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The Design Enhancement of a Pneumatic Cylinder for Valve Actuators by Integrating the "Design-by-Formulas" and "Design-by-Analysis" Approaches with Multiple Integrative Advanced Simulations †

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Abstract: The authors would like to make a meaningful contribution to product development strategies by introducing an original holistic design methodology for pressure-containing equipment. A case study is presented, focusing on the design enhancement of a pneumatic cylinder for valve actuators. The purpose of the discussed methodology is supplying designers with a full picture of the product, usually not directly available, thanks to the extensive use of advanced Finite Element Analysis (FEA) to increase confidence in the safety and reliability of the design at the virtual prototype stage while providing useful insights that can be used during product life in occasions such as root cause analyses. The new design approach is applied to the cylinder by analyzing it in special conditions like during a pressure burst test using an FEA. A comparison with the experimental results confirms that the proposed methodology can potentially contribute to introducing renewed, safe, reliable, and lighter components, reducing the cost and time of development.

Keywords: valve actuator; pneumatic cylinder; FEA; design-by-analysis; EN13445; burst pressure test

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1. Introduction

The availability of products that are increasingly safe and reliable when subjected to the most demanding applications is a top priority for industries where pressure-containing equipment plays an important role, such as oil and gas plants with valves operated by fluid actuators. This is the context in which this study would like to provide a contribution by introducing a new design methodology, using benefits from integrative advanced simulations. This contribution is part of other investigations conducted to introduce new cross-functional product development processes, exploiting in a holistic homogeneous approach the benefits of value engineering techniques, the potential of simulations, and the increase in productivity fostered by a positive work environment benefitting from teamwork and the exchange of know-how [1]. The benefits of an advanced FEA are used for integrating the design process, contributing to introducing economically efficient products in the market, offering customers the greatest possible value [2], and ensuring these products' survival in highly competitive sectors thanks to these fundamental requirements. From a literature analysis, it emerged that common design techniques for pressure-containing equipment are mainly based on "design-by-formulas" (DBF) and "design-by-analysis" (DBA) techniques [3],

which are performed according to procedures provided by pressure vessel standards [4]. The last developments of DBA foresee the use of a non-linear FEA in structural verification methods called the "direct route" [5], but the potential of other advanced simulations such as explicit dynamics or fatigue life simulations were not exploited. The proposed design methodology features the original contribution to combine existing procedures based on DBF and DBA techniques with an integrative advanced FEA, such as explicit dynamics or response spectrum analyses, to provide complete knowledge of the product design. Its main steps are summarized in the comparative flow charts in Figure 1, and its application to the case study is analyzed.

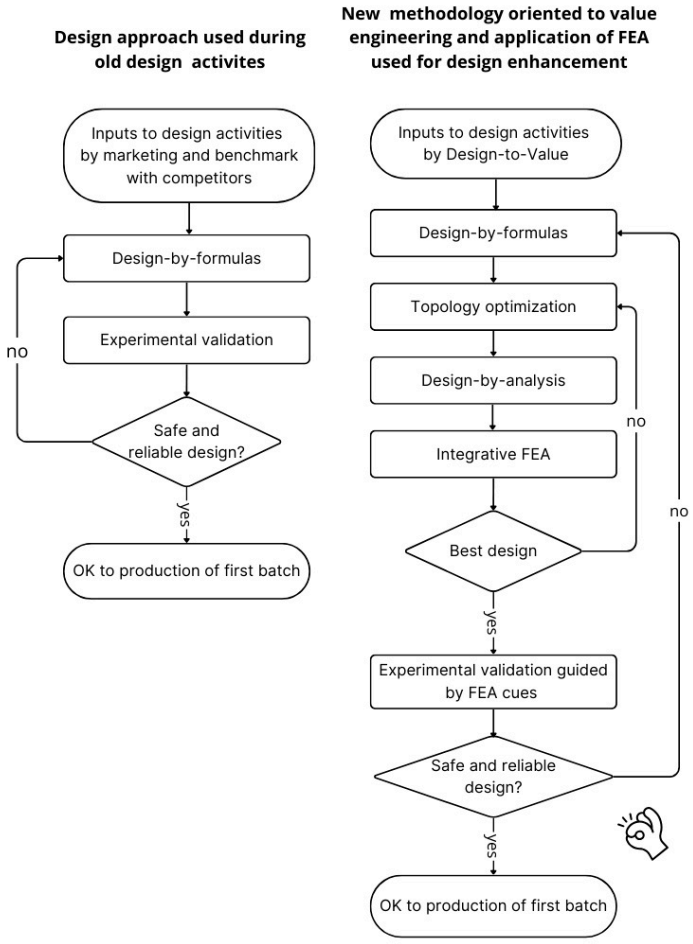


Figure 1. Flow charts of the cylinder old design process used in the past and of the new one based on the proposed methodology.

