A CASE STUDY ON CORROSION-THERMAL FATIGUE DAMAGE

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ABSTRACT

Huge hoods, made by welding low-carbon water-cooled steel tubes to form the knee that initiates the pipeline used to collect the hot gases produced during steel refining in an electrical furnace, prematurely and repeatedly cracked in service. These short hood lives have been mathematically reproduced by joining straightforward thermal stress analysis, low-cycle fatigue and elastic-plastic strain concentration models with simple metallurgical techniques.

KEYWORDS

Corrosion-thermal fatigue interaction, multi-source damage, low-cycle thermal fatigue.

INTRODUCTION

Several important equipments can present cracking problems caused by corrosion-thermal fatigue interaction. Such multi-source damage accumulation problems normally are carefully modeled when designing high tech components, such as gas turbine blades e.g., but they are far from trivial to quantify. Indeed, the modeling of compounded damage usually requires a lot of advanced engineering expertise and sofisticated numerical techniques, a combination that can demand more than know-why information available in the literature. Such problems frequently need know-how as well, and seldom can be properly treated when analyzing more prosaic equipments, where the cost of expert advice can be economically unviable.

This work presents an exception to this rule, an ordinary corrosion-thermal fatigue damage problem that could be satisfactorily modeled by a relatively simple approach: a quite interesting case of premature (in 3 to 4 month intervals) and repeated cracking of huge steel furnace hoods (furnace hoods failure mechanisms are reviewed in [1]).

The model used to explain why the hoods lives were so short joins mechanical tools such as basic thermal stress analysis, simple finite element calculations, corrosion pits stress/strain concentration effects, and low-cycle fatigue concepts, with simple and reliable fractographic techniques. This interdisciplinary approach is easily manageable by well trained engineers, thus it can profitably complement the pure metallurgical techniques that are more commonly used in failure analysis. Moreover, which is far more important for the purpose of this paper, this approach can also be easily incorporated into the hood mechanical design procedures.

THE STEEL FURNACE HOODS

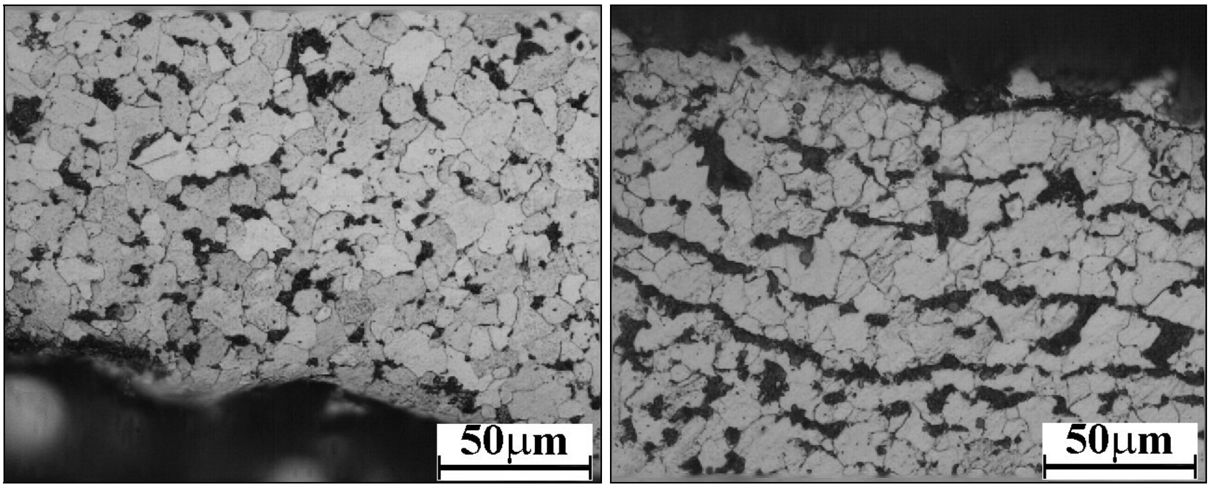
The hoods that cracked so prematurely in service collect the output gases of an electrical furnace used to produce steel. They are composed by several water-cooled low-carbon A106 Gr. B steel tubes bent and welded together to form a 2m diameter 70^o knee, the initial part of the pipeline that conducts the gases to their filter, see Fig.1. The tubes have external and internal diameters $\mathbf{D_e \cong 90mm}$ and $\mathbf{D_i \cong 74mm}$, and measured yield $\mathbf{S_Y}$ = **391MPa** and tensile $\mathbf{S_U}$ = **541MPa** strengths (well above the minimum ASTM standard requirements $\mathbf{S_Y}$ > **240MPa** and $\mathbf{S_U > 415MPa}$), and elongation ε_U = **26%** (which is within the **25-30%** required range). Water circulates inside those tubes for their indispensable refrigeration, since the hoods are not covered internally by a thermal insulation layer to save weight, as they must be repeatedly moved to allow the furnace to be charged and discharged.

