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FLUID CATALYTIC CRACKING (FCC) RISER SUBMITED TO FLOW INDUCED VIBRATION FATIGUE LIFE ESTIMATION

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Abstract: Process Plants in petroleum refineries have several systems which are in general submitted to a range of vibration frequencies (varying from low values up to 20 Hz) and moderate amplitude (lower than 20 mm). There are several sources of vibrations such as unbalanced rotating machines, dynamic wind action, turbulent fluid-flow caused by valve opening and other more complex inner fluid-dynamics forces. The fluid-flow induced dynamic forces in piping systems is very difficult to predict and to prevent or even to foresee at design stage. They are normally associated to two-phase flow (vapor-liquid or gas-solid). In this article it is presented the preliminary results of the study of a practical problem that has occurred in a FCC riser, a long vertical piece of pipe, of the converter section of a Fluid Catalytic Cracking Unit (FCCU), where the catalytic reaction takes place. The temperatures ranges from 540 °C to 760 °C and the so called fluid which is in fact composed by solid particles (fine grain size), water steam and vaporized hydrocarbon, creates a piston effect that induces low frequency vibration (1 to 2 Hz) with amplitudes lower than 10 mm. To investigate this problem vibration measurement campaigns were performed in the riser, and vibration spectra were obtained for specific points and directions of interest. A finite element model of the structure was used to determine the dynamic characteristics of the riser (vibration modes and associated natural frequencies) and the dynamic responses in terms of member forces, stresses and displacements. With these results in the time domain a fatigue analysis was performed by using the rain flow method to compute stress cycles. The results of the structural analysis and the fatigue life estimation are presented and discussed. Conclusions and recommendations for extension of operating lifetime of the equipment are presented.

1 - INTRODUCTION

Fluid Catalytic Cracking Units (FCCU) are refinery process plants that converts heavy hydrocarbon feed (heavy atmospheric residue or gasoil) from distillation plants to light hydrocarbons (mainly automobile gasoline and LPG). Those heavy feed has low value while the light weight ones are high market value.

The heart of those units is the FCCU Convertor, with the following pieces of equipment: disengager, regenerator, stripper and, at last, the FCC riser which is a long vertical piece of pipe where the cracking reactions take place. Figure 1 shows a stacked FCC convertor, that is, the disengager, the striper and the regenerator are stacked one top of the other. The heavy feed is injected in the riser and mixed to the catalyst flow by special injector nozzles. The reaction is carried out along the vertical riser at temperatures that ranges from 760 ^{o}C up to 540 ^{o}C. The catalyst is a small grain size solid powder, fluidized by steam injected at several points along the riser as well as by the hydrocarbon feed.

The small solid particles (catalyst) are very hard and may erode in a couple of months the steel walls of the shell and tubular components. This is one of the reasons the Convertor components are lined by a special refractory material that has the two following purposes: protect from catalyst erosion and reduce the metal skin temperature to reasonable values that allow the use of carbon steel plates for fabricating the components, including the riser.

The presence of a fluidized amount of solid particles mixed with steam and vaporized hydrocarbon feed results in a piston effect that excites the FCCU convertor and affects particularly the slender thin-walled FCC riser. In reality what happens is the formation of big pockets with higher solid concentration followed by a pocket of steam and cracked hydrocarbon vapour flowing along the riser. This creates a pistoned flow that excites the FCC convertor in low vibration frequencies (1 to 2 Hz) with displacement amplitudes up to 15 mm.

As a consequence such vibration may cause leakage, rupture of auxiliary lines and, in an extreme case, cracks at the welded connections between the riser and its components. The steel frame structures (elevator shaft, catwalks and accessories) connected to the FCC shell structure also vibrate along with the convertor causing great concern and worries to the plant operators who feel uncomfortable and unsafe under such work condition.

The present article presents the study of fatigue evaluation at the FCC riser to the inner flow induced vibration. This riser is placed at the FCCU of Presidente Bernardes de Cubatão Refinery (RPBC), located in the state of Sao Paulo. A modal analysis was performed by using the ANEST and CAESAR II softwares and the obtained displacements, inner forces and stresses are presented and discussed. The fatigue evaluation was performed by using the developed software DTempo.

The RPBC FCC riser was chosen as a case example because a full set of vibration measurements were performed in the plant and the results were used to estimate the dynamic loading which was applied to the numerical modeling simulation.

