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FINITE ELEMENT ANALYSIS OF A COMPRESSOR HOUSING USED IN HIGH PRESSURE REFRIGERATION SYSTEM

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ABSTRACT

In the design of compressors running with high pressure refrigerants, safety aspects must be a mandatory concern. Moreover, when dealing with high pressure levels, compressor components have their original design adapted to withstand such a high pressures, particularly acoustical mufflers, external housing, and compression mechanism. Regarding the external housing, the design approach goes beyond acoustical and aesthetics features as mostly observed in current refrigerating compressors. In order to safety enclose the compression mechanism the application of a proper design methodology is mandatory to safeguard the structural integrity of both the compressor external housing and the whole refrigerating system. Looking for acceptable, cost effective safety factors, a simultaneous design approach including advanced structural mechanics techniques, experimentation, safety Codes revision, and Computer Aided Engineering (CAE) tools application is mandatory. The aim of this work is to present a new development approach, concerning structural design of a compressor housing used in high pressure refrigeration system. Numerical and experimental results will be compared among each other aiming to evaluate some ASME Codes criteria and design procedures.

INTRODUCTION

DESIGNING A COMPRESSOR HOUSING PARTICULARLY DEVELOPED FOR USE IN HIGH PRESSURE REFRIGERATION SYSTEM

A typical refrigeration system used in household and commercial application is composed basically by a compressor, an evaporator and a heat exchanger. The propose of the compressor is to raise up the pressure from point 1 to point 2 in the refrigeration cycle diagram, depicted in Figure 1, as already depicted by Bosco [1].

The compressor itself is composed by the mechanical pump, electrical motor and a external housing enclosing the whole compressor system. In the most of refrigeration application, the working pressure in the housing lays in a very low value has not been a big concern regarding mechanical strength. The housing thus, has a esthetical and acoustic commitment more than a structural concern.

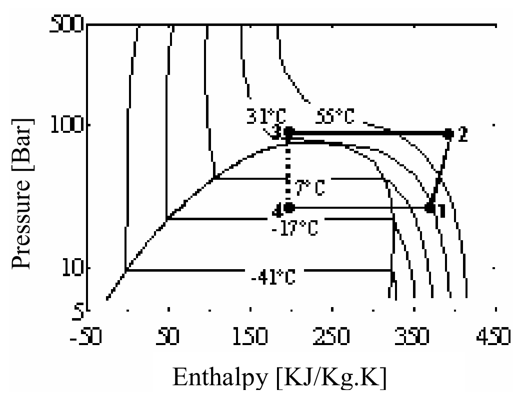

Figure 1 – CO_2 pressure-enthalpy diagram

However, when dealing with high pressure refrigeration system, the safety aspects and structural reliability of the compressor housing is the most concern. In this case, the housing is subjected to internal pressure in such level that structural strength must be very well analyzed, tested and evaluated.

Moreover, the compressor housing is formed by three parts joined by welding, as depicted in Figure 2. This shall be another important point of a deeply study and investigation, once it can lead to a weak point for the structure integrity.

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FEA MODELING OF THE COMPRESSOR HOUSING

The compressor housing is composed of several parts and assembled together by welding process. Aiming to make a finite element (FEM) analysis considering stress and deformation, some simplification should be done. Fortunately the power of FEM mesher have been achieve a such level that the original 3D CAD model Figure 2 can be used and directly transferred among CAD and CAE package tool.

However, it is notorious that a FEM analysis of a 3D full model takes long time and computer expensive. As 3D, understand a full CAD three-dimensional solid model. Thus, in the beginning of the analysis, a 2D axisymmetric simplification of the main body of the housing has been taken into account. A 2D or two-dimensional axisymmetric model can be illustrated as a revolution solid (axisymmetric) generated by a constant section (2D) rotating 360º around a revolution axes. Regardless that the model looks like a 180 symmetric model and not exactly an axisymmetric one, the last one has been chosen as a simplified analysis approach.

The 3D and 2D axisymmetric model used in the present simulation is depicted in Figure 32 e Figure 43.

The boundary condition in the FEM analysis consists in applying a internal pressure up to 350Bar and fixing the compressor housing externally at the base plate.