Fig.1: A steel furnace hood, made by welding properly curved low-carbon steel tubes.

During steel production the gas temperature varies between **$150 < \Theta < 1500^{o}$C**, and the hood sustains around 2200 of such thermal cycles per month. The cooling water is plenty and does not vaporize, but there are hot spots on the tubes hot walls (it is convenient to describe the tube surface by separating it into three parts: the “hot wall” that faces the output gas, the water cooled “inner wall”, and the “cool wall” that faces the atmosphere.)
These hot spots are identified in cracked tubes cross sections by the important changes they induced in the originally banded microstructure of the A106 steel, which indicated that such points had operated for days at temperatures around 500^{o}C [1], see Fig.2-4. Indeed, the material near the cold and the inner walls maintains its original microstructure with elongated and banded perlite colonies, whereas the material near the hot wall presents a clearly recrystallized microstructure, with relatively small grains. This indicates that the tube hot wall temperature in service was much higher than the service temperature of the water-cooled inner wall and of the cold outside wall, but not high enough to promote significant grain growth.
Note also that both the tubes hot and inner walls suffer corrosion during operation, characterized by some uniform thickness loss and by many rounded pits several grains deep.

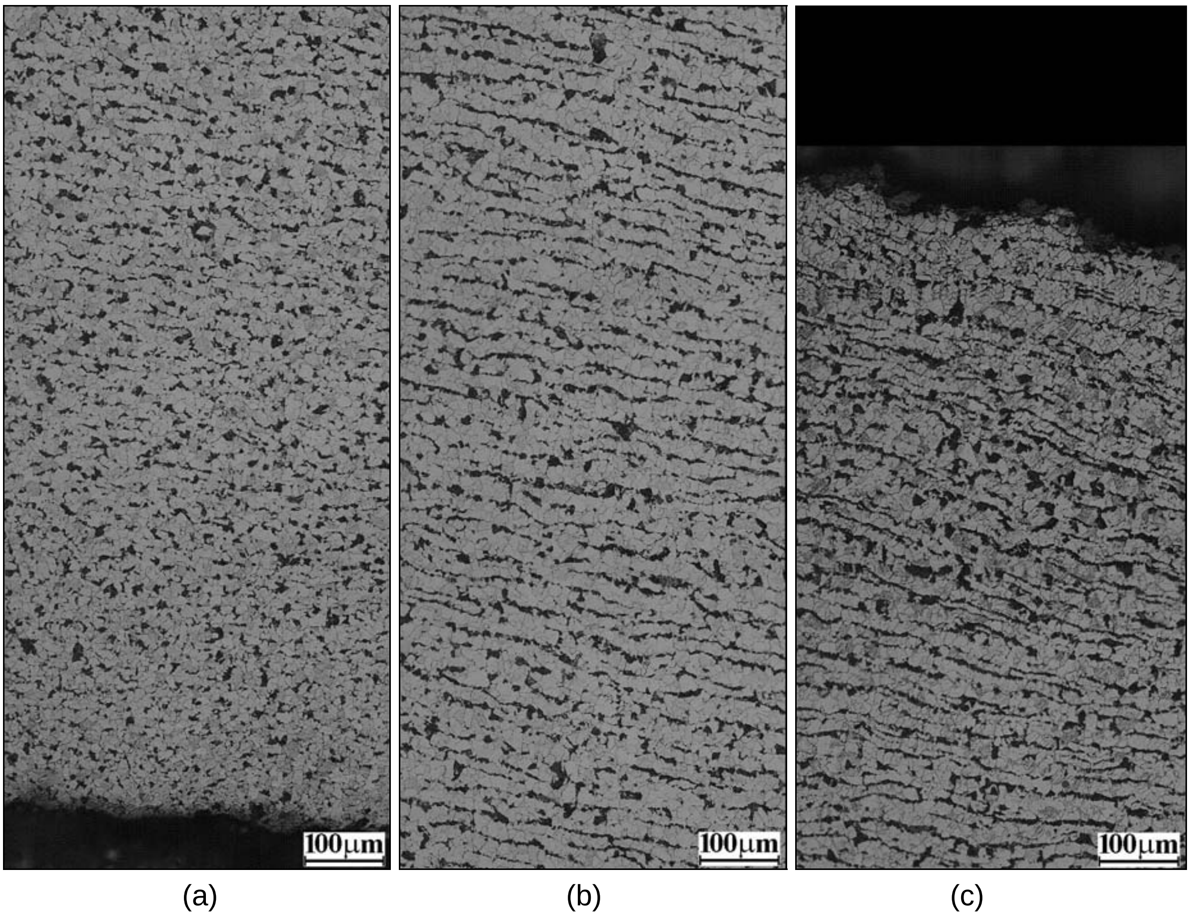
(a) (b)

