Assessment of critical defects in a PSA pressure vessel subjected to cyclical loading using BS-7910 procedures

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ABSTRACT. *In a regular inspection, some embedded cracks were found in the inlet nozzle-head weld of a PSA pressure vessel. Instead of advancing the retirement of this equipment and waste much time until the refinery purchase a new vessel, a structural stress analysis using the finite element method was performed in order to obtain the stress field around the crack site, considering the loading cycle. Despite the acting load is only the internal pressure, the nozzle-head weld is a region where a complex stress state is present (bending and axial stresses). ASME VIII Division 2, Appendix 5 addresses this issue by applying a rule for multiaxial fatigue life estimation for non-proportional loading. With the estimated life information, the next step was the application of the British Standard 7910 procedure to decide if the equipment can operate safely. The calculation also assesses the crack growing by using the Paris law. Then it was computed how long the cracks would take to get to their critical size and then retire definitively the equipment.*

INTRODUCTION

Brazilian law requires that all pressure vessels are periodically inspected, opened and tested hydrostatically. This is the case for most of such equipments in refining industry. For Pressure Swing Adsorption (PSA) vessels this is not always valid: besides the equipment is externally inspected, it is not practical open it, due to the very expensive catalyst. Then, external ultrasonic inspection is widely used. Further, this pressure vessel works in a severe cycle of pressurization/depressurization (from maximum to minimum in about 30 minutes). These equipments are designed by ASME VIII Division 2 [1], and, theoretically, they are intended to work during 20 years without any problem.

However, during a periodic external ultrasonic inspection, some cracks were found nearby the upper nozzle (where the purified hydrogen leaves the vessel). Since the opening of this vessel leads to the purchase of a new one, it was decided by the refinery inspection team perform a fitness for service analysis in order to decide if the equipment can safely operate with those defects.

The first goal of this study is to perform a fitness for service analysis based on the British Standard BS-7910 [2] requirements in order to access the flaws criticality and acceptability. Input data for this analysis are both material characterization and a finite element analysis considering the cyclic pressure load. Due to the location of the cracks (near nozzles), the monotonic pressure load leads to a complex stress distribution varying in time, one of the characteristics of multiaxial fatigue.

After assessing the cracks, it is necessary to evaluate the new fatigue life of the structure, since the defects can (and do go) growth in operation. For this purpose, a simple crack grow rule based on Paris Law was performed, using the pressure gradient calculated in the first step of this study. This is considered the second goal of this analysis.

The theoretical basis for these analyses will be shortly presented in the first part of this paper, addressing the main concepts of PSA process, fatigue and fracture mechanics (emphasizing crack growth). This is the background for the procedure compiled in BS-7910, which will be applied for the real defects assessment in the second part of this paper. After, we will try to estimate the crack growth using Paris Law and the cyclical load. Finally, some conclusions concerning the structural integrity of this pressure vessel will be presented.

THEORETICAL BASIS

Pressure Swing Adsorption (PSA) process

With the need for processing increasingly heavy oils and the increasing requirements of environmental laws that require the production of cleaner oil products, refineries need more hydrogen used in plants such as hydrotreating and hydrocracking process units. The hydrogen can be generated through two processes: partial oxidation of heavy fractions of hydrocarbons (eg. fuel oil) or steam reforming of light hydrocarbon fractions (eg, natural gas, fuel gas, naphtha etc.). To obtain high purity hydrogen, it is necessary to pass this gas through a CO/CO_2 absorption section, such as the Swing Pressure Adsorption (PSA) units.

The PSA process is based on the principle that adsorbents are capable of adsorbing more impurities at a higher partial pressure of the gas phase than in a lower partial pressure. Impurities are adsorbed in a fixed bed adsorption at high pressure, where the separation of impurities from the raw gas is made to the pressure adsorption which is provided by the pressure of the gas supply. The feed gas flows through the adsorber in an upward direction inside the equipment, and impurities such as CO, CO_2, hydrocarbons and nitrogen are selectively adsorbed on the adsorbent. The purified hydrogen exits the adsorber at the top and is discharged into the sink product.

Over time, the purity of hydrogen can no longer be guaranteed, and the adsorber needs to be regenerated. For this reason, this process is cyclical, consisting of 3 phases: depressurization, purging and pressurization. All cycle takes about 30 minutes to be completed.

Multiaxial Fatigue Assessment in ASME VIII Division 2

When there is a need to evaluate a situation where fatigue is a failure mode, following to ASME VIII Division 2 is the correct way. For such situations, the code requires a full stress analysis which, will provide the maximum stress at a given point or location (caused by thermal stresses or geometric notches).