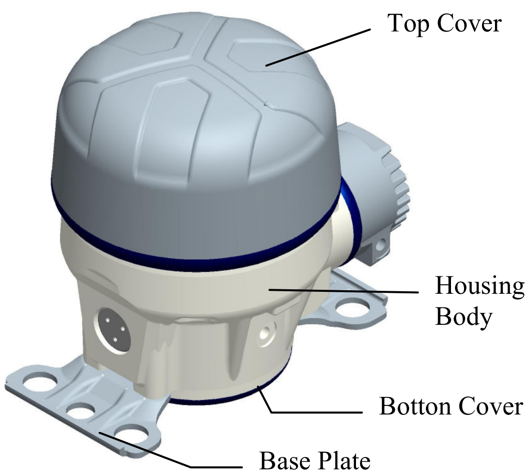

Figure 2 – 3D CAD model of the compressor housing

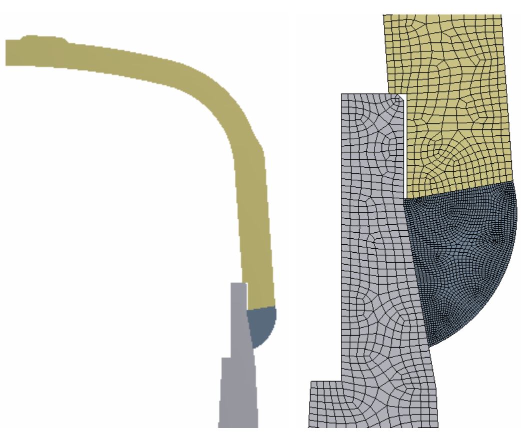

Figure 3 – Axisymmetric model of the top cover of the compressor housing and FEM mesh refined at welding

In the present analysis of even 2D and 3D, the welding region has been modeled as a 3rd part in the welding joint as can be seen Figure 3 and the mesh has been refined aiming to get a representative stress across its section for further analysis. The 3D FEM mesh is depicted in Figure 4 and is formed by 177,000 parabolic tetrahedral elements having a total of 600,000 Degree of Freedom (DOF). The mesh is so there are at least three elements through the thickness.

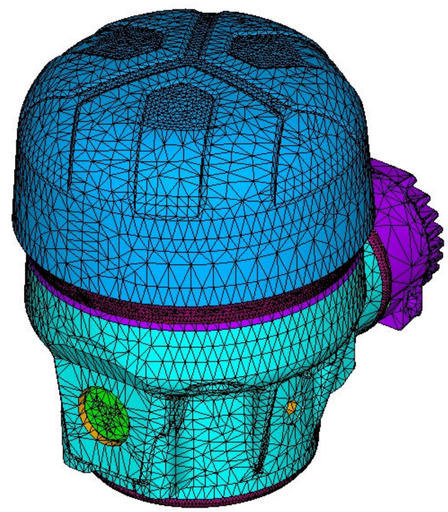

Figure 4 – 3D FEM mesh for elasto-plastic analysis.

NUMERICAL ANALYSIS OF THE COMPRESSOR HOUSING

MATERIAL CHARACTERIZATION

In order to have a well understanding of the material used in the compressor housing, a deep analysis of material standards has been performed. Concerning the possibility of having the housing in cast, both gray cast iron and nodular cast iron have been evaluated. Due the notorious superior performance of the nodular comparing to the gray cast iron, this has been chosen as the material for the cast vessel.

For the cast iron specifications, the following ASME Code has been analyzed and taken into account: ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, Rules for Construction of Pressure Vessels, 2004 E according to [2]. A brief summary of the Code concern is depicted in Table 1

Table 1 – Material specification and its restriction according the ASME code

Elongation	ASME Part code	Restrictions	Material
≥18%	Part UCD e Code Housing 1939	$P_{design} \leq 7$ MPa Weld is allowed	SA-395 Class 60-40-18
≥ 15% e ≤ 18%	Part UCD	$P_{design} \leq 7$ MPa Weld is not allowed	SA-395
≤ 15%	Part UCI	$P_{design} \leq 1{,}1$ MPa	SA-278 SA-47

The first approach was to develop a cast material that reaches the specification of Part UCD given by Table 1. However, the pressure limitation for the use of Part UCD is unfortunately less than the working pressure for this compressor. This led us to develop our in-house approval test and material analysis. Thus, aiming to evaluate a reasonable and cost effective material and make a trade-off analysis among feasibility and safety, the following approach has been considered in the material development:

(a) Quality and metallographic homogeneous of the present material;
(b) Weld joints evaluation in order to get all of their characteristics;
(c) Mechanical properties of all material, including the main body material and welding properties;
(d) Fatigue and fracture mechanics properties;
(e) Experimental and a complete experimental work to evaluate all materials and components.
(f) A structural analysis by the FEM, in which the Project by Analysis is taken into account.

Regarding the stress-strain relation for the cast material, a curve has been obtained with specimens considering the raw material as cast, after welding and after welding plus thermal treatment as is depicted in Figure 5.