Fig.2: Sample of a cracked hood tubular wall (a) cold part, which faced the atmosphere, and (b) hot part, which faced the furnace output gases. Note the corrosion with several pits on this hot wall surface and also on the water cooled wall surfaces, inside the A106 steel tubes.



(a) (b)

Fig.3: (a) Microstructure near the hot wall, showing recrystallized grains; and (b) originally banded microstructure of the A106 steel near the inner wall, see Fig. 4.



(a) (b) (c)

Fig.4: Steel microstructure along the wall thickness: (a) hot wall, (b) middle and (c) inner wall.

Therefore, the temperature gradient $\Delta\Theta$ associated with the tube hot points is quite steep, since the cooling water and the tube inner wall are maintained under 100°C at only 8mm from them. The linear elastic thermal stresses generated by $\Delta\Theta$ in the tube walls can be analytically calculated for axi-symmetric tubes of internal and external radii $\mathbf{R_i}$ and $\mathbf{R_e}$ subjected to (constant) external and internal wall temperatures $\Theta_\mathbf{i}$ and $\Theta_\mathbf{e}$ [2], and are given by:

$$\sigma_\theta(r)=\frac{\alpha E\Delta\Theta}{2(1-\nu)\ln(R_e/R_i)}\left[1-\frac{R_i^2(R_e^2+r^2)}{r^2(R_e^2-R_i^2)}\ln\left(\frac{R_e}{R_i}\right)-\ln\left(\frac{R_e}{r}\right)\right]$$

$$\sigma_r(r)=\frac{\alpha E\Delta\Theta}{2(1-\nu)\ln(R_e/R_i)}\left[\frac{R_i^2(R_e^2-r^2)}{r^2(R_e^2-R_i^2)}\ln\left(\frac{R_e}{R_i}\right)-\ln\left(\frac{R_e}{r}\right)\right] \tag{1}$$

$$\sigma_l(r)=\frac{\alpha E\Delta\Theta}{2(1-\nu)\ln(R_e/R_i)}\left[1-\frac{2R_i^2}{R_e^2-R_i^2}\ln\left(\frac{R_e}{R_i}\right)-2\ln\left(\frac{R_e}{r}\right)\right]$$

where α is the linear expansion coefficient (for steels about α = **12μm/m/°C** at room temperature), **E** is Young's modulus and ν is Poisson's coefficient. Therefore, for axi-symmetric tubes that exchange heat in steady state under $\Delta\Theta = \Theta_e - \Theta_i$ and have $R_e = \beta R_i$, the maximum and minimum thermal stresses act in their internal and external walls surfaces, and are given by:

$$\sigma_\theta(R_i)=\sigma_l(R_i)=\frac{\alpha E\Delta\Theta}{2(1-\nu)\ln\beta}\left[\frac{2\beta^2\ln\beta}{\beta^2-1}-1\right]$$

$$\sigma_\theta(R_e)=\sigma_l(R_e)=\frac{\alpha E\Delta\Theta}{2(1-\nu)\ln\beta}\left[\frac{2\ln\beta}{\beta^2-1}-1\right] \tag{2}$$