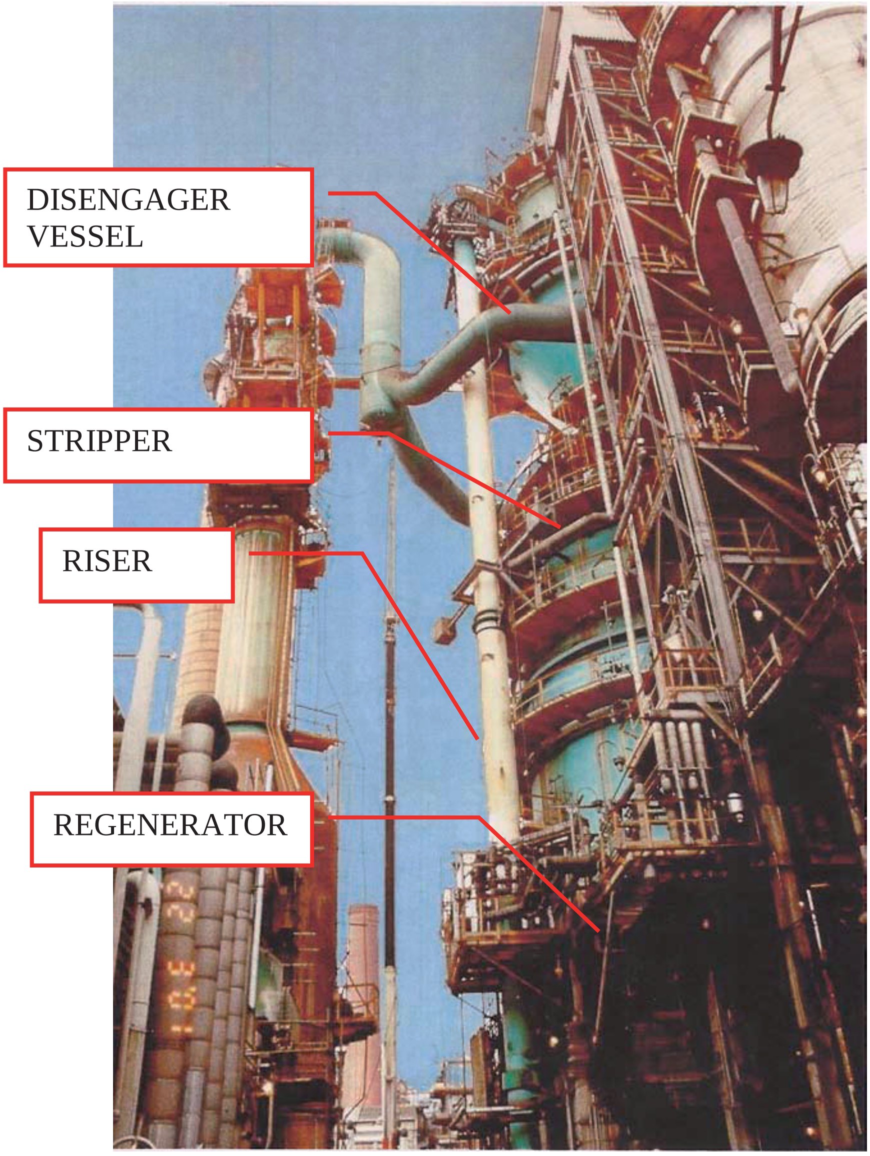


Figure 1 – FCC Convertor at RPBC refinery (left).

2 - DINAMIC ANALYSIS OF RISER

2.1 - Riser characteristics

The FCC risers at PETROBRAS refineries are “cold wall design” components, i.e, they are provided with an internal lining of 125 mm thick of cast vibrated refractory material, in order to protect against erosion to the steel riser pipe wall, as well as to reduce the metal temperature to values between 160 °C up to 230 °C. This inner protection makes it possible to use carbon steel plates to build the riser pipe and reduces the cost of construction and maintenance. The steel used in fabrication is the ASTM A 515 Gr. 70, a carbon steel with good mechanical strength.

The refractory material is high alumina concrete, reinforced with metallic fibers and anchored to the riser wall by V-shape stainless steel anchors welded to the riser wall. The thickness of the riser wall varies from 12,5 mm up to 31,5 mm, depending on the riser diameter and the quantity of connections welded.

The inner diameters vary from 1440 mm at the base of the riser up to 2180 mm at the top of the riser.

The riser is 40 m long and is connected to the bottom of the regenerator and to the side wall of the disengager. At the base of the riser there are two hinged expansion joints that absorb the thermal differential displacement between disengager/regenerator assembly and the riser.

2.2 - Experimental measurements

To perform the dynamic numerical analysis the main issue was to define the dynamic loading to be applied. Since the direct measurement of dynamic pressure loading on the shell wall was not feasible, the used approach was to measure the dynamic time response in terms of the vibration amplitudes of the shell wall and by doing a retro analysis to find out the dynamic pressure loading which would cause that measured response. The measurement was performed by installing accelerometers along the riser's height, especially at its base where the displacement amplitudes were higher. These experimental measurements were fundamental for understanding the structural dynamic behavior and for estimating the fatigue life of welding details of the riser. Figure 2 shows one of the accelerometers (covered with a thermal protection amianthus coat) installed at the base of the riser.

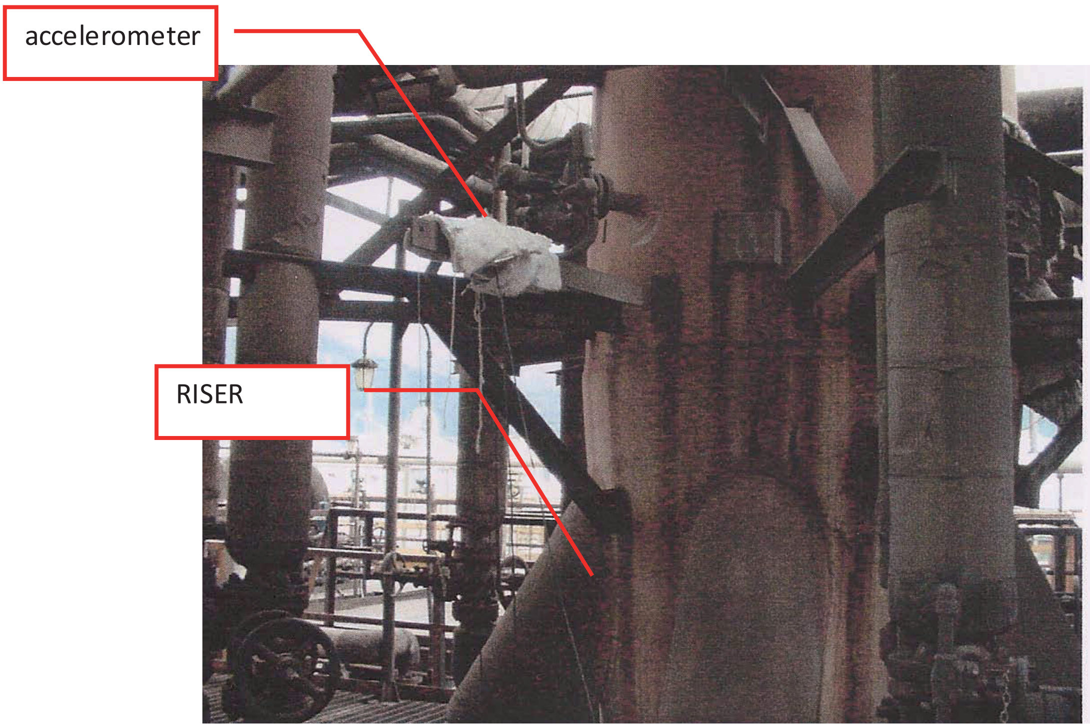


Figure 2 – Accelerometer at the riser base.

Figure 3 presents the experimental horizontal displacement x time signal, while Figure 4 presents the corresponding spectrum response. It can be observed that the response is dominated by the fundamental vibration mode with frequency f~1.07 Hz.