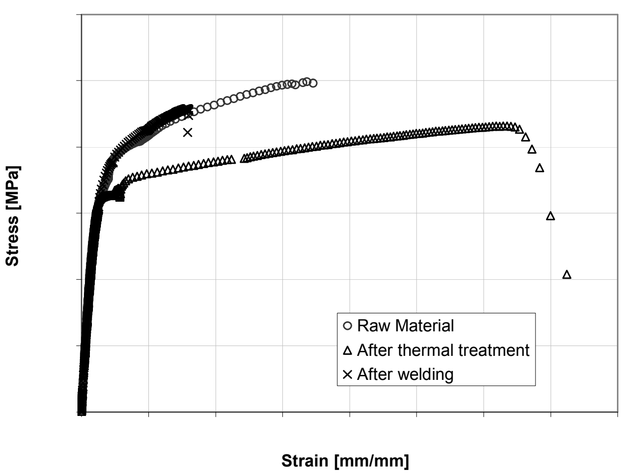

Figure 5 – Stress-Strain curve for raw cast iron, after thermal treatment and after welding.

FEA ANALYSIS AND STRENGTH CRITERIA

A similar analysis based on ASME Codes procedure and criteria as described below has already been performed by Bosco [1]. Some Code descriptions are listed below:

According to [3], ASME BPV Code Section VIII, Division 2, Mandatory Appendix 4, Design Based on Stress Analysis, the following definition is described:

Plastic Analysis. Plastic analysis is a method of structural analysis by which the structural behavior under given loads is computed by considering the actual material stress–strain relationship and stress redistribution, and it may include either strain hardening, change in geometry, or both. The limits of general membrane stress intensity (4-131), local membrane stress intensity (4-132), and primary membrane plus primary bending stress intensity (4-133) (Appendix 4) need not be satisfied at a specific location if it can be shown that the specified loadings do not exceed two-thirds of the plastic analysis collapse load determined by application of 6-153, Criterion of Collapse Load (Appendix 6), to a load deflection or load strain relationship obtained by plastic analysis. When this rule is used, the effects of plastic strain concentrations in localized areas of the structure such as the points where hinges form must be considered. The effects of the concentrations of strain on the fatigue behavior, ratcheting behavior, or buckling of the structure must be considered in the design.

According to [3], on the ASME BPV Code, Section VIII, Division 2, Subsection NB, Article NB-3000 Design, and Mandatory Appendix 6, the following definitions are described:

Plastic Analysis — Collapse Load. A plastic analysis may be used to determine the collapse load for a given combination of loads on a given structure. The following criterion for determination of the collapse load shall be used. A load–deflection or load–strain curve is plotted with load as the ordinate and deflection or strain as the abscissa. The angle that the linear part of the load– deflection or load–strain curve makes with the ordinate is called θ. A second straight line, hereafter called the collapse limit line, is drawn through the origin so that it makes an angle $\phi = \tan^{-1} (2 \tan \theta)$ with the ordinate. The collapse load is the load at the intersection of the load–deflection or load–strain curve and the collapse limit line. If this method is used, particular care should be given to ensure that the strains or deflections that are used are indicative of the load carrying capacity of the structure.

This procedure is depicted in Figure 6.

Plastic Hinge. A plastic hinge is an idealized concept used in Limit Analysis. In a beam or a frame, a plastic hinge is formed at the point where the moment, shear, and axial force lie on the yield interaction surface. In plates and shells, a plastic hinge is formed where the generalized stresses lie on the yield surface.

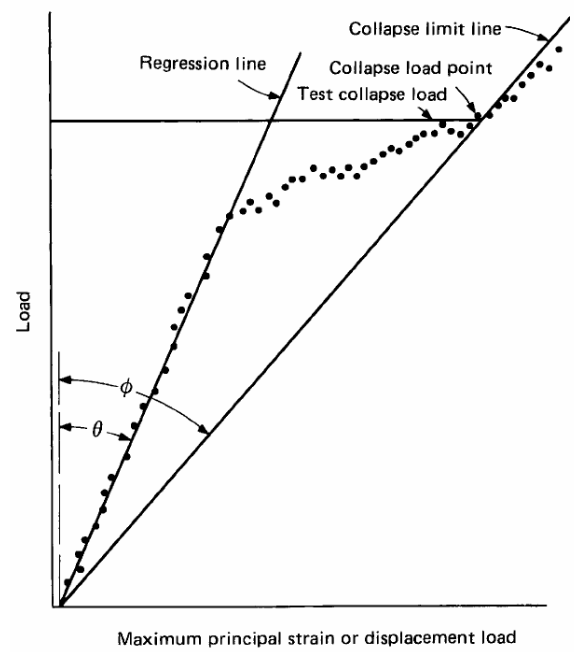

Figure 6 – Construction of curve to determine collapse load according to 6-153 [03]

Depending on the region of the shell, the corresponding theory should be applied in order to evaluate their safety factory, meanly in discontinuity region.