As the hood tubes have β = **45/37** = **1.216**, if they were axi-symmetric, these extreme thermal elastic stresses for small $\Delta\Theta$ (where **E** and α can be assumed constant) would be given by $\Delta\sigma_\theta(R_i)$ = **1.80**$\Delta\Theta$ and $\Delta\sigma_\theta(R_e)$ = **–1.58**$\Delta\Theta$ (σ in MPa and $\Delta\Theta$ in °C). Thus a temperature drop of $\Delta\Theta$ = **100°C** across the tube wall would cause a $\Delta\sigma$ **= 180MPa** (tractive) stress range at its inner cooled wall and a $\Delta\sigma$ = **–158MPa** (compressive) stress range at its hot outside wall. Or else, since the minimum yield strength required from the A106 steel is S_Y = **240MPa**, to maintain a nominally elastic stress range $\Delta\sigma = 2S_Y$ in the cooled wall surface, the maximum allowed thermal cycle across the tube wall should be $\Delta\Theta$ = **480/1.80** = **267°C** in this case.

But the hood tubes are not axi-symmetric, and thus require a much more elaborated thermo-elastic stress analysis that can be, however, approximated by a simpler model which obeys their contour conditions: a high temperature Θ_{eh} at the external hot half wall facing the output gases, a colder temperature Θ_{ec} at the other half facing the atmosphere, and a temperature Θ_i at the water-cooled inside wall. This simple model can be easily solved in any good finite element code, as long as a few precautions are taken (such as to use a tube long enough to avoid undesirable numerical noise at its ends, and to recognize that the elastic modulus varies significantly in the required temperature range [3].) The extreme calculated values for the biaxial stresses $\sigma_{\theta i} \cong \sigma_{li} = \sigma_i$ and $\sigma_{\theta e} \cong \sigma_{le} = \sigma_e$ and for the corresponding Mises stresses σ_{Mi} and σ_{Me} (which must have an assigned signal for fatigue analysis) are given in Table 1.

Table 1: Maximum and minimum stresses at the internal and external tube surfaces.

Θ_{ec} (°C)	Θ_i (°C)	Θ_{eh} (°C)	σ_i (MPa)	σ_{Mi} (MPa)	σ_e (MPa)	σ_{Me} (MPa)
80	100	500	363	364	–315	(–)290

FATIGUE ANALYSIS

This analysis assumes that: (i) the temperatures of the tubes walls cyclically vary between the water temperature, when the furnace is being loaded, and the extremes given in Table 1, during the steel manufacturing process; (ii) the cyclic mechanical properties can be estimated by the median rule [4], see Fig.5; and (iii) the corrosion pits are rounded and not too deep, thus they induce a stress concentration factor between say $2 < K_t < 3$ [2]. Note that the maximum thermal stresses that cause the fatigue cracking oscillate between **0** and σ_{Mi} (therefore, the nominal fatigue loading is given by $\sigma_a = \sigma_m = \sigma_{Mi}/2$ at the inner tube surface).