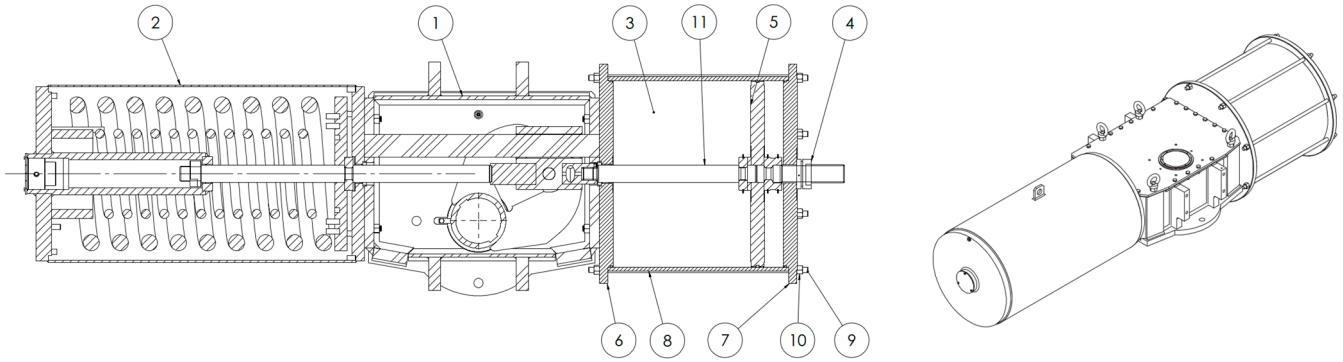
The Purpose of the Methodology

The peculiarity of the methodology presented here is the effort of supplying designers with information that is not often available concerning a broad picture of a product. Often during design activities, questions arise regarding the behavior of the equipment in special situations, for example, "how much could the supply pressure increase (by accident) until the equipment would reach failure, and how would it fail?" The answer is very useful for confirming the safety of the product and is usually obtained indirectly, by individually studying the single components' behavior or by performing expensive tests. This approach can be improved by introducing integrative advanced FEA, which are currently more accessible. To answer the previous question, it is possible to perform an explicit dynamics simulation that provides a complete picture of the failure process. In this way, a higher level of understanding of the design is achieved, with information provided on its behavior

during extreme events or misuse. The approach of the new proposed methodology is affordable in terms of resources and aims to provide designers with original insights, which are useful for obtaining a deep understanding of the product behavior. A working method valorizing teamwork and the exchange of mutual know-how is an additional element further fostering the success of the process [6].

2. The Study of the Cylinder with an Initial "Design-by-Formulas" Approach

The pneumatic cylinder is coupled with a scotch yoke modular actuator with an architecture very common in the industry, shown in Figure 2. The actuator converts the linear movements of a piston (5), which is fed by a compressed fluid and connected to a stem (11), into the angular motion of a scotch yoke mechanism in actuator central body (1). The yoke is coupled with a joint to the stem of a valve with a 90° stroke, such as a ball or butterfly valve. Two cylindrical flat ends, called "head" (6) and "tail" (7) blind flanges, are joined to a cylindrical tube (8) by means of long threaded tie rods (9) and nuts (10).



(**a**) (**b**)

Figure 2. (**a**) Sectional view of scotch yoke actuator: (1) central body; (2) spring cartridge performing "fail safe" action; (3) cylinder; (4) stopper bolt with fixing and sealing systems; (**b**) 3D view of actuator.

2.1. Detailed Objectives of the Design Enhancement Plan for the Cylinder

The first purpose of the design enhancement is to perform a series of interventions to improve the lateral flat ends by applying the DBF approach and then the DBA techniques foreseen by EN13445-3 [7], for achieving a deeper verification of load cases not covered by formulas. A second purpose is to verify if improvements of tube and tie rods are possible, maintaining compliance to EN13445-3. Once the achievable improvements are implemented, the final purpose of the design enhancement is to achieve a high degree of confidence on the safety and reliability of the new design from the virtual prototype stage, by analyzing the results of integrative verifications based on the advanced FEA, providing extremely useful information, such as the map of the fatigue life of the components or a description of their behavior in the occasion of an earthquake.

2.2. The Sources for Formulas

The first verifications for ensuring the safety and reliability of the cylinder components rely on the issue of an Excel electronic spreadsheet with formulas for structural verifications. This phase, on which the old design of the cylinder was entirely based, is now a first step for making initial choices on minimum dimensions and thicknesses, which will be refined thanks to simulations in the following steps of the design process. The safety of the actuator against the risks of pressure is an absolute priority, and for this reason, the main product standards, such as EN15714-3, ISO 12490 [8], or API6DX [9], require that pressure-containing parts are built in compliance with an international standard for

pressure vessels, such as ASME BPVC Sec. VIII Div. 1 [10] or EN13445-3; this last one chosen as a reference for the investigations of this case study. EN13445-3 contemplates the use of "design-by-analysis" as an integration or as an alternative to "design-by-formulas", and this opportunity has been fully exploited, in line with the philosophy of the newly proposed methodology. For pressure load, the formulas made available by EN13445-3 have been followed, but the same parts retaining pressure towards the external environment (tube, lateral closing flat ends, tie rods, and nuts) can simultaneously be subjected to loads of a different nature, for which no suitable formulas are available (for example, a concentrated load on a tail flange due to a stopper bolt). For these situations, after a preliminary check with the spreadsheet, by means of structural verifications based on formulas from Roark's Manual [11], the main verifications have been performed with EN13445-3 DBA procedures.