The feed injection at the base of the riser establishes a piston flow effect with frequencies around 1,05 Hz, extremely close to one of the natural frequencies of the riser pipe.

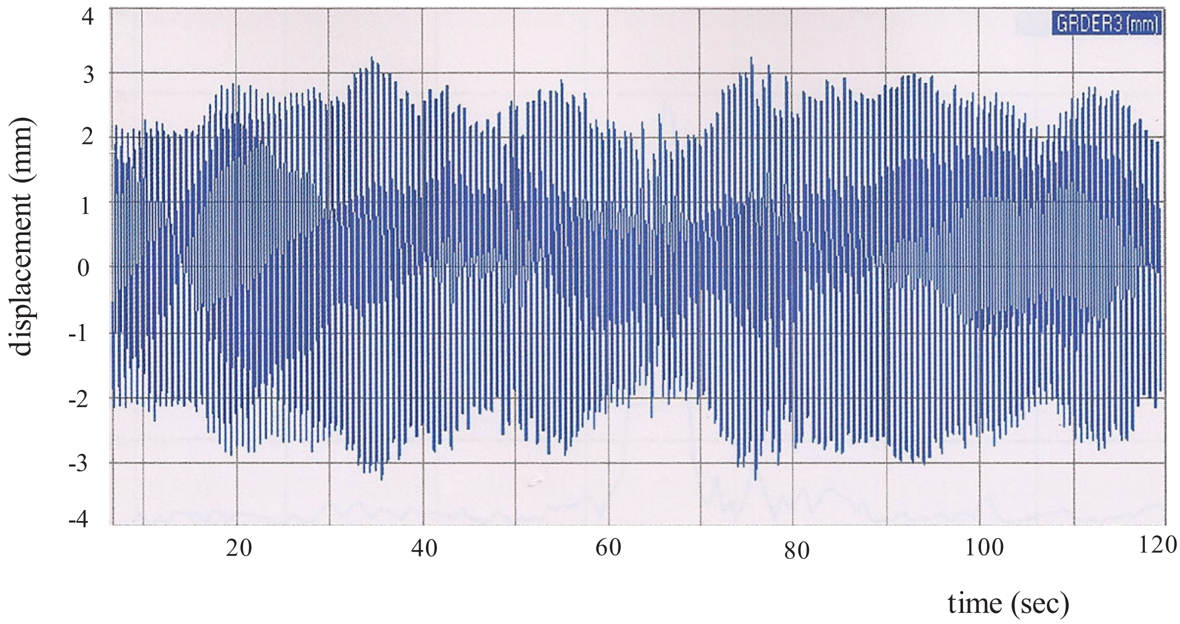


Figure 3 – Experimental horizontal displacement x time at the riser base

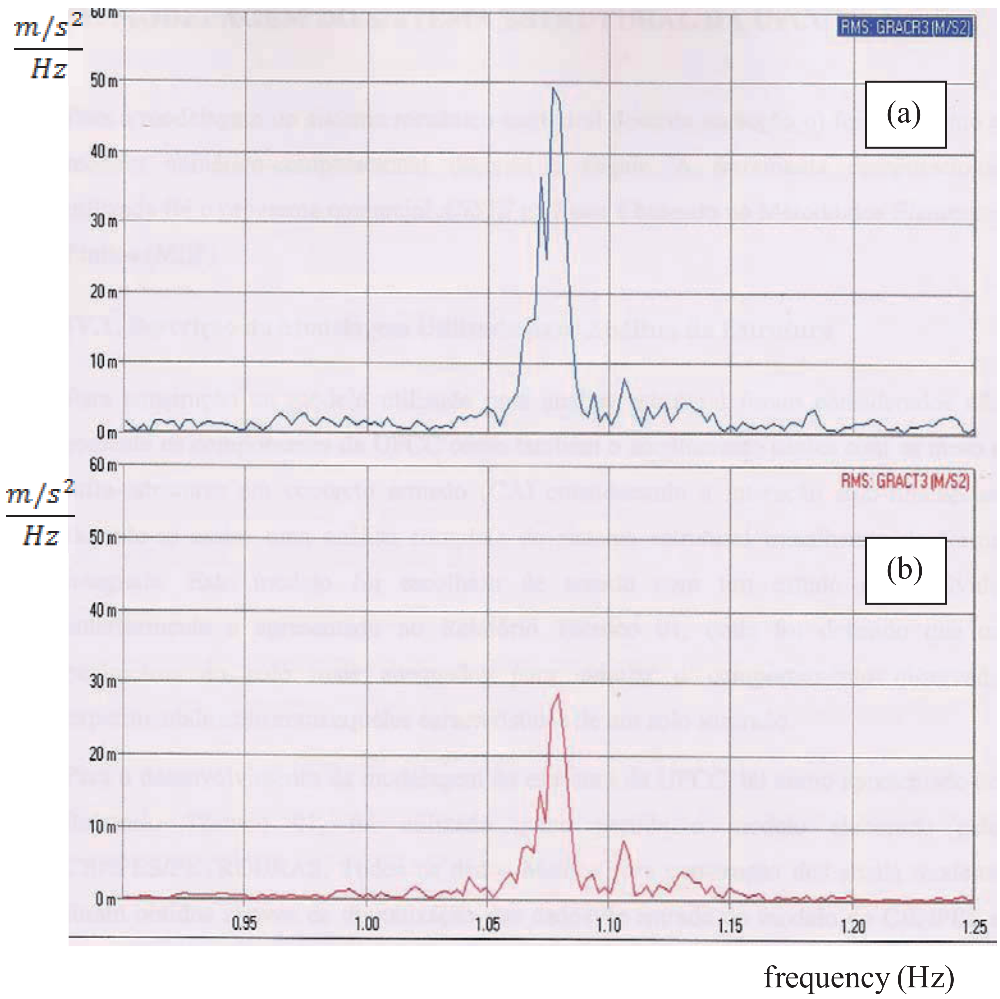
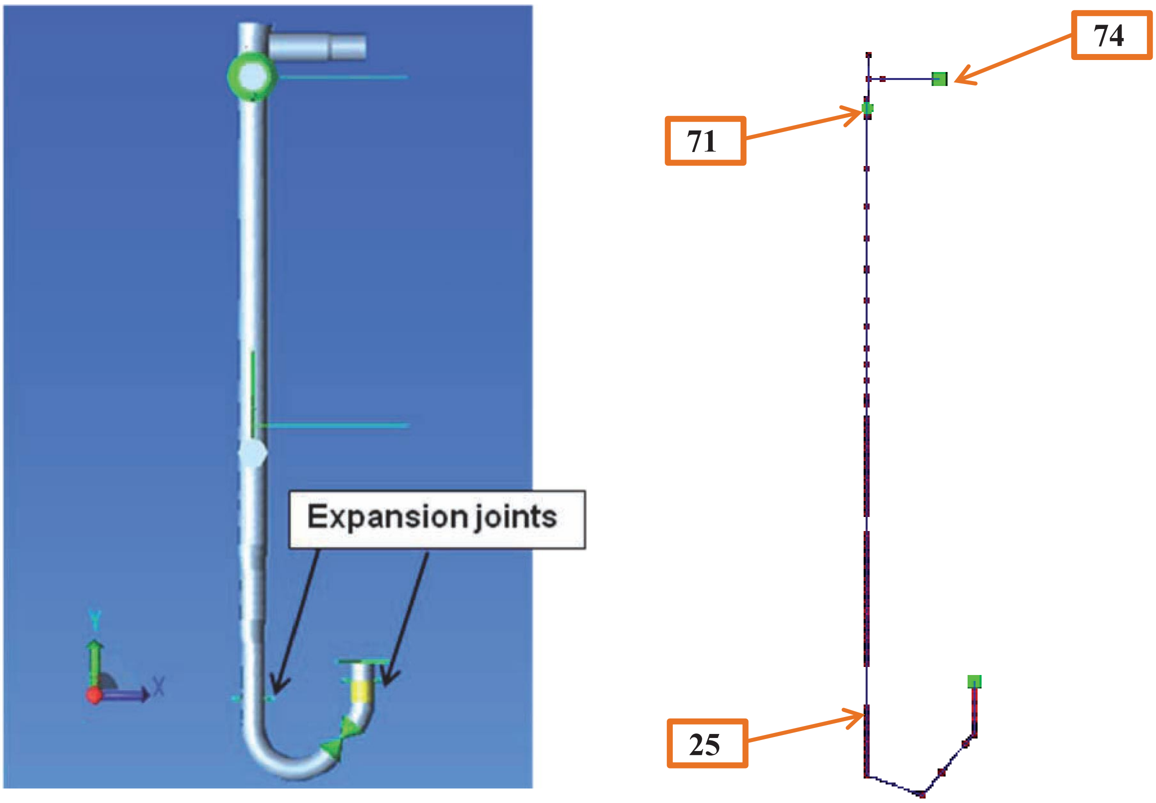