The precise determination of peak stresses is essential to properly establish an allowable fatigue life of the mechanical component. To determine the stresses ASME VIII division 2 lists some typical stress concentration factors, but handbooks such as in Peterson's book [3], experimental data or a detailed stress evaluation using the Finite Element Method are utilized.

Complex loading cycles often occur where there is reversal of stresses. In these cases, a structure can be loaded such that multiple components of stress are induced at critical regions (e.g. bending). Such loading is referred to as multiaxial, and the evaluation of fatigue damage accumulation under this condition has proven to be an extremely complex phenomenon.

Multiaxial loading can be categorized as proportional or non-proportional [4]. Proportional loading results in principal stresses fields that remain fixed in proportion with time. In other words, the stress components fluctuate at constant ratio.

For non-proportional loading (NPL) a number of theories have been developed. The ones receiving more attention consider strain and/or stress histories over specific planes at highly stressed regions to compute accumulated fatigue damage.

The NPL procedure used in this work requires the computation of the alternating stress (S_{alt}) [1] in order to determine the allowable number of cycles of the component. This stress is function of the difference between principal stress (σ_1, σ_2, σ_3) calculated for the extreme points of the cycle. Figure 1 shows the pressure loading of the analyzed equipment, obtained directly from the real-time data management refinery system.

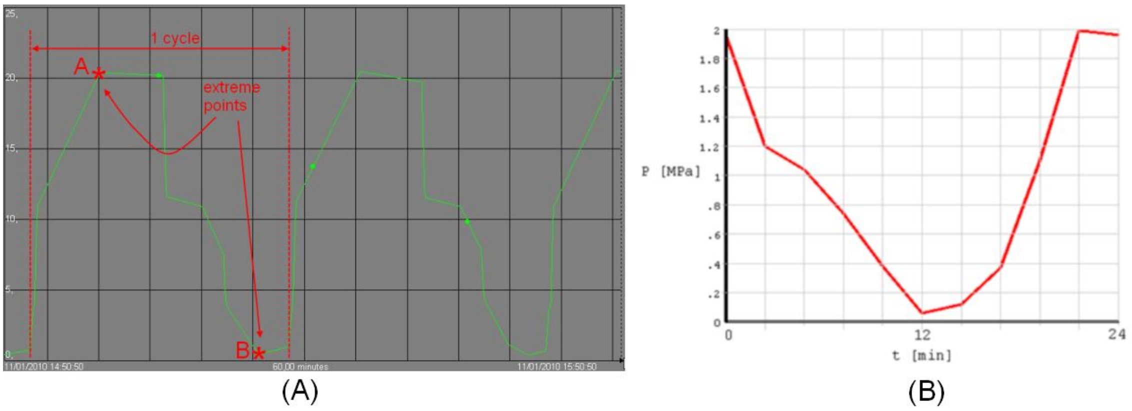

Figure 1: (A) Cycle obtained from the online monitoring system; (B) Processed cycle used for calculations.

It can be notice from Figure 1 the principal stress varying their direction when the load changes from point A to B. The stress components at the time of A and B were computed (σ_{xx}, σ_{yy}, σ_{zz}, τ_{xy}, τ_{xz}, τ_{yz}) and similar stresses components were combined obtaining the "stress range" (Ex:$\sigma'_{xx} = \sigma^A_{xx} - \sigma^B_{xx}$). Stress range in turns enable the calculation of principal stresses (σ'_1, σ'_2, σ'_3) and these principal stresses are used in order to obtain the stress components used to calculate (S_{alt}), as stated by the Eq. 1. The whole process follow the procedure described in Appendix 5 and is summarized in detail by Freire and Tinoco [5]). In Eq. 1, E_m is the mean Young Modulus at the operating temperature and S''_{ij} are the principal stress differences:

$$S_{alt} = \tfrac{1}{2} \times E_m \times \max\left\{S''_{12}, S''_{23}, S''_{13}\right\} \quad (1)$$

Fracture Mechanics and Crack Growth

Similar to the classical solid mechanics, fracture mechanics relates the crack geometry and size with some allowable quantity as stated Eq. (2):

$$Z(geometry, loads, crack\ size) \leq Z_c(material, temperature, load\ ratio) \quad (2)$$

In the context of this study, Z and Z_c in Eq. 2 can represent the mode I stress intensity factor (K_I) represented by the Eq 4 and the critical stress intensity factor, respectively.