For that design pressure, each specific transition should be analyzed concerning its Tresca Stress and whether or not the presence of plastic hinge.

Welding Analysis

In the design of a weld joint in a pressure vessels, special attention has been taken to follow or at least figure out how the ASME code deal with this kind of joint. Joint efficiency is defined as the ratio of strength of a joint to the strength of the base metal, expressed in percentage.

In the ASME BPV Sec. VIII Div. 2 Code the section in which this issue is dealt with is the Part UW, Requirements for Pressure Vessels Fabricated by Welding. In this section, various type of joints, like *butt joint* and *lap joint* are specified with their corresponding joint efficiency. Thus, one should use the corresponding strength ratio multiplier in the design of pressure vessel for a particular joint. See Table 2.

From Shigley [4], using the equilibrium condition shown in Figure 7, the equations (1) to (3) can be developed where σ_{eq}' is the von Mises equivalent stress.

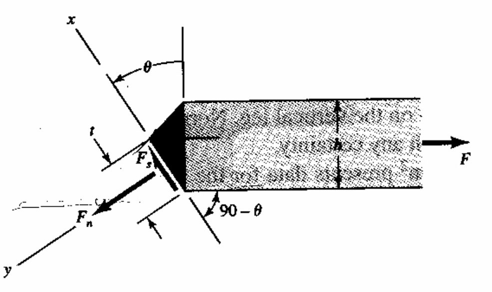

Figure 7 – Equilibrium of force in a weld joint section

$$\sigma = \frac{F_n}{A} = \frac{F\cos\theta(\cos\theta + \sin\theta)}{hl} = \frac{F}{hl}\left(\cos^2\theta + \sin\theta\cos\theta\right) \quad \textbf{(1)}$$

$$\tau = \frac{F_s}{A} = \frac{F\sin\theta(\cos\theta + \sin\theta)}{hl} = \frac{F}{hl}\left(\sin\theta\cos\theta + \sin^2\theta\right) \quad \textbf{(2)}$$

$$\sigma_{\acute{e}qv} = \left(\sigma^2 + 3\tau^2\right)^{1/2} = \frac{F}{hl}\begin{bmatrix}\left(\cos^2\theta + \sin\theta\cos\theta\right)^2 + \\ 3\left(\sin^2\theta + \sin\theta\cos\theta\right)^2\end{bmatrix}^{1/2} \quad \textbf{(3)}$$

Table 2– Maximum allowable joint efficiencies for Arc and Gas welded joint, from ASME Sec. VIII Div. 2, Table UW-12

Joint Type	Degree of Radiographic Examination		
Butt joint	1.00	0.85	0.70
Single full fillet lap joint without plug welds	NA	NA	0.45

Safety factor for working pressure

Regardless the table 1 states that the limit pressure should be 70Bar, for such application of referred compressor, the pressure should higher than is allowed. This is why the development of a special material, extensive tests and analysis has been taken mandatory.

For this compressor, the working pressure is P_w = 9 MPa (90Bar), the material yielding limit is σ_y = 313MPa and the angle varies from θ=0º, 36º and 90º

Appling equation (3) for the critical angle θ one gets:

$$\sigma'_{eqv} = 89MPa$$

Considering a joint efficiency according to the table 2,

$$\sigma'_{eqv} = \frac{89}{0.45} = 197{,}8MPa$$

this results in a safety factor of

$$s = \frac{313}{197{,}8} = 1{,}6$$

Even the consideration of the joint efficient may be conservative, the results have achieved a good safety margin. However, in order to optimize the compressor housing development, the actual welded joint efficiency should be evaluated and experimental tests take place aiming to reduce the conservativeness of such code pursuance.

Furthermore, the fatigue analysis of the weld joint is still a challenge to be evaluated once a lot of factors influence the welding strength, like heating, residual stress, cooling cracking and material of the wire. This fatigue strength performance has been evaluated in experimental fatigue tests as can be seen in the Figure 21.

A FEA analysis of the weld joint considering the joint coefficient above is depicted in the Figure 8.

FEA Analysis – Collapse load

Aiming to evaluate the collapse load in a plastic analysis, two different approaches have been performed for material behavior: Bilinear Elastic-plastic and Multilinear elastic-plastic.

The first method is easier to implement since only two parameter are requested: Young modulus and yield strength. The multilinear elastic-plastic method otherwise, request an actual strain-stress curve and must be taken from experimental results in material specimens. However, the second approach tends to be more precise since the actual elastic-plastic behavior of the material is taken into account as shown in Figure 9.