Figure 4 – Frequency spectrum of riser's response: (a) radial acceleration; (b) transversal acceleration.

2.3 - Numerical structural model

The numerical model was built with 76 (seventy six) pipe elements. There are also 2 (two) flexible elements to simulate the two hinged expansion joints installed at the base of the riser. The pipe elements are beam like elements with two end nodes and with circular section properties. The riser numerical model is shown in Figure 5(a) and the nodes distribution along the model and the node numbers of the three nodes selected for evaluation are shown on Figure 5(b).



(a) (b)

Figure 5 – Riser (a) schematics; (b) finite element model

To get good dynamic responses the structure's numerical model needs to be adjusted to simulate the actual riser characteristics. The main adjustment is for the refractory lining effect on the stiffness properties of the piping section; since the lining is very thick and has a high elastic module. The composite section flexural property may be calculated by using the expression (1) in which the equivalent bending stiffness EI of the combined section (steel pipe + refractory lining) is given as the sum of the bending stiffness of each part, neglecting the biding effect of the steel anchors:

$$(E \cdot I)_{EQ} = E_{STEEL} \cdot I_{STEEL} + E_{REF} \cdot I_{REF} \tag{1}$$

This expression is widely used to find composite section properties of components which are not binded together. The numerical model was further simplified by taking only the steel pipe cross-section but applying an equivalent elastic module of the composite section to get the bending stiffness $(EI)_{EQ}$, as given by expression(2). Although useful for modeling the riser (and of apparent common-sense because it is the steel pipe section that withstands the internal pressure and avoids leakage) this simplification has to be used carefully, since the stresses in the steel wall are to be used to check the allowable stress limit prescribed in the design codes.

$$E_{EQ} = \frac{E_{STEEL} \cdot I_{STEEL} + E_{REF} \cdot I_{REF}}{I_{STEEL}} \quad (2)$$

So, an equivalent elasticity module was calculated to each pipe section to be multiplied by the moment of inertia of the steel pipe cross section to get each individual bending stiffness.

The two hinged expansion joints were simulated by using link elements having high axial and translational stiffnesses and bending stiffness as given by the manufacturer.

While the bending stiffness of the hinged expansion joints has its value defined by the manufacturer, the axial and shear stiffnesses have to be estimated in such a way to restrain the axial and translation relative motions. They have to be given high numerical values which may represent rigid connections, without causing ill-conditioning in the structural stiffness matrix. For that same the equivalent stiffnesses have be estimated from a piece of pipe of the diameter of the connected run of pipe.

Table 1 shows the comparison between the estimated theoretical vibration frequencies for different numerical modelings and the value obtained from experimental measurement. Since the theoretical bending mode vibration frequency (f~1.05 Hz) resulted very close to the experimental one the model was considered ready for dynamic stress calculation. It can be noticed in Table 1 that the first four vibration modes of the riser have frequencies lower than dominant bending vibration frequency as shown in the frequency spectra of Figure 4. These four first modes are low frequency oscillation modes associated with rigid body motions of the riser pipe structure under the modeled existing support and connections conditions (i.e. its links with the disengage and regenerator vessels' structures). But these fundamental overall modes are not easily excited by the fluid-flow induced dynamic forces in piping system.