$$K_I = Y \times \sigma \times \sqrt{\pi \times a} \quad (3)$$

Equation 3 is well suited for brittle materials. In turns BS-7910 goes further, since it uses the concept of FAD diagram, based on the R6 theory, which deals with both brittle fracture and plastic collapse. FAD (and all methodology described in BS-7910) is applied to cracked structures subjected to static or monotonic loading. However, under cyclic loading Paris et. al [6] proposed a useful tool to compute crack growth behavior through the equation (Eq. 4) which is known as a Paris Law

$$\frac{da}{dN} = C \cdot (\Delta K)^m \quad (4)$$

In Eq. 4, ΔK is the variation of stress intensity factor that can be computed through Eq. 3, varying the crack length a. C and m are material parameters determined experimentally. The linear part of this equation is useful for predict the crack growth.

METHODOLOGY

Material information

Based on the above theory, an assessment of the PSA vessel (Figure 2) was performed.

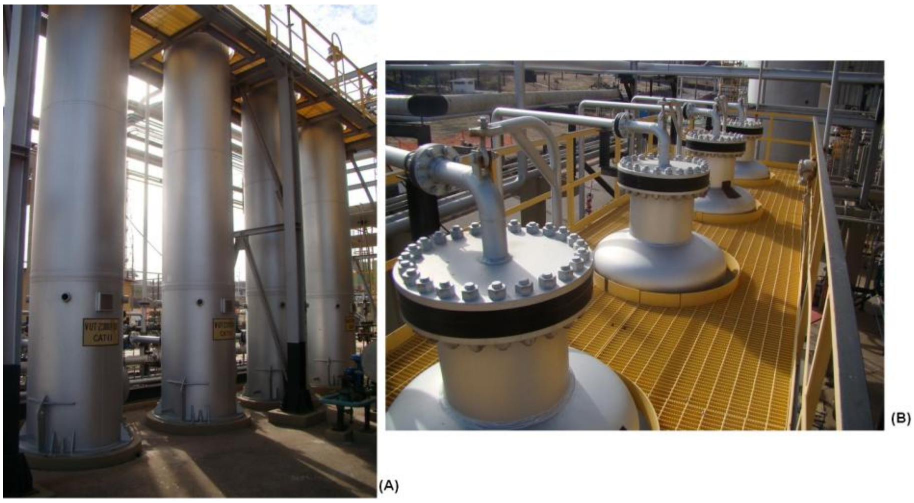

Figure 2: Analyzed PSA vessels.

The PSA equipment (Figure 2) is a carbon steel **SA-516 gr. 70,** vessel whose mechanical properties for the operational temperature **(40°C)** are: Young modulus (207,000 MPa), Poisson ratio (0.3), Yield stress (262 MPa) and Ultimate tensile stress (483 MPa), obtained from the ASME and data sheets.

Toughness is not obtained from tables. Instead, it is necessary to measure or calculate. This work employs the Lower Bound Method, described by the API 579 [7].

$$K_{IC} = \min\left\{110 MPa\sqrt{m}\,;36.5 + 3.084\exp\left[0.036\left(T - T_{ref} + 56\right)\right]\right\} \tag{5}$$

In Eq. 5, T is the evaluation temperature, **40ºC,** and T_{ref} is equal to **-6 °C,** obtained from the Impact Test Exemption curve (ASME Section VIII Division 1 – UCS-66) for the SA-516 gr. 70 and a thickness of **23.5 mm** (head thickness). Replacing all values in Eq. 5, we obtain $K_{IC} = \min\left\{110\, MPa\sqrt{m}\,;158\, MPa\sqrt{m}\right\}$ and then we select $K_{IC} = 110 MPa\sqrt{m}$, the maximum allowable toughness according to this methodology.

Material constants for using with Paris law was obtained from BS-7910 for steel in a marine atmosphere. Point 1: ΔK=0, m=3.42, C=1.72x10^{-13}; Point 2: ΔK=748 $N/mm^{-3/2}$, m=1.11, C=7.48x10^{-7}.

Finite element model and fatigue results

Stresses were calculated using a linear elastic analysis on an axisymmetric Finite Element model, implanted in the software Ansys Mechanical R. 11.0 SP1. Stresses were used for both crack assessment and fatigue crack growth evaluation. Figure 3 shows both the geometric model and some mesh details from the interest regions.