2.3. Design-by-Formulas of Flat Ends

For determining the minimum thickness of the side flat ends when they are subject to pressure load only, they have been schematized as "Flat ends with a narrow-face gasket", following the rules of paragraph 10.5.2 of EN13445-3. Figure 3 shows the tail flat end characteristic thicknesses. The minimum flanged extension e_1 is found by Formulas (1)–(4), with the following meaning of the main parameters: e_A, e_{p1}, and e_{pt1} are the minimum thicknesses required in assembly, operating, and testing conditions; C is the tie rod pitch circle; C_F is a bolt pitch correction factor; G is the gasket reaction diameter; W is the design bolt load; P and P_t are the design and test pressures; f, f_A and f_{test} are the material design stresses at ambient, calculation temperature, and testing conditions. A gasket factor m should appear in the formulas but is set equal to 0 in the case of the use of O-rings. The analyzed cylinder has an internal diameter of 885 mm and a design pressure of 12 barg. The flat ends are made of P355NH EN10083-3, and the maximum design temperature is 100 °C. An iterative process starting from a first-attempt thickness allowed the identification of the following minimum necessary thickness of the flanged extension e_1.

$$e_1 = max\{e_A; e_{P1}; e_{Pt1}\} = 18 \text{ mm} \quad \text{where} \tag{1}$$

$$e_A = \sqrt{C_F \frac{3(C-G)}{\pi G}\left(\frac{W}{f_A}\right)} = 18 \text{ mm} \tag{2}$$

$$e_{P1} = \sqrt{3C_F\left(\frac{G}{4}\right)(C-G)\left(\frac{P}{f}\right)} = 17 \text{ mm} \tag{3}$$

$$e_{Pt1} = \sqrt{3C_F\left(\frac{G}{4}\right)(C-G)\left(\frac{P_t}{f_{test}}\right)} = 16.5 \text{ mm} \tag{4}$$

The minimum value for the central thickness e (ref. Figure 3) is identified by Formulas (2) and (5)–(7), where ν is the material's Poisson's ratio.

$$e = max\{e_A; e_P; e_{Pt}\} = 43 \text{ mm} \quad \text{where} \tag{5}$$

$$e_P = \sqrt{\left[\frac{3(3+\nu)}{32}G^2 + 3C_F\left(\frac{G}{4}\right)(C-G)\right]\left(\frac{P}{f}\right)} = 43 \text{ mm} \tag{6}$$

$$e_{Pt} = \sqrt{\left[\frac{3(3+\nu)}{32}G^2 + 3C_F\left(\frac{G}{4}\right)(C-G)\right]\left(\frac{P_t}{f_{test}}\right)} = 41 \text{ mm} \tag{7}$$

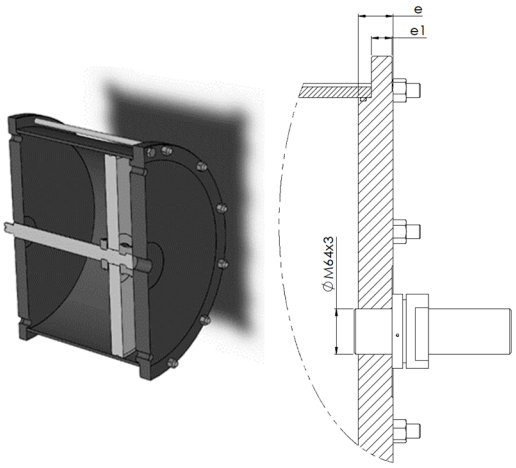


Figure 3. Three-dimensional view of the cylinder and sectional view of the "tail" flat end.

The "tail" flat end is provided of a threaded hole for an angular stroke adjustment stopper bolt represented in Figure 3 and another one for the pneumatic supply, both requiring a verification for the need of a reinforcement thickening, according to par. 10.6 of EN13445-3, by means of the identification of factors of thickness increment, called $Y_{2,}$ and depending on hole number, size, and distance. The maximum identified amplification factor Y_{2MAX} concerns the distance between the two holes and brings to the required total thickness found with Formula (8).

$$e = Y_{2MAX}\, e_0 = 1.09 \cdot 43 = 47 \text{ mm} \quad (8)$$

e only considers pressure loads, but a concentrated load due to the action of the stopper bolt is present. As already clarified, dedicated formulas are not available, and for this reason the DBA tool provided by EN13445-3 is an effective solution, presented in next paragraphs. As anticipated, for analyzing what has been done in the past and for having a first term of comparison with the FEA results, an analysis of the effects of the concentrated load based on Roark's manual was performed, referring to the formulas in par. 11.2, "Formulas for Flat Circular Plates of Constant Thickness". Many schematizations present in the manual were analyzed to identify which Roark's case best fit the real situation shown in Figure 3. Numerical values will be presented during the comparison with the DBA results. Finally, in the last part of this study, the validity of the results obtained with formulas and the FEA have been validated with experimental tests. For understanding the final chosen thickness of the flat ends, it is necessary to consider that they must ensure sufficient stiffness, avoiding excessive deformations and showing stress levels meeting the acceptability criteria foreseen by EN13445-3 DBA Annexes when subject to all the loads. To satisfy these requirements, for example the chosen thickness for the "tail" plate is equal to 70 mm instead of the 47 mm due to the pressure load only. In the old design, it was equal to 80 mm, with a significant weight reduction. The example here presented focuses on the "tail" flat end, but considerations are valid for the "head" flat end also, subject to similar loads.