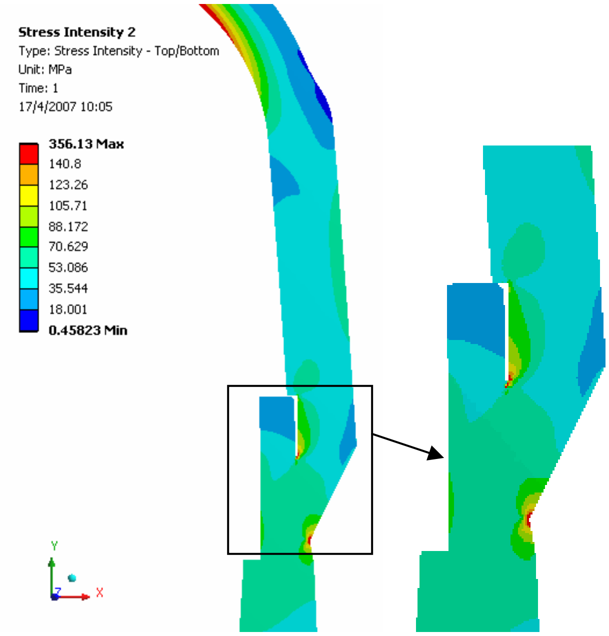

Figure 8 – Elastic-plastic Axisymmetric analysis of the Weld joint for 90Bar post-processed for yield stress multiplied by joint efficiency.

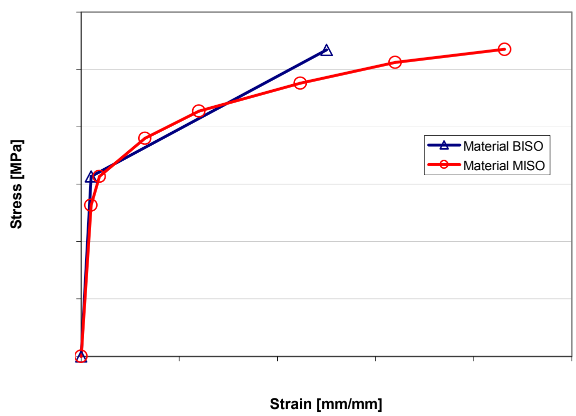

Figure 9 – Stress-Strain relation used in Elasto-plastic analysis. Bilinear (BISO) and Multi-linear (MISO)

Two main critical points have been evaluated according ASME procedure given in 6-153 depicted in Figure 2: Top cover and bottom cover. The result is depicted in Figure 10.

And applying the criteria of twice the tangent according Figure 6, the results are the depicted in Figure 11 and Figure 12.

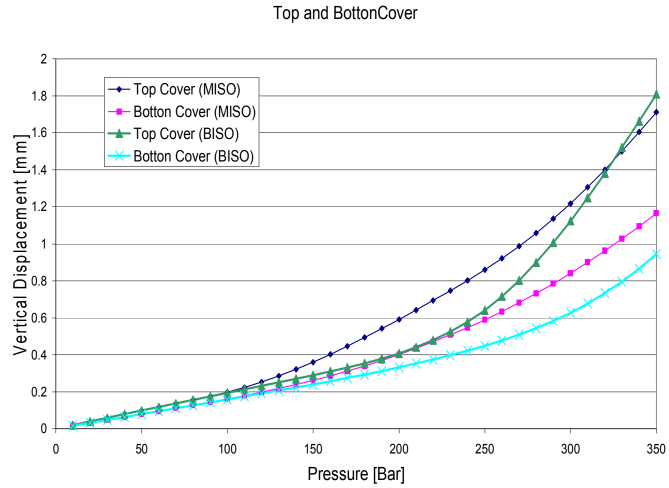

Figure 10 – Vertical displacement versus pressure for two critical points in the housing and two material behavior

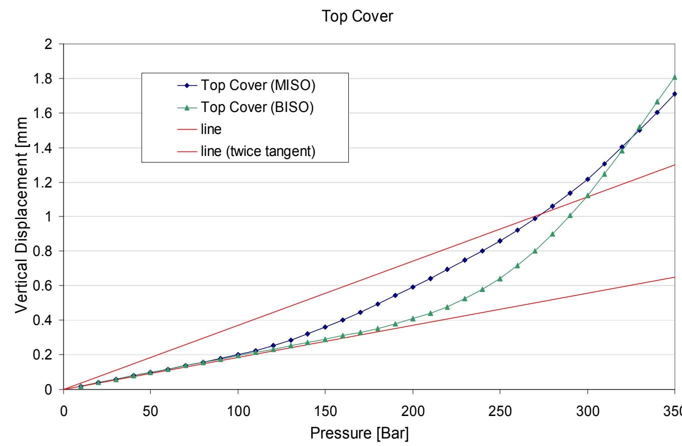

Figure 11 – Collapse load result for the critical point at the top cover.