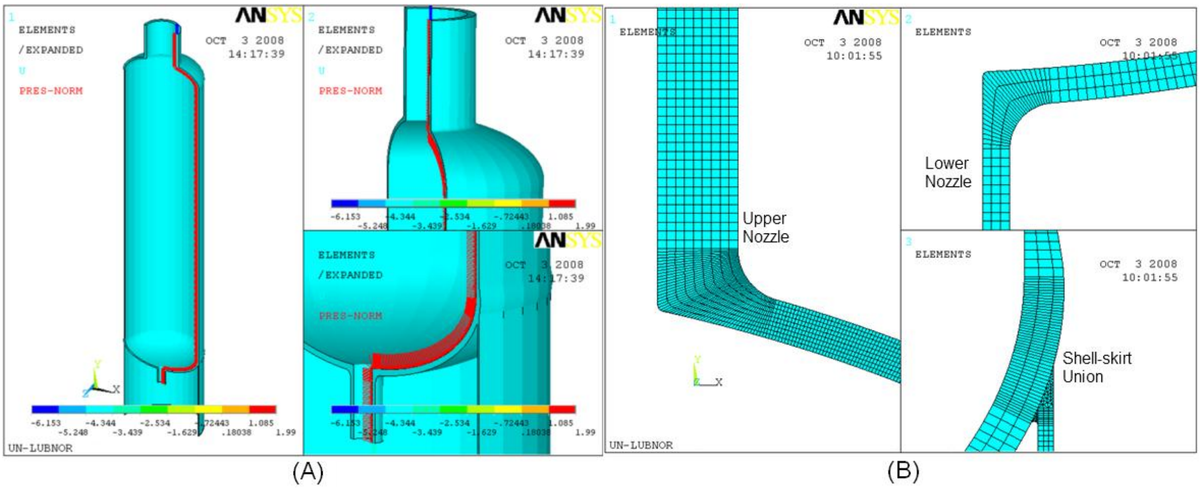

Figure 3: (A) Geometry; (B) Mesh details.

Stresses were computed for the upper nozzle and the principal stress differences are shown in Figure 4:

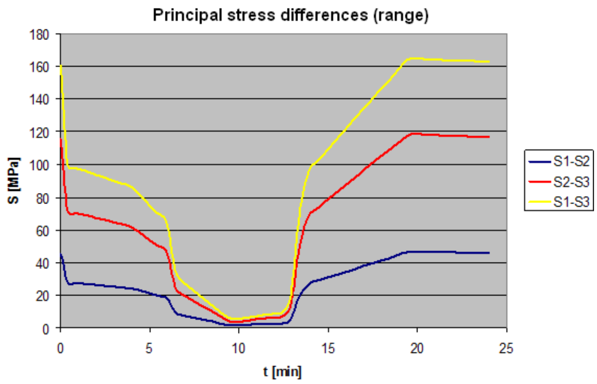

Figure 4: Principal stress differences for the upper nozzle.

Applying the multiaxial fatigue rule described before, we obtain an alternating stress **S_{alt}=85 MPa** (a half of the difference between **S_1-S_3=170 MPa**). Plotting this value in the graph shown in the ASME fatigue curve for carbon steel, we obtain **N = 10^6 cycles.** Considering a mean cycle of 24 min, we obtain a fatigue life of **45 years** (the equipment was originally designed for 20 years). However, this value does not consider the existence of defects in the structure.

Cracks evaluation

All cracks and were sized using the phased array ultrasonic technique. In Table 1, 2c, 2a and p are, respectively, the crack length, crack height and ligament.

Table 1. Size of discontinuities.

Discontiunuity	2c [mm]	2a [mm]	p [mm]
1	3,0	2,0	5,0
2	10,0	6,0	2,5
3	4,0	3,0	5,5
4	8,0	3,0	2,0
5	6,0	3,0	8,5
6	4,0	5,0	2,0
Lack of fusion	15,0	5,0	6,0

Crackwise v. 4.1.5616, computational implementation of BS-7910 was used for calculations (critical size and crack growth). Figure 5 shows the perpendicular tangential stress distribution nearby the cracks location.