2.4. "Design-by-Formulas" of Tube and Tie Rods

The approach used in the occasion of the old design activities for sizing the tube and the tie rods has been analyzed for verifying whether by using the DBA it is possible to improve their design and increase their weight efficiency. Investigations confirmed that the best achievable design is already present, allowing to comply with EN13445 requirements and to respect important construction constraints. The cylinder tube verifications are based on formulas of Chapter. 7 of EN13455-3 "Shells under internal pressure", considering a

compression load due to the pre-tensioning of the tie rods and the action of a uniform internal pressure. The tube is made from a rolled and welded sheet of P355NH EN10028-3. The minimum thickness found with formulas (EN13445-3:2021, eq. 7.4.2) for complying with the required safety factors is low, equal to 4 mm, but a piston must slide without jamming along its inner surface, requiring a high concentricity tolerance and the absence of ovalization. To comply with these additional constraints, a thickness of 12.5 mm is chosen, based on proven in use production practices, and it is then not possible to further reduce it. The tie rods are verified by following par. 11.4 and 11.5 of EN13445-3. The chosen configuration, based on 12 tie rods of dimension M24 made in ASME/ASTM SA-A193 Grade b7, provides a total resistant area very close to the minimum necessary one to respect the required safety factors. Further improvements are then not possible.

3. "Design-by-Analysis" of Flat Ends

In a second step after DBF, the second extremely effective method used to make the final choices about the cylinder flat ends design is based on the DBA routes of EN13445-3 [12], allowing for an examination of complex geometries and load conditions deviating from common ones considered in formulas. Two strategies are provided by the standard for performing the DBA [13]: the "Direct Route" (Annex B of EN13445-3) and the "Stress Categorization Route" (Annex C). The method based on stress categories uses linear elastic analyses and involves a detailed procedure where several sections of the component presenting stress concentrations are analyzed. Linear paths crossing them are chosen and the total stress in each point is divided into stress categories ("global or local range", "membrane", "bending", and "peak") with maximum limits not to be exceeded [14]. The "direct route" method allows for the use of a non-linear FEA [15], associated with criteria for evaluating the acceptability of the results. The "Direct route" approach is applied here for the design enhancement of the cylinder flat ends by checking for the absence of ductile ruptures and excessive deformations.

3.1. Application of EN13445-3 Annex B "DBA—Direct Route"

The "Direct route" takes into consideration different types of actions that are imposed on the structure causing its stresses: pressures, forces, forced displacements, and temperature variations. The imposed actions can be classified according to their variation in time as permanent, variable, exceptional, operating pressures, and temperatures. The "tail" flat end is subject to operating pressures and a variable action produced by the concentrated load of the stopper bolt. Their design values are increased during the analyses by introducing required partial safety factors. In the "Direct route" procedure, the proof of strength is provided by a series of "checks" that must be fulfilled [16]. Each one has been analyzed by assessing its applicability:

Design check for gross plastic deformation: this check has been performed to verify the behavior of the flat end when subjected to its maximum allowable loads, consisting of the test pressure (1.5 times the design pressure) and of the maximum stopper bolt load.

Design check for fatigue—this check has been performed by integrating the FEA and the design-by-formulas, as exposed in following paragraphs.

Design checks for progressive plastic deformation, instability, and static equilibrium: these checks were deemed as unnecessary due to the range of operative pressures, operating temperatures, dimensions and geometry of the cylinder [17].

3.2. Implementation of the Design Check for Gross Plastic Deformation

The FEA model was set up by following the assumptions of EN13445-3, Annex B, par. B.8.2 "Gross Plastic Deformation (GPD)". Material models with a linear elastic ideal

plastic constitutive law were initially used and, for a result comparison, the analyses were repeated with linear elastic linear plastic material models also. The design strength parameter of each material is defined by dividing its yield strength by the required partial safety factor. Finite elements of the first order were used as required by the procedure, but the analyses were repeated by using second order elements also for a comparison of the quality of results. Annex B indicates the use of Tresca's yield criterion as a first option for the assessment of stress distribution. The von Mises criterion is also allowed, but the design strength of the materials must be decreased by a sqrt(3)/2 factor, equal to a reduction of 15.5%, and this approach was chosen. The FEA model evaluated the effect of the maximum pressure and of the most severe concentrated load case, occurring when the design pneumatic load acting on the piston is transferred to the stopper bolt. The FEA results, shown in Figure 4, confirm that the components are verified, as the maximum deformation is lower than the admissible one of Annex B par. B.8.2.1, equal to 5%. An alternative "application rule" was also applied, as suggested by par. B.5.1.1, for having a comparative evaluation criterion of the acceptability of the results, by assessing them with criteria taken from "Stress Categorization Method" of Annex C also. The analysis of the results confirmed that primary, secondary, and peak stresses respect the maximum allowable values of Table C-3 of Annex C. Figure 4 highlights how the FEA performed with this method allow for the precise identification of the areas with high stress concentrations, occurring in correspondence of the threads of the stopper bolt hole, with "peak stresses" lower than three times the design stresses "*f*" and then acceptable.