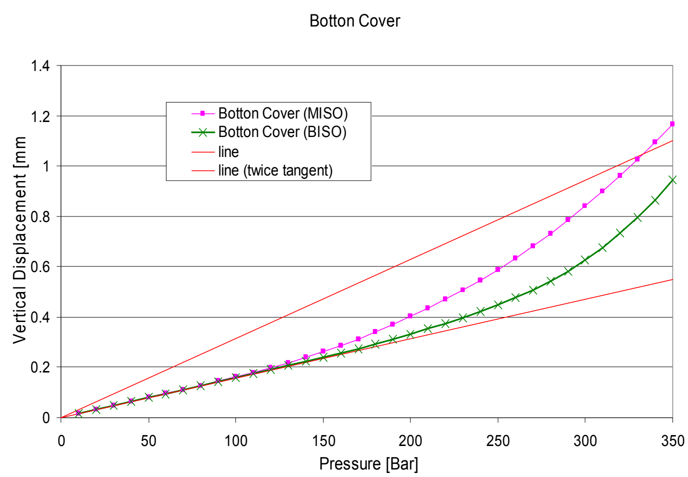

Figure 12 – Collapse load result for the critical point at the bottom cover.

Analyzing the Figure 11 and Figure 12 above, the pressure limit given by collapse load criteria in the top cover and bottom cover are those according to Table 3.

Table 3– Collapse load pressure for the housing

Material behavior	Top Cover	Bottom Cover
BISO	300	360*
MISO	275	325

* extrapolated

Thus, it can be seen the weakest point of the housing is the top cover considering the collapse load criteria.

Moreover, to check for the presence of plastic hinge in the housing for such level of pressure (given by collapse load analysis) a static non-linear analysis have been performed and the results are the Figure 13.

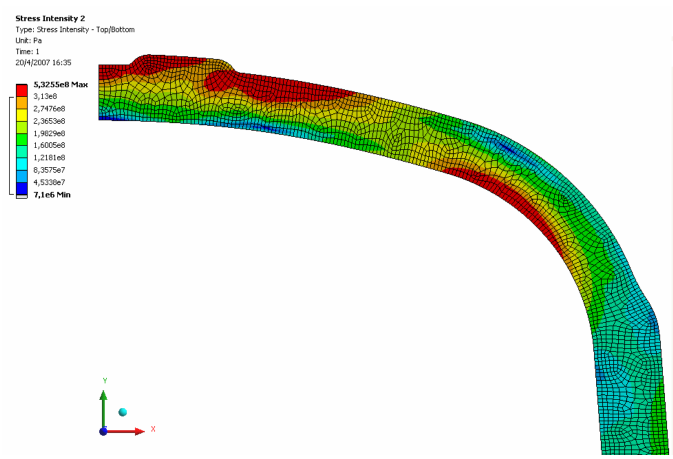

Figure 13 – Plastic hinge evaluation at top cover for 275 Bar of internal pressure.

It can be noticed in the Figure 13, there is no region where the maximum stress intensity across the thickness reaches value grater than the yield strength of the material (313MPa). That means the housing would not face collapse and neither any plastic hinge has been formed at this given pressure. The same analysis has been done on the 3D model for bottom and top cover Figure 14 and Figure 15. Maximum stress intensity is defined as equation (4):

$$\sigma_i = \max\left(\left|\sigma_1 - \sigma_2\right|, \left|\sigma_2 - \sigma_3\right|, \left|\sigma_1 - \sigma_3\right|\right) \qquad \textbf{(4)}$$

EXPERIMENTAL RESULTS AND FEA COMPARISON

In order to evaluate quantitatively the FEA analysis and validate de numerical model, an experimental measurement of an instrumented compressor had been done. The compressor was gauged with several electric strain-gages in its most critical points regarding deformation and strength, see Figure 16. Those weak regions were evaluated in a 3D FEM analysis.

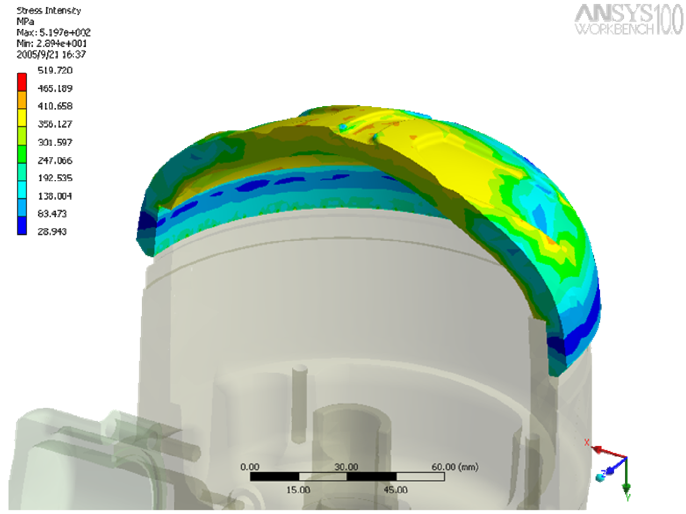

Figure 14 – 3D stress evaluation checking the presence of general plastic hinge formation in the top cover of the housing.