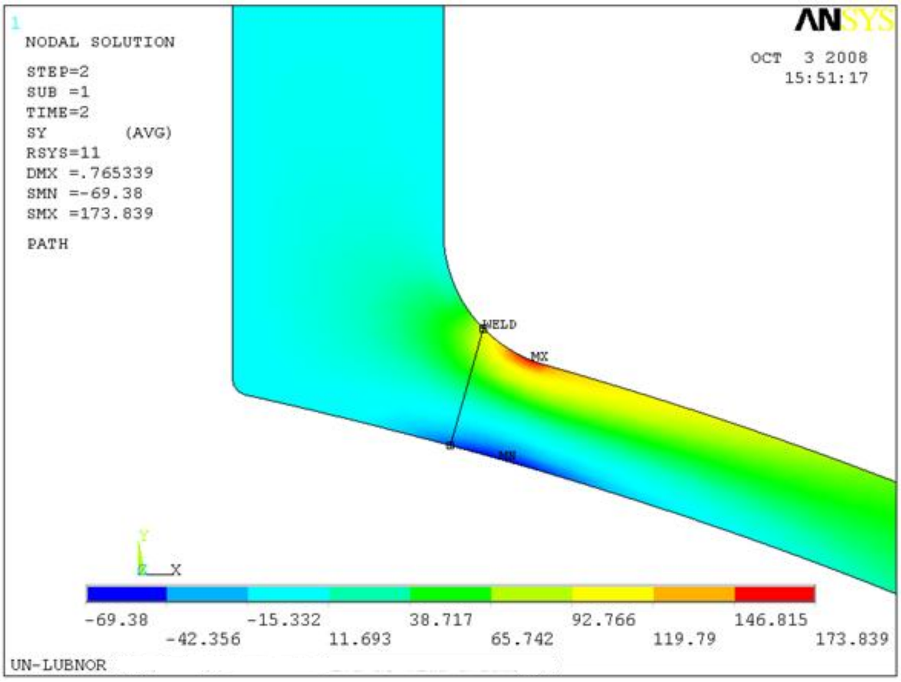

Figure 5: Perpendicular stress distribution in the crack site for the maximum pressure.

For obtaining the membrane and bending stress components linearization is necessary. Then we obtain the following stress values:

- For minimum internal pressure load case: **P_m = 0,96 MPa; P_b = 2,5 MPa**
- For maximum internal pressure load case: **P_m = 32 MPa; P_b = 82 MPa**

Applying the material, stress and crack geometric information to Crackwise, we obtain the criticality size, plotted in the FAD diagram, shown in Figure 6:

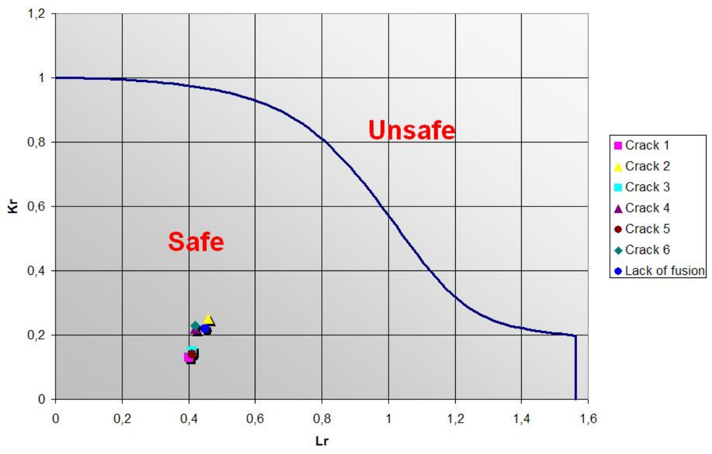

Figure 6: FAD for the assessed cracks.

Since all defects are held inside the FAD area, at the time they were evaluated they are considered safe.

Crack growth evaluation

As reported previously, the cyclic loads lead to a crack growth assessment. This equipment has been operating for 10 years. Considering the mean period of 24 min/cycle, it has already spent approximately **$2.2x10^5$ cycles.** Then we still have **35 years (780,000 cycles)** to operate, considering that the mean cycle does not change considerably. Again, applying the material data and the stress variation calculated before in the Crackwise, we obtain the number of cycles for each discontinuity reaches its critical size. This is shown in below:

Table 2. Number of cycles to make the embedded crack become a surface crack.

Defect	**1**	**2**	**3**	**4**	**5**	**6**	**Lack of fusion**
Cycles / time (yr)	780,000 / 35	39,000 / 2	698,000 / 31	44,000 / 2	780,000 / 35	53,000 / 2.5	780,000 / 35

The surface crack were there assessed again, and all were considered safe.

CONCLUSIONS

This work presents a practical application of several complex engineering concepts, such as fracture mechanics and stress analysis. The presence of defects greatly reduces the equipment life. Computations showed that the theoretical final life is about **2.5 years** (in the absence of defects, this fatigue life grows up to 45 years). Then, it was required to the refinery increase their inspections in the equipment in order to guarantee its structural integrity and reability.

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