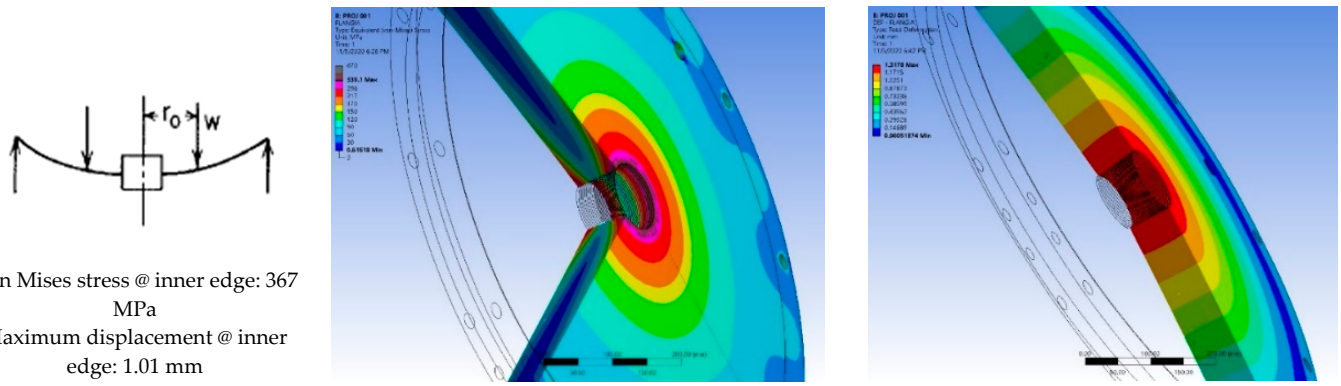


Figure 4. Comparison between formulas and FEA for the verification of a tail flat end subjected to a concentrated load: (**a**) results of formulas of case 1b of Table 11.2 of Chap. 11 of Roark's Manual, "Annular plate, uniform annular load at a radius r0; outer edge simply supported"; (**b**) peak stress values found with DBA "Direct Route"; (**c**) maximum displacement from same FEA model.

3.3. Topology Optimization of the Tail Flat End

In the context of the newly proposed methodology, a first application of the integrative advanced FEA consisted of an optimization study of the geometry of the "tail" flat end by means of a "topology optimization" FEA, performed with the dedicated tool present in Ansys® Mechanical vers. 19.2. The output of the analysis is a 3D model, shown in Figure 5, featuring a more efficient geometry, with an optimal distribution of the thicknesses for respecting the requirements of deformation and admissible stresses, while at the same time minimizing the component weight. The model allows to identify with visual impact the regions requiring a higher contribution of material, being the most stressed ones. The thickest region is the central one, receiving the concentrated load from the stopper bolt. The load is then distributed radially, through a series of ribs, towards the external region,

where the average thickness is lower. The study provides useful insights for shaping the geometry in case, in the future, production techniques such as casting are adopted, instead of the current one based on metal sheet cutting and milling.

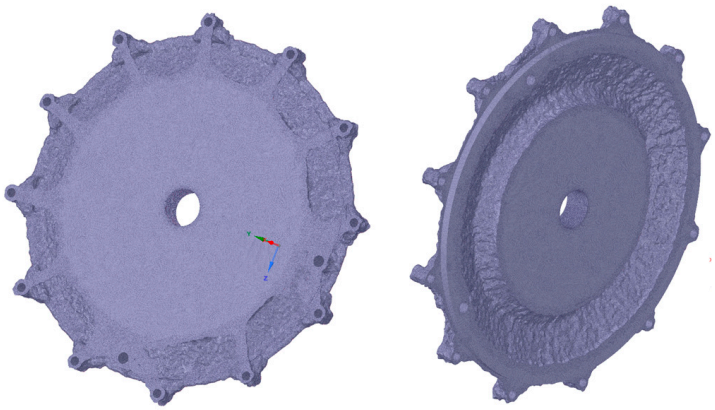


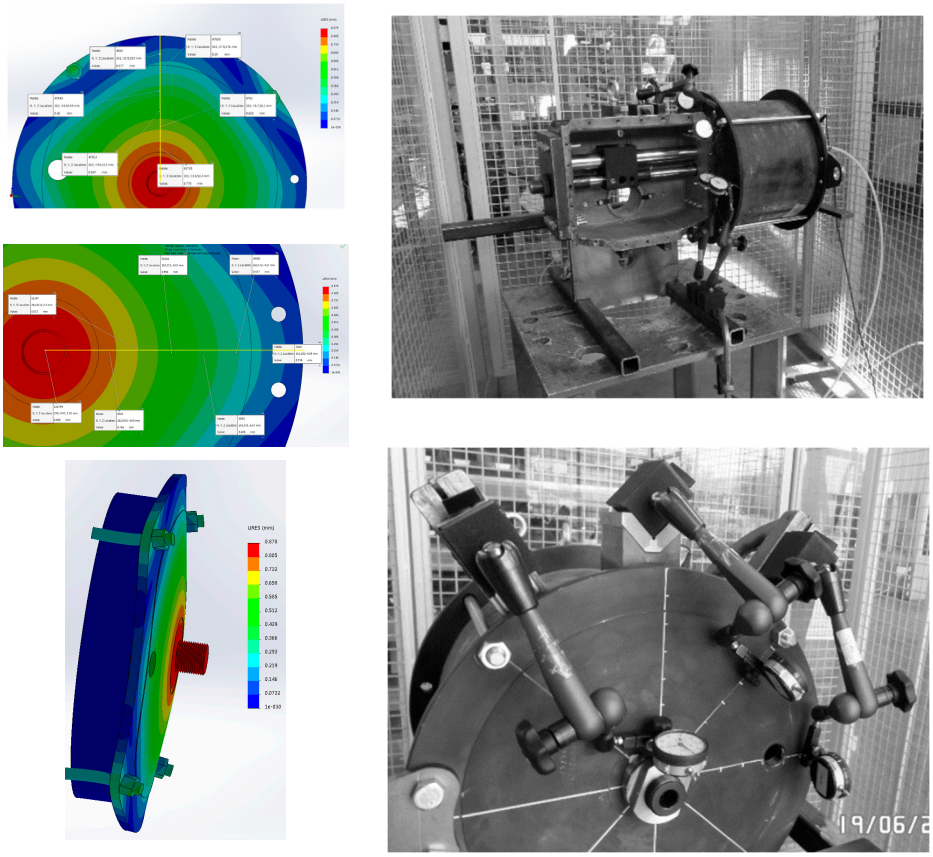
Figure 5. Study for an enhancement of the "tail" flat end topology.