Mode	**Frequency (Hz)**			
	Theoretical values			**Experimental values(4)**
	Steel Pipe(1)	**Steel Pipe + Refractory lining (2)**	**Adjusted values(3)**	
1	0.388	0.337	0.079	
2	0.397	0.685	0.303	
3	0.655	0.963	0.616	
4	0.927	1.066	0.926	
5	1.050	1.274	1.047	1,075

Table Notes:
(1) Model with original pipe steel properties with no corrections;
(2) Included refractory lining stiffness corrections (expression 2);
(3) Adjusted values for expansion joints stiffnesses to reach numerical convergence;
(4) Experimental values according to field measurement.

Table 1 – Comparison between theoretical and experimental frequencies for the riser structure (Figs. 5)

The displacement x time response (see Figure 3) was applied as the exciting action in the numerical model. This displacement profile was applied at node corresponding to the measurement point (node 25). This effect is simulated applying an equivalent reaction node according to illustrated in Figure 6.

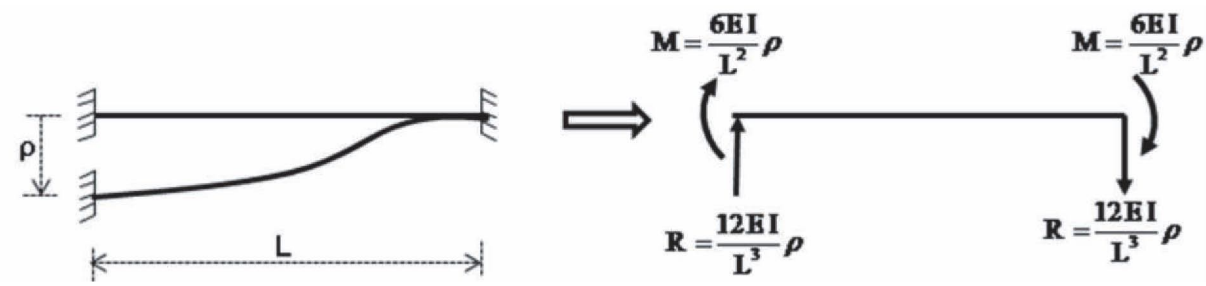


Figure 6 – Equivalent reaction node

The problem was solved in the time domain and the bending moment variation was calculated. The result on the welded cross-section corresponding to node 25 (see Figure 5(b)) is presented in Figure 7. The variation of stresses under axial force and bending moment was then calculated to perform the fatigue analysis.

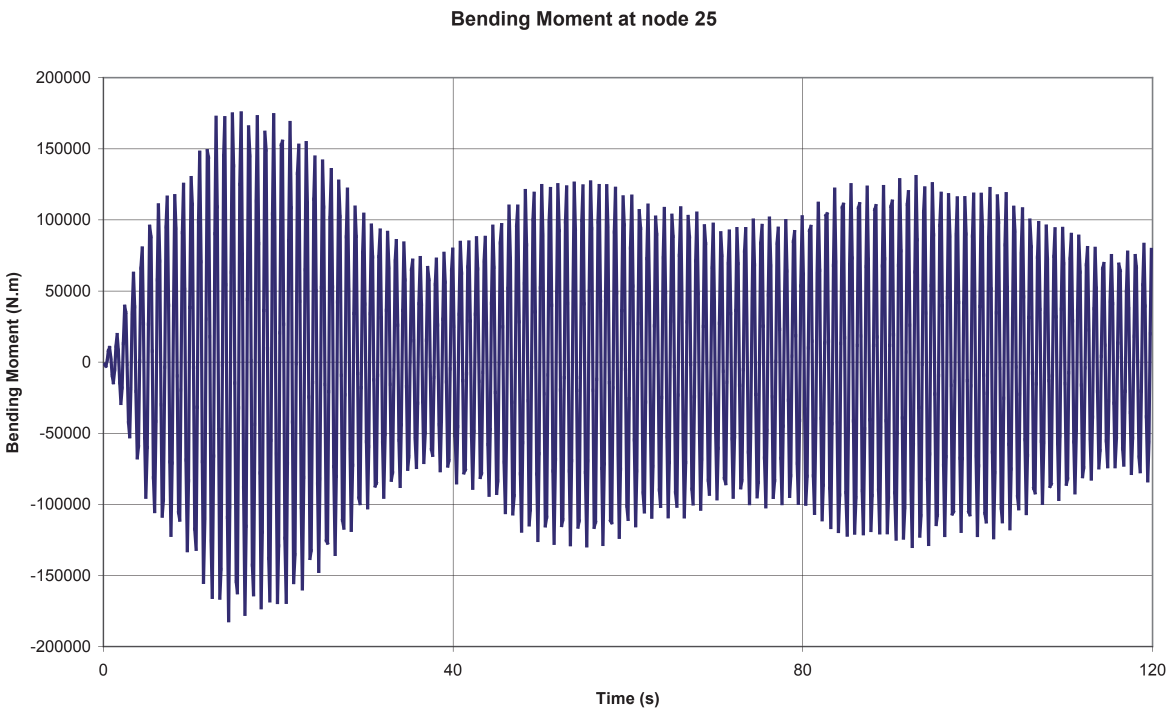


Figure 7– Bending Moment x time on the node 25.

3 - FATIGUE ANALYSIS

The Rainflow Method was used to count the number of cycles within the same stress range. The software used for fatigue analysis (Dtempo) was based on the algorithm proposed by Branco, C. F. (1986), using the vertical orientation graph (see Figure 8).

At the end of the counting and sum of cycles, the fatigue damage referred to each amplitude can be calculated and composed using the Palmgren-Miner rule, assuming that the damage in a certain element under any level of cyclic stress is cumulative:

$$D = \sum_{i=1}^{k} \frac{n_i}{N_i} \leq 1 \qquad (3)$$

where: D - Cumulative fatigue damage ratio;

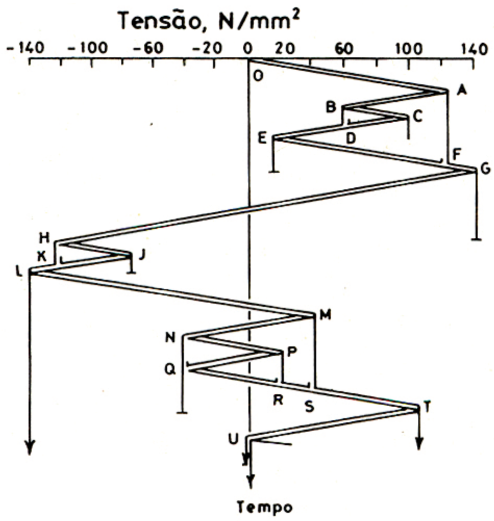
n_i - Number of cycles associated to stress level variation i;

N_i - Number of cycles to failure at a certain level of Stress S_i, obtained from the S-N curves.