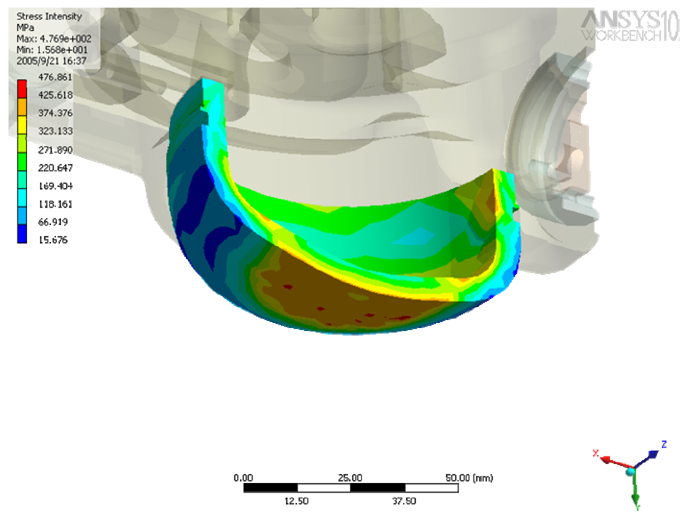

Figure 15 – 3D stress evaluation checking the presence of general plastic hinge formation in the botton cover of the case.

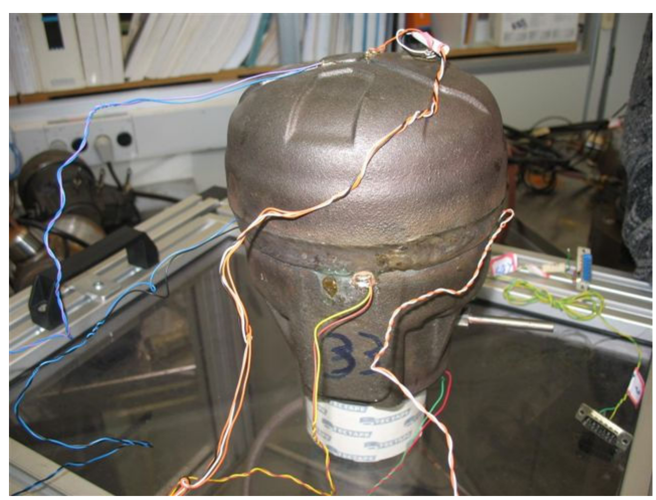

Figure 16 – A CO_2 compressor gauged with several strain gages in the hidrostatic pressure test.

The following figures depict the results of the FEM plastic analysis in comparison with the gauged compressor. The graphs in the Figure 17 and Figure 18 compare the experimental results measured by the strain-gages with the numerical FEM elastic-plastic analysis. The curve is a composition of elastic plus plastic element deformation.

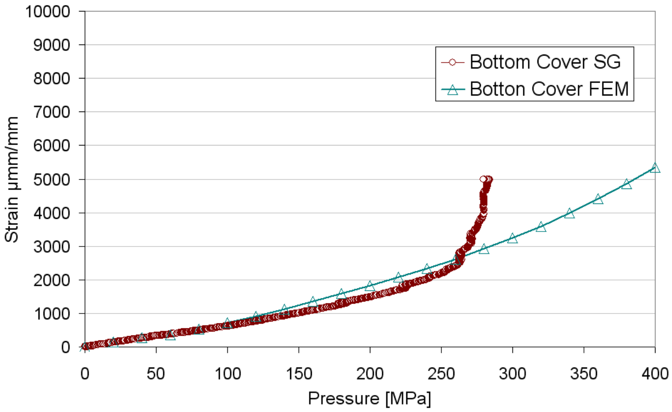

Figure 17 – Pressure x Strain in the Strain-gaged located at the bottom cover of the pressure vessel. FEM plastic analysis and experimental comparison.

Hydrostatic pressure test

For each design proposes aiming to check the extreme pressure resistance of the compressor housing, a hydrostatic pressure test performed. This is a complete static strength test where all the housing components, that is, the main body material and weld joint are subject to the same pressure till its burning collapse.