3.4. Comparison of the Results by Formulas with the Results by Analysis

This comparison confirmed the consistency of the results obtained with the two approaches. An example is reported in Figure 4, where it is possible to notice that there is correspondence with a good approximation between the peak stresses in the central area of the FEA stress map and the value predicted by Roark's formulas. The analysis of the results confirmed that the DBF approach remains a reliable tool; also, if it is applied as the only tool, it leads to a conservative thickness of the tail flat end equal to 80 mm, while when integrated with DBA, the combined approach allows lowering it by 10 mm. In addition, the formulas do not provide exhaustive information as in the case of the FEA maps, regarding the distribution of stresses and deformations inside the component [18]. The application of the new methodology applied to the tail flat end allows achieving a weight reduction of 13.3%, supporting to enhancing the design from an economic point of view, to increasing the detail of the verifications contributing to a higher safety and reliability of the product and finally to reducing the development time.

3.5. Experimental Validation and FEA Calibration

For confirming the validity of the results obtained with formulas and the FEA, they were compared with data from an experimental test [19]. The FEA cannot be seen as a panacea, but as an instrument for increasing the precision of design verifications, and, for being reliable, when possible, its accuracy must be verified with laboratory tests. A prototype has been then assembled, as shown in Figure 6, and the observations conducted during the tests confirmed with good approximation the behaviors predicted by the simulations. The deformations of the tail flat end obtained with the FEA are aligned with the measured ones, as reported in the graph in Figure 7. The FEA predictions have then direct confirmation in the experimental data. This is a further confirmation that the proposed methodology can be considered as a reliable tool that, by promoting the execution of tests on virtual prototypes, helps designers to arrive at the testing of real prototypes with the most promising solutions already deeply verified.



(**a**) (**b**)

Figure 6. (**a**) Deformation map of the tail flange obtained with an FEA model of the real prototype; (**b**) phases of the direct deformation measurement on the real prototype.

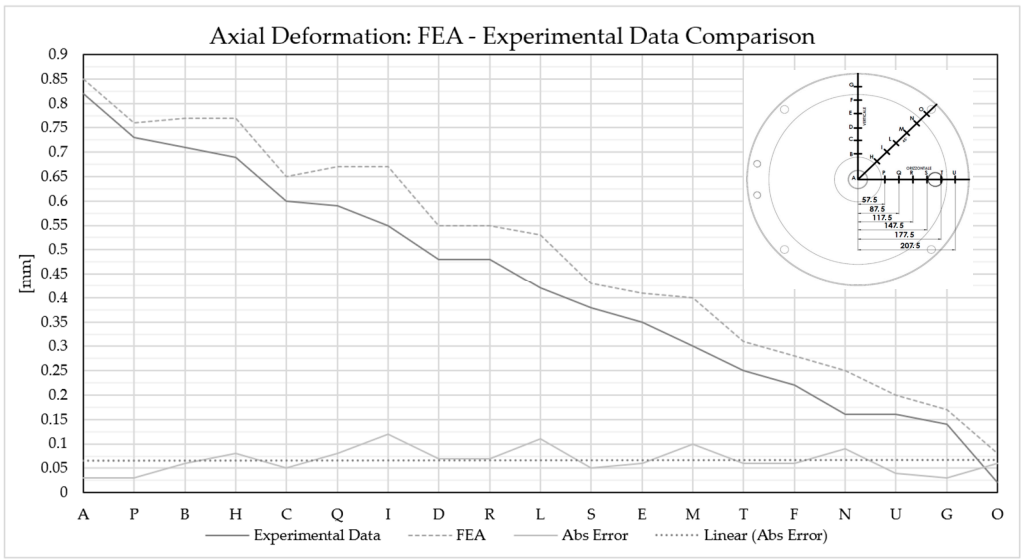


Figure 7. Comparative analysis of tail flange deformation found with the FEA and with a real prototype.