This analysis takes into consideration that no effect due applied load sequence exists.

It is assumed that the damage failure occurs when the estimated cumulative damage reaches the unit. Hence, the structure lifetime is defined as the inverse of the total damage.

$$T = \frac{1}{D} \qquad (4)$$



Gama Tensão N/mm²	Meios Ciclos (+)	Meios Ciclos (−)	Total
20			0
40	BC, HJ	CD, JK	2
60	QR	PQ	1
80	NS	MN	1
100	EF	AE	1
120			0
140	OG		1/2
160			0
180			0
200			0
220			0
240			0
260			0
280			0

Figure 8– Rainflow method scheme (Branco, C. F., 1986).

3.1 S-N Curve

The S-N curves vary with the material specification. In this work it was used the curves form 2007 ASME Boiler and Pressure Vessels Code, that groups per material tensile strength. The material used in the fabrication of the riser is the steel ASTM A 515 Gr. 70. This steel has minimum tensile strength $S_U = 482{,}63$ MPa.

The corresponding S-N curve is presented in Figure 9.

The graph can be represented by the following expression:

$$N \cdot \Delta S_{range}^{h} = \left(f_{MT} \cdot C\right)^{1/h} \qquad (5)$$

where: N – Number of cycles;

ΔS – Cyclic Stress Range (MPa);

f_{MT} – Fatigue reduction factor due the temperature;

C – Coefficient given in Table 2 (Table 3.F.11 from 2007 ASME Boiler and Pressure Vessels Code);

h – Coefficient given in Table 2 (Table 3.F.11 from 2007 ASME Boiler and Pressure Vessels Code).

In the software used in this analysis the following expression was used:

$$N \cdot \Delta S_{range}^{h} = \bar{a} \tag{6}$$

with:

$$\bar{a} = \left(f_{MT} \cdot C \right)^{1/h}$$
$$f_{MT} = \frac{E_T}{E_{ACS}} \tag{7}$$

Where:

E_T – Modulus of elasticity of the material at the evaluation temperature = 203395.3 MPa;
E_{ACS} – Modulus of elasticity of the material at room temperature.

The metal skin temperature at the normal operating condition is T = 210 ^oC. So the corresponding metal property to be applied is E_{ACS} = 190.709 MPa, what results in f_{MT}=1,0665.

On the other hand, the coefficients C and h used in the life prediction are the most conservative ones (lower than 99% and prediction interval -3σ), what results in C = 11577.9 and h = 0.31950.

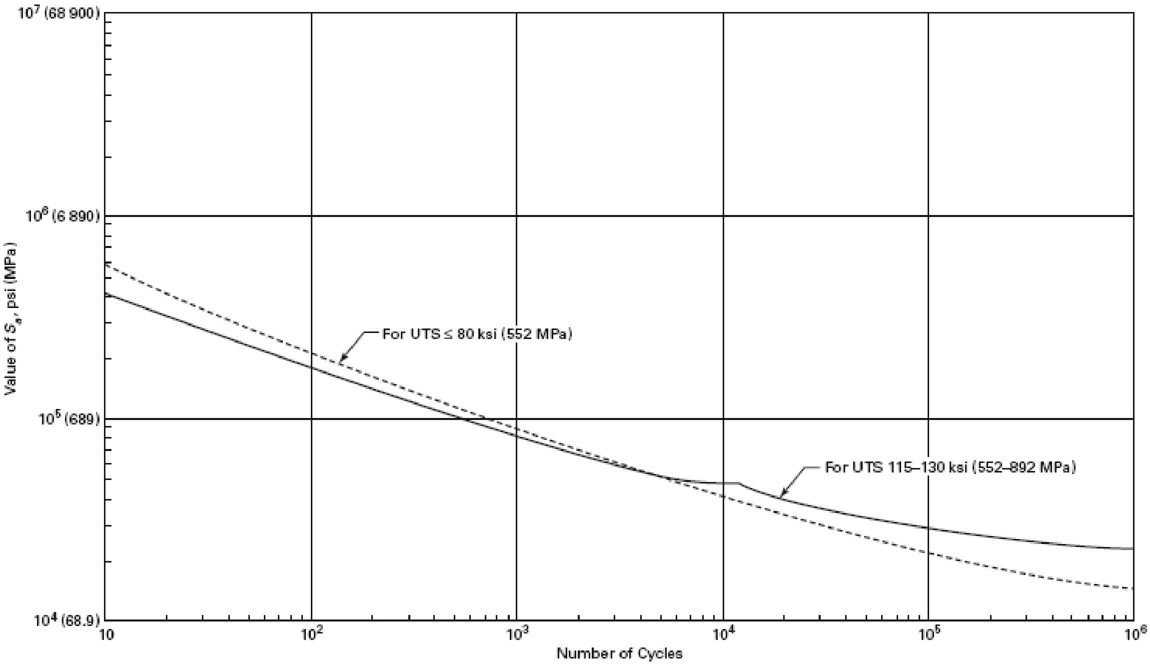


Figure 9– S-N curve for carbon steel used for pressure vessels and big diameter piping.

Statistical Basis	Ferritic and Stainless Steels		Aluminum	
	C	h	C	h
Mean Curve	19930.2	0.31950	3495.13	0.27712
Upper 95% Prediction Interval $(+1\sigma)$	23885.8	0.31950	4293.19	0.27712
Lower 95% Prediction Interval (-1σ)	16629.7	0.31950	2845.42	0.27712
Upper 95% Prediction Interval $(+2\sigma)$	28626.5	0.31950	5273.48	0.27712
Lower 95% Prediction Interval (-2σ)	13875.7	0.31950	2316.48	0.27712
Upper 99% Prediction Interval $(+3\sigma)$	34308.1	0.31950	6477.60	0.27712
Lower 99% Prediction Interval (-3σ)	11577.9	0.31950	1885.87	0.27712
Note: In SI Units, the equivalent structural stress range parameter, ΔS_{range}, in paragraph 3.F.2.2 and the structural stress effective thickness, t_{ess}, defined in paragraph 5.5.5 are in $MPa/(mm)^{(2-m_{ss})/2m_{ss}}$ and mm, respectively. The parameter m_{ss} is defined in paragraph 5.5.5.				

