EFFECT OF MEAN LOAD ON THE FATIGUE LIFE OF STUDLESS CHAIN LINKS

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Abstract. The integrity of chain link mooring lines is a critical point for the safety of semi-submersible offshore platforms. Production offshore units have a relative long operational life (about 20 years), during which are submitted to the ocean adverse environment loading produced by the combination of wind, waves and currents. This complex loading history can promote the nucleation and propagation of cracks in mooring line components. The presence of defects establishes a critical situation that can lead to catastrophic failures. Traditional design methodologies, as the semi-empirical *TN* method, are based on the load range that the chain link experiments in operation. These simple methodologies do not consider effects as residual stresses and mean stresses that can considerable reduce the fatigue life of the component. In this work, a methodology for chain link life prediction based on the *SN* method is developed. To establish a relationship between the applied load and the stress distribution two models are developed: a linear-elastic parametric model, based on the energy method, and an elastoplastic finite element model. Results of fatigue life predictions obtained with both methodologies suggest that for situations where high values of mean load are present the *TN* methodologies could estimate unrealistic large fatigue lives.

Keywords: *Chain Link, Fatigue, Finite Element, Modeling, Residual stresses*.

1. INTRODUCTION

Production offshore units have a relative long operational life (about 20 years), during which are submitted to ocean adverse environment loading produced by the combination of wind, waves and sea currents. This complex loading history can promote the nucleation and propagation of cracks in mooring line components that can lead to catastrophic failures. The failure of a single element in a mooring line of an offshore oil exploitation platform can produce incalculable environment damage as well as human and material losses (Chaplin *et al*., 2000; Papazoglou, *et al.*, 1991; Paiva, 2000).

Chain links, as other offshore mooring line components, must be submitted to a mandatory proof test dictated by offshore standards. During the proof test, where loads higher than the operational load are applied to the mechanical component, extensive plasticity occurs resulting in high levels of residual stresses. Traditional design and inspection methodologies are very conservative (API, 1995; Hasson and Crowe, 1988; Stern and Weatcroft, 1978). Simple design methodologies, where effects like the presence of residual stress fields prior to operation are not considered, are extensively applied. Meanwhile, it is well known that residual stress plays a preponderant part on the structural integrity of a mechanical component, especially in nucleation and propagation of cracks (SAE, 1988, Almer *et al.*, 2000; Webster and Ezeilo, 2001). Tensile residual stresses can be especially dangerous since fatigue cracks usually propagate in the presence of tensile stress fields. During operation, tensile stresses promoted by operational loading acts together with residual stresses resulting in much higher tensile stress levels than the ones predicted by traditional design methodologies. Therefore, a detailed description of the residual stress field is required for an accurate assessment of mechanical component integrity (Pacheco *et al.*, 2002, 2003*a*, 2003*b*; Shoup *et al.*, 1992; Tipton and Shoup, 1992).

In this work, a comparative study is done to establish a relationship between the applied load and the stress distribution. Two models are developed: a linear-elastic parametric model, based on energy methods and a bidimensional elastoplastic finite element model. Results of fatigue life predictions obtained with both methodologies suggest that for situations where high values of mean load are present the *TN* methodologies could estimate unrealistic large fatigue lives.

2. ANALYTIC MODEL

Stress levels promoted by the axial loading (P) in studless chain links can be estimated using a simple analytic elastic model which considers the mechanical component as a straight beam and a curved beam with a radius of curvature R, both with a circular cross section of diameter d. This model does not considers complex effects as the contact phenomena that occur between two chain links. Two planes of symmetry are considered in the analysis. Figure 1*a* shows the geometry of the model that represents one-fourth of a studless chain link.