4. Integrative FEAs: Burst Pressure Test Simulation

The verifications described in the previous paragraphs tap the full potential of the DBF and DBA techniques provided by the latest versions of pressure vessel standards. However, during the design stage, some questions arise, whose answers would be very helpful to designers for having a complete overview of the equipment design, and they are not easily identifiable with standardized design procedures. For example, "Which

pressure level would the structure collapse at? Which components would fail first and in which way?" To answer these questions, a consolidated practice consists of executing expensive burst pressure tests on prototypes. Thanks to advanced simulations, it is possible to achieve an in-depth understanding of the structural failure mechanism of the equipment by resorting to "Explicit Dynamics" simulations, now more easily available and affordable in terms of resources, thanks to tools integrated in FEA packages, as in Ansys Mechanical® vers. 19.2 used here. An FEA model representing the cylinder during a pressure test was prepared using symmetry boundary conditions for reducing the execution time and calculation resources. The simulation results significantly helped us to understand the failure dynamics, shown in Figure 8. After an initial phase of bending of the lateral closing plates and of tube deformation, the elongation of tie rods becomes predominant, progressively increasing until two situations are possible: in the first one, the deformations cause the O-rings to be partially expelled from their slots, so that a fluid leakage towards the external environment starts, and the process of collapse stops. In a second case, the tie rod elongation continues until collapse is reached, with the area of occurrence of their breakage placed immediately below the nuts. Equation 9 reports the value of pressure identified by means of the FEA that could cause cylinder collapse when applied suddenly, such as following the failure of a supply pressure regulator. The advanced FEA introduced in the context of the proposed methodology helped to confirm that also in the event of a control system failure or of improper use of the product, the pressure level required to cause an accident with possible damage to people and things is high, about 2.7 times the design pressure, reducing then the probability of occurrence.

$$P_{burst} = 2.7 \times P_{design} = 2.7 \times 12\ barg = 32.5\ barg \quad (9)$$

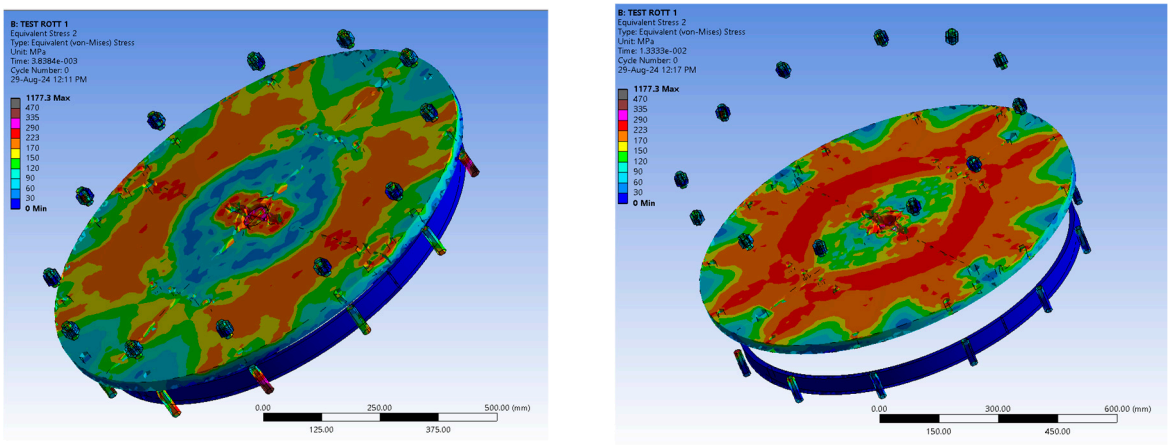


Figure 8. Explicit dynamics simulation of a cylinder burst pressure test. Images show successive stages of the progress of the collapse process.

4.1. Fatigue Failure Simulation

According to the aim of the proposed methodology, another application where an integrative FEA can bring a useful contribution for deepening the design knowledge is fatigue verification. In the fatigue study of the cylinder, FEA simulations have been introduced with a double purpose. First, according to EN13445-3 Annex B, the results of a static FEA can be introduced in the formulas of EN13445-3 Chapter 17, "Simplified assessment of fatigue life", for evaluating the maximum allowable number of full pressure cycles, from 0 barg to design pressure, that pressure-containing parts can withstand. The peak stress value obtained with the non-linear FEA of the tail plate (339.3 MPa) is introduced in Formula (10), that is, eq.17.6-18 of EN13445-3, as the value of $\Delta\sigma^*$ for the determination

of admissible fatigue cycles. This component is the one having the shortest life among all cylinder ones; therefore, the result of Formula (10) is also equal to the maximum number of fatigue cycles of the whole cylinder.

$$N = \left(\frac{46000}{\Delta\sigma^* - 140}\right)^2 = \left(\frac{46000}{339.3 - 140}\right)^2 = 53,272 \ cycles \ (0 - 12 \ barg) \quad (10)$$

Formula-based fatigue checks provide data on the single components, and the designer must rebuild with a deduction the overall view of the life of the equipment as a whole. For this reason, a second purpose of an FEA is to help designers by providing an immediate, more accurate overall view of the fatigue behavior of the cylinder as a whole thanks to fatigue life tools. For this second purpose, an integrative fatigue simulation with the Ansys Mechanical® vers. 19.2 fatigue tool was performed. The results reported in Figure 9 show the minimum allowable fatigue cycles predicted by the FEA (57,129), aligned with the result of Formula (10) (53,272). Figure 9 also shows the areas where a failure could arise, in correspondence with the first threads of the stopper bolt hole. By repeating the simulation with lower operating pressures, it was possible to check and have confirmation in a single overall view of the infinite fatigue life of all the components.

