – review & editing, Visualization, Conceptualization. **A. Tarallo:** Writing – review & editing, Writing – original draft, Visualization, Investigation. **C. Rossetti:** Writing – review & editing, Writing – original draft, Visualization, Formal analysis. **J. Bajari:** Writing – review & editing, Visualization, Investigation, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

No data was used for the research described in the article.

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