Table 2 – Coefficients for the welded joint fatigue curves (Table 3.F.11M from 2007 ASME Boiler and Pressure Vessels Code)

3.2 Analysis result

Three pipe sections were selected for stress analysis, corresponding to the nodes indicated in Figure 5(b). Those sections were selected because they have points of stress concentration. The stress responses for these three sections are presented in Figure 10-12. It can be observed that the results at section/ node 25 are the most critical. The stress levels presented on the reference graphs are already corrected to include the stress intensification factors for each section.

At section/node 25, there is a branch connection for the feed inlet nozzles of the riser. To consider the stress concentration at this discontinuity, stress concentration factors needed to be used. Two correction factors were used together. One of these is the wall stress intensification factor developed by Markl(1960), that was incorporated in appendix D of ASME B31.3/2008 – Process Piping Code and partially presented at Table 3.

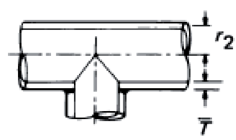
Description	Flexibility Factor, k	Stress Intensification Factor [Notes (2), (3)] Out-of-Plane, i_o	In-Plane, i_i	Flexibility Characteristic, h	Sketch
Unreinforced fabricated tee [Notes (2), (4), (9), (11)]	1	$\frac{0.9}{h^{2/3}}$	$\frac{3}{4}i_o + \frac{1}{4}$	$\frac{\bar{T}}{r_2}$	

Table 3 – Stress Intensification Factors (ASME B31.3 – 2008).

It should be observed that Markl (1960) established these factors based on fatigue tests results, performed in welded pipe fittings with strain gages properly installed. The stress intensification factors were obtained by comparison to a welded straight run of pipe tested under the same applied displacements. Then, it is necessary, for vibration fatigue analysis, to multiply the stress intensification factor by the butt weld stress concentration factor (Becht IV, C, 2006) $K_T = 2.0$.

The stress intensification factor for in-plane stress calculation can be determined by the following expressions:

$$\begin{aligned} i_i &= \frac{3}{4} \cdot i_O + \frac{1}{4} \\ i_O &= \frac{0{,}9}{h^{2/3}} \\ h &= \frac{\overline{T}}{r_2} \end{aligned} \tag{8}$$

Where:

$\overline{T}$ - Nominal thickness of the riser = 25 mm;

r_2 – Mean radius of the riser = 813 mm;

h – Flexibility factor;

i_O – Out-of –plane stress intensification factor;

i_I – In-plane stress intensification factor = 7.62/

$$k_i = i_I \cdot K_T = 15.24 \tag{9}$$

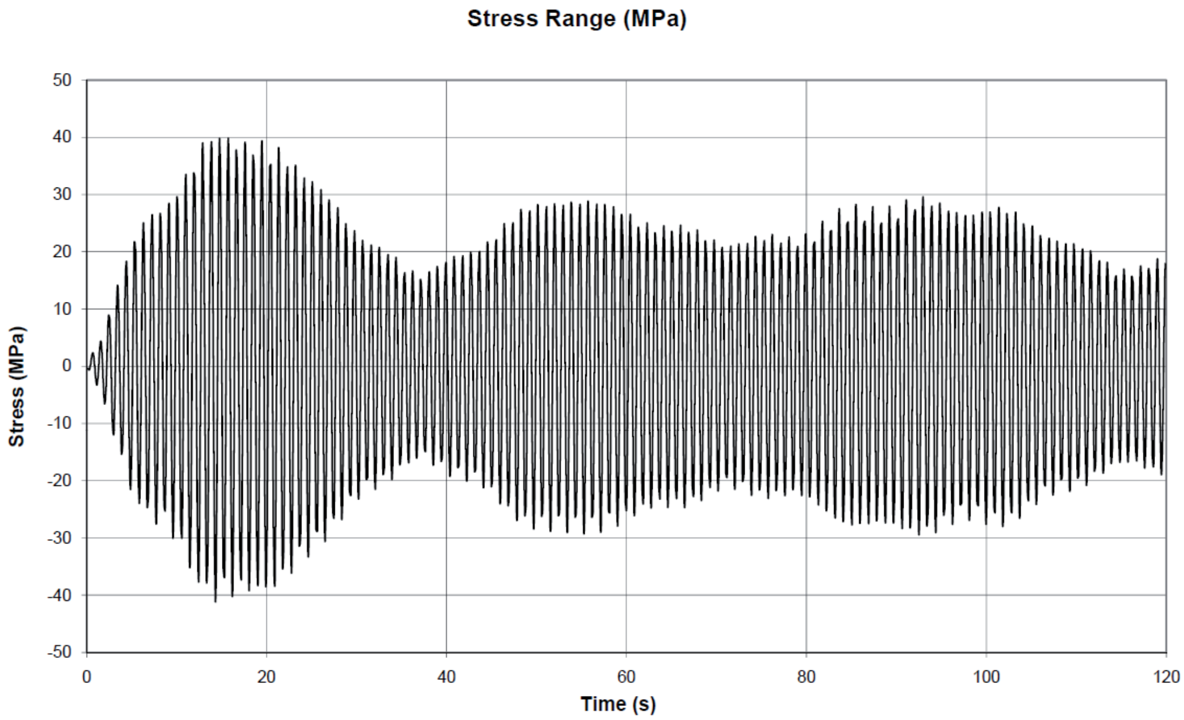


Figure 10– Stress variation with time for node 25.