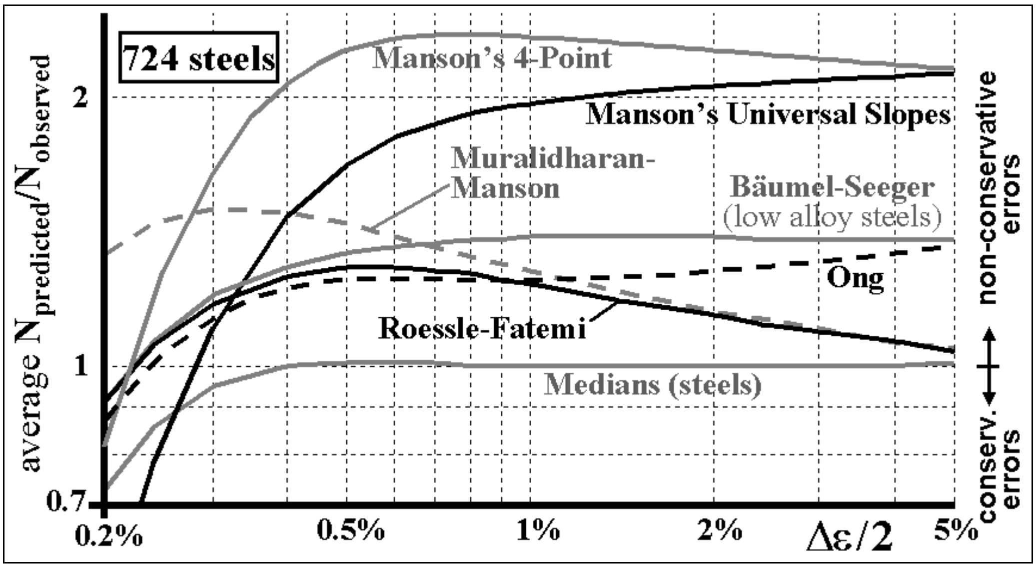


Fig.5: Average ratios (life predicted)/(life calculated from measured data) obtained by several estimation methods for 724 steels, under strain amplitude levels between 0.2% and 5% [4].

In the absence of reliable experimental data, the median estimate for steels gives:

$$\Delta\varepsilon = \mathbf{(3S_U/E)(2N)^{-0.09} + 0.9(2N)^{-0.59}}, \ \mathbf{h_c = 0.153}, \ \mathbf{H_c = 1.69S_U/E} \tag{3}$$

The elastic analysis used here for calculating the nominal thermal stresses is valid, since in the inner surface $\Delta\sigma_{Mi} < 2S_Y$. However, $\Delta\sigma_{Mi}$ is close to the elastic range $2S_Y$, and in these cases Neuber's rule should be used in fatigue analysis using Ramberg-Osgood to describe both the nominal and the root stress/strain ranges. This practice avoids the logical inconsistency and also the absurd predictions (such as notch stresses lower than nominal stresses) that can be made when using the traditional hookean description of the nominal stresses [5]. It is also necessary to correct the predicted elastic-plastic hysteresis loops to reflect all the sequence effects associated with the loading order, as done in the ViDa software [6], the tool used to make all the calculations required for the predictions presented below. But such results can be calculated by hand in this simple constant amplitude load problem. When describing the stress/strain concentration effect of the corrosion pit by Neuber, the pit root stress $\Delta\sigma$ and strain $\Delta\varepsilon$ ranges are obtained from the nominal stress range $\Delta\sigma_n$ by:

$$\Delta\sigma^2 + \frac{2E\Delta\sigma^{(h_c+1)/h_c}}{(2H_c)^{1/h_c}} = K_t^2 \cdot \left(\Delta\sigma_n^2 + \frac{2E\Delta\sigma_n^{(h_c+1)/h_c}}{(2H_c)^{1/h_c}}\right), \ \Delta\varepsilon = \frac{\Delta\sigma}{E} + 2\left(\frac{\Delta\sigma}{2H_c}\right)^{1/h_c} \tag{4}$$

And, when using Molsky-Glinka to describe the stress/strain concentration effect, by:

$$\Delta\sigma^2 + \frac{4E\Delta\sigma^{(h_c+1)/h_c}}{(1+h_c)(2H_c)^{1/h_c}} = K_t^2 \cdot \left(\Delta\sigma_n^2 + \frac{4E\Delta\sigma_n^{(h_c+1)/h_c}}{(1+h_c)(2H_c)^{1/h_c}}\right), \ \Delta\varepsilon = \frac{\Delta\sigma}{E} + 2\left(\frac{\Delta\sigma}{2H_c}\right)^{1/h_c} \tag{5}$$

Then the fatigue damage can be finally estimated by Coffin-Manson, but this equation probably over-estimates the life since it does not recognize the mean load effect. It is much more reasonable to use Morrow or Smith-Watson-Topper to estimate the fatigue life in this case, but the first load event must be separated from the others to quantify the mean stress at the pit root. For example, using Neuber, SWT, a stress concentration factor $K_t = \mathbf{2.5}$ for the (supposed approximately semi-spherical) corrosion pit, and a hot wall temperature $\Theta_{eh} = \mathbf{500^oC}$, which causes a nominal Mises stress range at the inner water-cooled wall $\Delta\sigma_{Mi}$ = **364MPa** and induces the stress/strain loop shown in Fig.6 at the pit root, the estimated fatigue life turns out to be **N** = **7300 cycles**, or a little more than 3 months. Needless to say, certainly not by coincidence or fortune, this is exactly the lifecycle of the hoods.