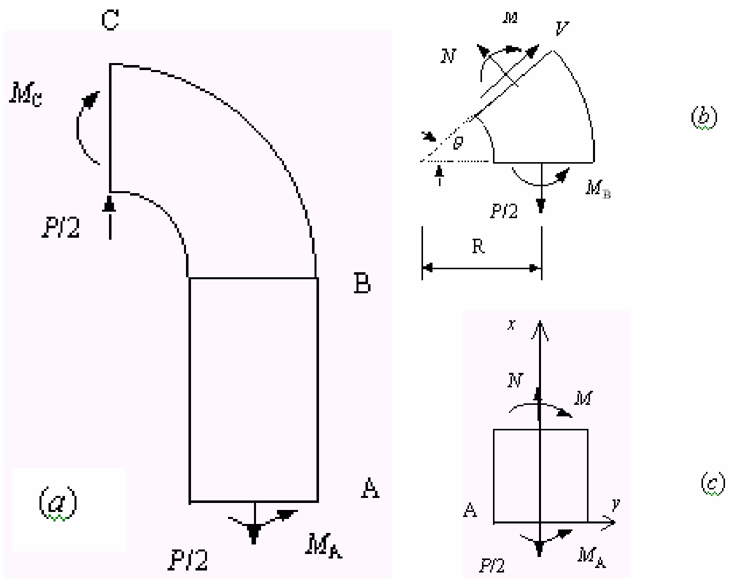


Figure 1. Analytic Model. External loads (*a*). Internal loads in curved part *BC* (*b*) and straight part *AB* (*c*).

The reactions at the two symmetry planes (M_A, M_C and $P/2$) are also shown (note that in Figure 1*a*, by symmetry considerations, only $P/2$ is used). Figures 1*b* and 1*c* show the internal loads (N, V and M) in the straight part *AB* and curved part *BC*, respectively.

From equilibrium, the internal loads can be expressed as $N = P/2$; $V = 0$; $M = M_A$ for the straight beam *AB* and $N = (P/2)\cos\theta$; $V = (P/2)\sin\theta$; $M = M_A - (PR/2)[1-\cos\theta]$ for the curved beam *BC*. The longitudinal stress distribution can be expressed for the straight and curved parts of the chain link, respectively, as:

$$\sigma(y) = \frac{(P/2)}{A} - \frac{M_A \ y}{I} \tag{1}$$

$$\sigma(r,\theta) = \frac{(P/2)\cos\theta}{A} + \frac{\left(M_A - \frac{PR}{2}(1-\cos\theta)\right)(A - A_m r)}{Ar(RA_m - A)} \tag{2}$$

where r is measured from the center of curvature of the curved beam. Therefore, r varies from $R - d/2$ to $R + d/2$ and θ varies from 0 to $\pi/2$. A and I are the area and the moment of inertia of the circular cross section, respectively, and for circular sections $A_m = 2\pi\left(R - \sqrt{R^2 - (d/2)^2}\right)$.

This mechanical component is statically indeterminate and additional equations, necessary to solve the problem, can be obtained through energy methods (Paiva, 2000; Medeiros, 2003; Boresi and Sidebottom, 1985). For the two parts, the elastic strain energy can be expressed as:

$$U_{AB} = \int_0^{L_{AB}} \frac{N^2}{2AE} dx + \int_0^{L_{AB}} \frac{M^2}{2EI} dx \tag{3}$$

$$U_{BC} = \int_0^{\frac{\pi}{2}} \frac{N^2 R}{2AE} d\theta + \int_0^{\frac{\pi}{2}} \frac{A_m M^2}{2A(RA_m - A)E} d\theta - \int_0^{\frac{\pi}{2}} \frac{M \ N}{AE} d\theta + \int_0^{\frac{\pi}{2}} \frac{kV^2 R}{2AG} d\theta \tag{4}$$

where E is the elastic modulus and G the shear modulus; L_{AB} is the distance between sections A and B, and k = 4/3 (Boresi and Sidebottom, 1985). The total elastic strain energy is $U = U_{AB} + U_{BC}$ and the symmetry condition requires that the section A rotation must be zero ($\phi_A = 0$). Applying Castigliano's theorem in section A furnish an additional equation that permits to obtain M_A and, therefore, the stress distribution in the chain link:

$$\phi_A = \frac{\partial U}{\partial M_A} = 0 \tag{5}$$

3. NUMERICAL MODEL

A simplified bidimensional elastoplastic model with large deformations is developed (Pacheco, 2003*a*). This bidimensional model uses plain stress hypothesis and has an equivalent cross section area with a width d and a thickness $\pi d/4$. Two planes of symmetry are considered for this situation in order to reduce the computational costs. Elements PLANE82 (plain strain 8 nodes - 2 degree of freedom per node) are used (Ansys, 2001) for the spatial discretization and the final meshes are defined after a convergence analysis. Figure (2) shows the mesh, load and boundary conditions. In order to simulate the loading condition, a pressure distribution is applied to the contact area between the chain link and the other mooring elements. This pressure distribution is equivalent to the resultant load in the contact region, and varies linearly from a maximum value to zero at the border of the contact area. An angle of 35° with the axial direction (y axis) is selected to represent the contact area. Since contact phenomenon is not incorporated into the analysis, it is expected that results near this region are not representative. Nevertheless, results of this model may be useful for the analysis of points far from this region. The geometry of the studless chain link is in accordance with ISO 1704 recommendation (ISO, 1991). For a circular section with diameter d, the length is equal to $6d$ and the maximum width is $3.35d$. In this study the maximum width value is adopted.

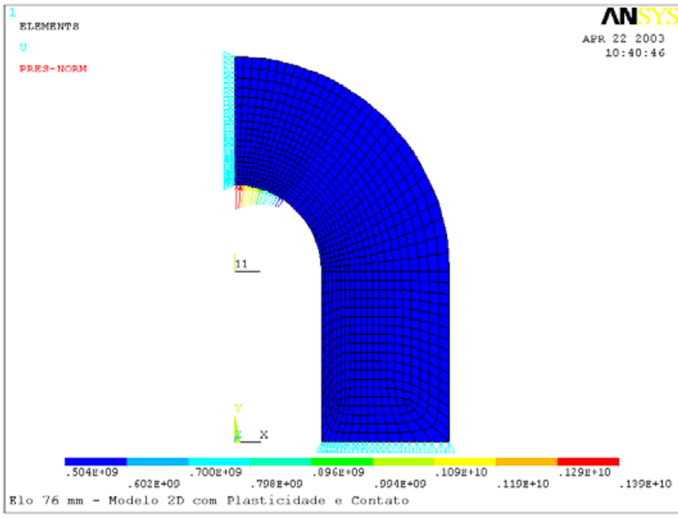


Figure 2. Bidimensional elastoplastic finite element model.