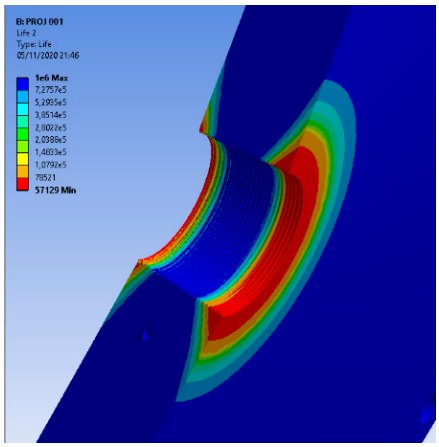
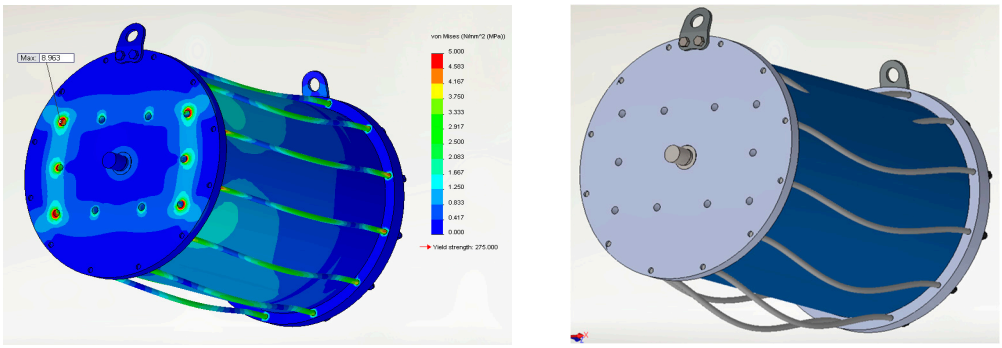


Figure 9. Fatigue life simulation of the tail flat end subject to 0–12 barg pressure cycles with indication of the sections having lower fatigue life (57,129 cycles).

4.2. Seismic Simulation

The last application where the integrative advanced FEA allowed us to effectively increase the knowledge of cylinder behavior was a seismic verification, performed to comply with PED (Pressure Equipment Directive 2014/68/UE) certification requirements. The analysis considers the action exerted on the cylinder by a spectrum of horizontal and vertical accelerations, simulating a strong entity earthquake, exciting its structure with frequencies equal to its first natural frequencies. The analysis was performed with the "Response spectrum Analysis" tool of Ansys® Mechanical vers. 19.2 by following the rules of EN1998-1 [20] and by imposing a value of peak ground acceleration provided by European seismic maps equal to 0.5 g. The structural response and the cumulative stress were calculated by summing the contributions from each vibration mode with the technique of combination of the modes called "Square Root of the Sum of Squares" (SRSS). As shown in Figure 10, the higher stresses are present in the head flat end near the stud bolt holes for the connection to the actuator central body, in the tube and tie rods. The simulations helped to confirm the safety of the design by showing von Mises stresses below the maximum admissible ones.



(**a**) (**b**)

Figure 10. Response spectrum simulation for a seismic verification of the cylinder: (**a**) map with the areas most stressed by the acceleration spectrum an earthquake; (**b**) representation of one of the cylinder vibration modes.

5. Conclusions and Proposals

All the presented results contribute to showing that a new design methodology integrating the consolidated "design-by-formulas" and "design-by-analysis" approaches with additional verifications based on an advanced FEA could bring significant improvements in the design process of pressure-containing equipment. This methodology can support manufacturers in introducing products guaranteeing high levels of safety, reliability, quality, and performance. The case study presented here demonstrates that it is possible to arrive at the phase of testing of real prototypes with a design deeply improved thanks to virtual prototypes, reducing the time required for the product development, the cost of testing, and the risk of design changes during the final development stages. The application of the proposed methodology allows one to obtain, with affordable resources, precious insights that are useful during product life, like in the occasion of a root cause analysis due to a failure in the field, allowing one to ascertain if the issue is due to improper use of the equipment rather than to an intrinsic design defect. The proposed design methodology, when applied during the development of a product together with value engineering tools and inserted in a work context based on teamwork aimed at mutual enrichment, constitutes an essential component of a new cross-functional and holistic product development strategy proposed by authors in other works already mentioned and aimed at bringing a positive and original contribution to the world of design. The final purpose of this study could be resumed in a double proposal. First, we suggest that designers apply the methodology and issue as the final output a "diary" with useful information to be consulted in the future, integrating the classical "design manual" with presentations of the advanced FEA results. Secondly, we suggest bringing the results presented here to the attention of the EN13445-3 standard technical committee for contemplating the use of advanced FEA simulations as a possible option during pressure vessel design. The new methodology could be the subject of a new "informative" annex, where the benefits of integrative "design checks" based on advanced FEA are described with the aim of developing new pressure-retaining equipment that is safer and more reliable.

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