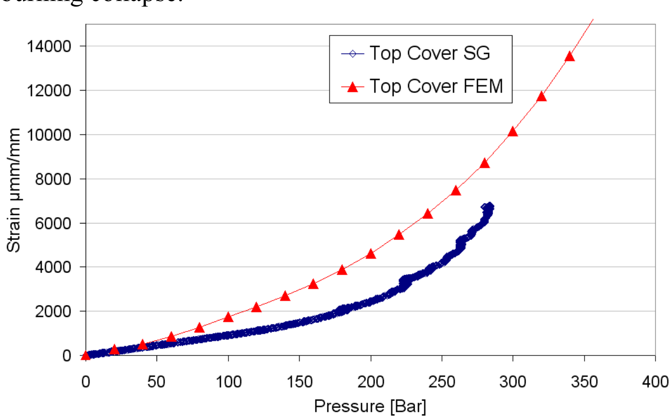

Figure 18 – Pressure x Strain in the Strain-gaged located at the top cover of the pressure vessel. FEM plastic analysis and experimental comparison.

In the Figure 19 is depicted a FEM analysis of the pressure vessel in an extreme collapse load simulating the hydrostatic pressure test done in the test machine show in Figure 20, where the experimental stress results have been also measured.

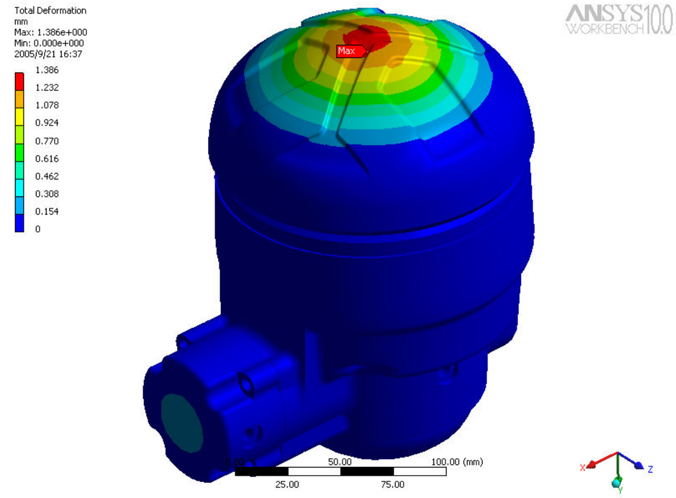

Figure 19 – FEM analysis of maximum deformation depicting the weakest vessel region.

Figure 20 – A CO_2 compressor during the extreme hydrostatic test pressure.

Fatigue Test

Aiming to evaluate the fatigue performance of the compressor itself and also different configurations for material, welding and so one, fatigue tests have been performed in special specimen samples and in the pressure vessel.

The fatigue test in the specimens has followed standardized sample tests (ASTM E399/90) and it has been cut from welded cast iron plates.

The fatigue limits have been evaluated in different places (melted zone, heat affected zone and base metal) with special indentation on the samples. For the fatigue test of the final compressor housing, both high and low side of the enclosed vessel part have been submitted to pressure load according working condition for each compressor side. A fatigue life testing running in the high pressure side of the compressor housing can be seen on

Figure 21, where few housings are submitted to a variable pressure cycle. This is a special company's in house development machine designed concerning safety aspects for the operator. The pressure cycle loop is monitoring by software and the number of cycles and pressure conditions data is registered. The actual load condition and number of cycles at those pressures for housing approval depending on each refrigeration system.

The notorious advantage of submitting the whole housing to fatigue load test is the evaluation of all vessel issues, like welding, materials, and all other possible weak points of the housing design, checking the safety factor at all.

Figure 21 – A fatigue test of the compressor housing running on a special developed machined

CONCLUSION

The Code evaluation and analysis is surely an excellent and mandatory approach in the design of such kind of compressor. It takes the advantage of a huge experience on the development of pressure vessels, if we considered the ASME for instance, but also the intrinsic experiences of each specific standard committee. Moreover, following some good practices of those Codes leads a development to be more assertive in the very beginning of the design, like the material selection, the process chosen, and the manufacturing issues for example.

An efficient way to design such compressor would be that one where all involved issues are very well understood and deeply analyzed. For example, figuring out the actual strength of the component throughout advanced FEA analysis and experimental verification like has been done in the present work.

The Finite Element Analysis allows the design engineering to have a deep understanding of the component structural behavior and to achieve a better design where each part of the structure are optimized for the such working condition.

The present work has figured out the application of advanced tools like FEA and hybrid analysis in the design of a special pressure vessel. Since the compressor casting consists in a special pressure vessel, where there are no analytical and close equations to analyze it, the FEA becomes a powerful design tool. As can be seen on FEA and experimental results comparison, this analysis approach enables the engineers to evaluate their design even in the beginning of the product development predicting the further strength performance.

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