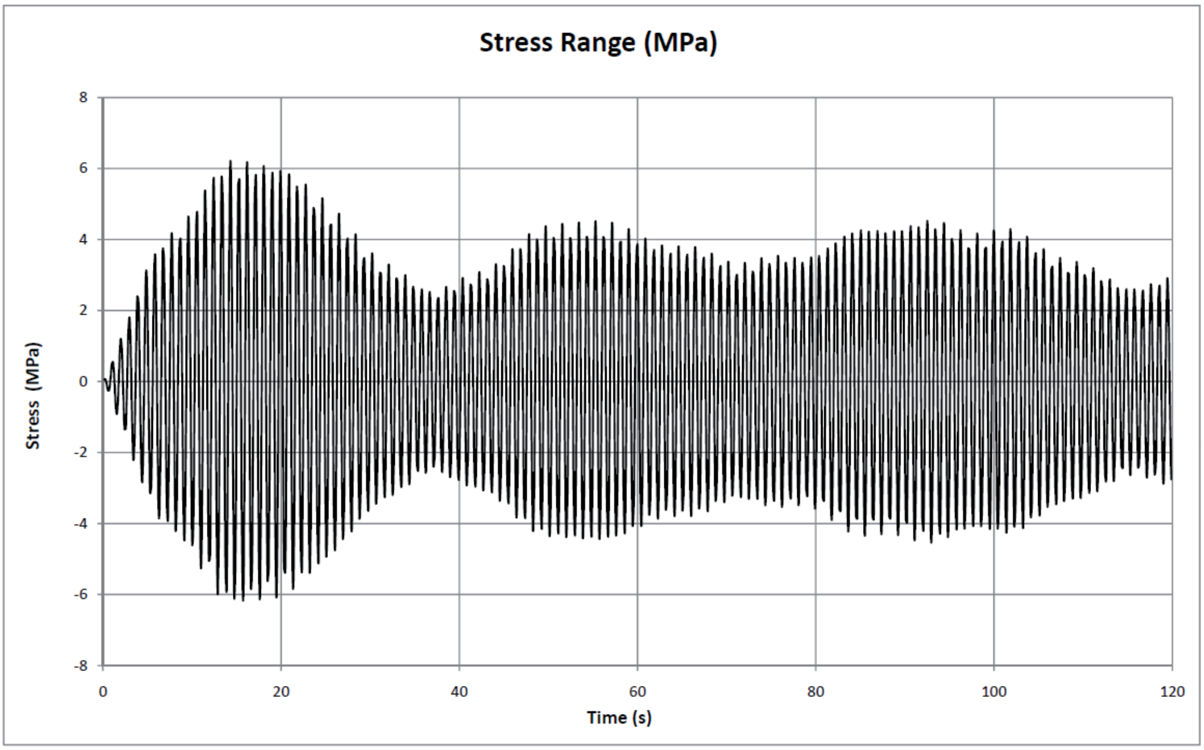


Figure 11– Stress variation with time for node 71.

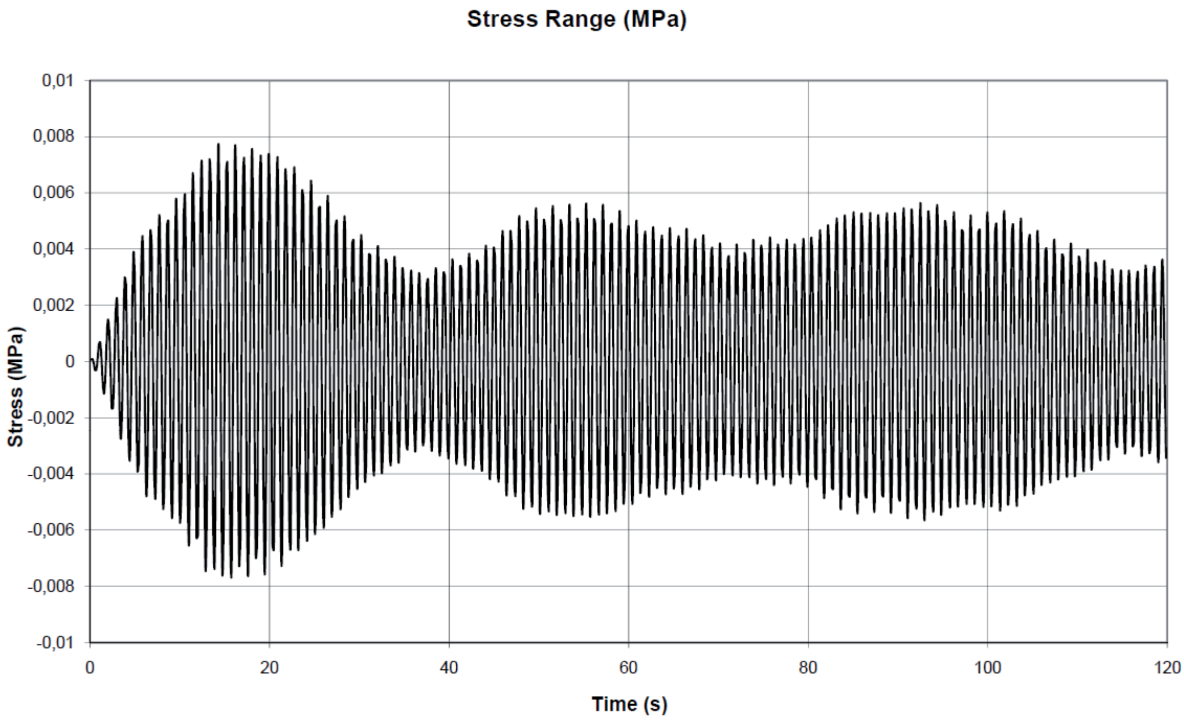


Figure 12– Stress variation with time for node 74.

For the conditions presented herein, the fatigue life for the pipe riser is infinite, since the displacement amplitude is relatively small (6.4 mm), to result in mechanical damage due to fatigue.

4 - CONCLUSIONS

The feed in let at the base of the riser is the source of pulse vibration on the riser. Since the vertical riser is quite long the rigid body frequencies are quite low, close the excitation frequencies imposed by the catalyst flow in the riser (~1 Hz).

The maximum vibration amplitude is around 6.4 mm, what is not enough to result in fatigue failure even considering the high concentration stress at certain points of the riser.

But, on the other hand, the low frequency vibration may cause problems on small branch piping connected to the riser as well as the operation team comfort and safety has to be evaluated and may justify means to control the vibration amplitude.

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