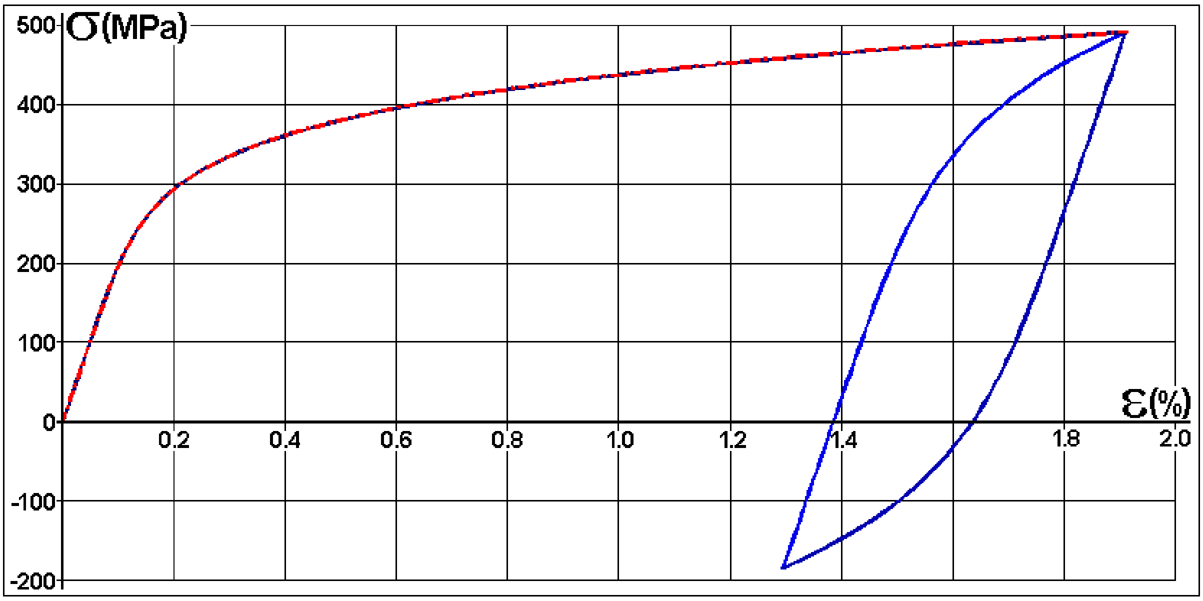


Fig.6: Stress/strain loops at a pit root predicted by Neuber and Ramberg-Osgood for $\mathbf{K_t}$ = **2.5** and a nominal stress range $\Delta\sigma_{Mi}$ = **364MPa**, caused by a hot wall temperature Θ_{eh} = **500°C**.

The main advantage of this modeling is its versatility to answer important designer questions. For example, if the corrosion is eliminated and $\mathbf{K_t}$ = **1**, the estimated fatigue life increases for **N** = **670000 cycles**, a remarkable improvement. Or if the hot wall temperature decreases to Θ_{eh} = **400°C**, then the nominal Mises stress range at the water-cooled wall also decreases to $\Delta\sigma_{Mi}$ = **273MPa**, and for $\mathbf{K_t}$ = **2.5** the estimated live increases to **N** = **32800 cycles**.

CONCLUSIONS

The simple hybrid model successfully used in this paper to evaluate the combined effect of corrosion and fatigue damage in steel furnace hoods can be easily adapted to model similar problems, confirming not only the usefulness of these simple but sound tools, but also the fundamental importance of the nowadays somewhat old-fashioned engineering feeling.

REFERENCES

[1] R.E. Hargraave, D.J. Kotwica, “A Review of Common Failure Mechanisms in BOF Hoods”, Water & Process Technologies, 2005.
[2] S.P. Timoshenko, J.N. Goodier, Theory of Elasticity, McGraw-Hill 1970.
[3] K.M. Ralls, T.H. Courtney, J. Wulff, Introduction to Materials Science and Engineering, Wiley 1976.
[4] M.A. Meggiolaro, J.T.P. Castro “Statistical Evaluation of Strain-Life Fatigue Crack Initiation Predictions”, International Journal of Fatigue, v.26, n.5, pp.463-476, 2004.
[5] M.A. Meggiolaro, J.T.P. Castro “Evaluation of the Errors Induced by High Nominal Stresses in the Classical εN Method”, in Fatigue 2002 v.2, A.F. Blom ed., p.1451-1458, EMAS 2002.
[6] www.tecgraf.puc-rio.br/vida