4. NUMERICAL SIMULATIONS

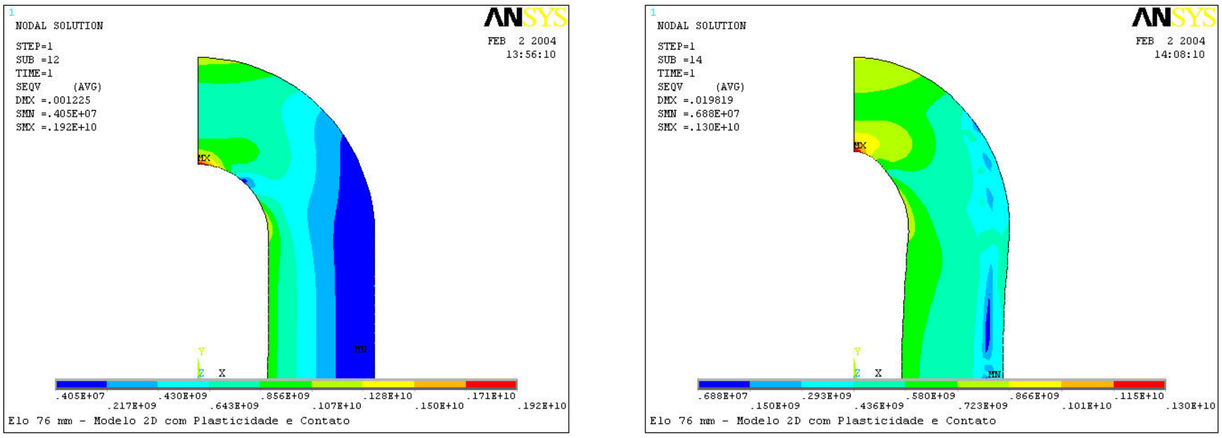
For the material, an elastic modulus (E) of 207 GPa and a Poisson ratio (ν) of 0.29 are used. For the elastoplastic analyses, the minimum mechanical properties dictated by IACS-W22 standard (IACS, 1993) are also adopted: yield stress of 410 MPa (S_Y), tensile strength of 690 MPa (S_U) and a tensile strength strain (ε_u) of 8.5%. These stresses and strains values are transformed from engineering values to real values and are used as parameters for a bilinear kinematic hardening model (Ansys, 2001) adopted to represent the elastoplastic behavior of the material. Also, a proof load of 3242 kN is adopted, according to IACS-W22 (IACS, 1993) for a 76 mm diameter chain link.

Numerical simulations with the elastoplastic model are developed to estimate the residual stress distribution present in the studless chain links before they enter in operation. The condition prior to operation is achieved by first applying and then removing the recommended proof load, considering an initial material condition prior to the proof load. In this work the condition obtained after the application of the proof load is called as *tested*. For the elastic models there are no residual stresses prior to operation.

Numerical results show that high values of stress are present after the application of the proof load. The elastoplastic model shows that extensive plastic deformation develops in the whole component due to initial proof load. As a consequence of this load, high values of residual stresses (of initial yield strength magnitude) are present after load removal.

Figure (3) shows *von Mises* equivalent stress distribution present during the proof load considering elastic (Fig 3*a*) and elastoplastic (Fig 3*b*) materials. Figure (4) shows *von Mises* equivalent residual stress distribution present after the proof test load. Figure (5) shows *von Mises* equivalent stress distribution present during a subsequent operational loading equal to one fourth of the proof load (810 kN) considering elastic (Fig 5*a*) and elastoplastic (Fig 5*b*) materials. A direct comparison between elastic and elastoplastic models shows that elastic model predicts, as expected, considerable higher levels of stress.

It is important to mention that high residual compressive stresses are observed in regions where higher stresses develop during the initial proof load. This effect can be highly beneficial for the fatigue strength of the component as the operational load is lower than the proof load and, therefore, these regions work with a lower mean stress component or even with only compressive stresses.



(*a*) (*b*)

Figure 3. *von Mises* equivalent stress distribution for the proof load – (*a*) Elastic and (*b*) elastoplastic models.

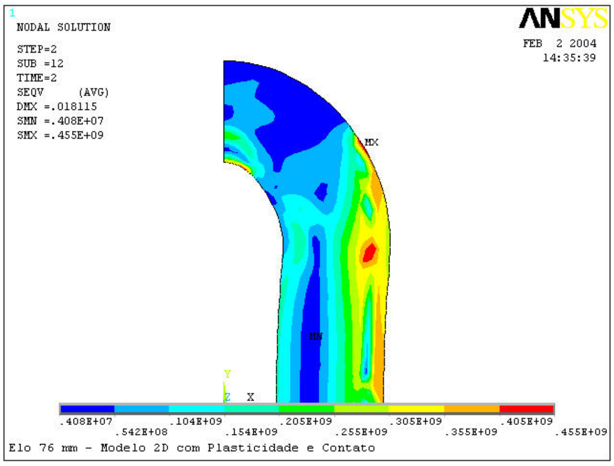
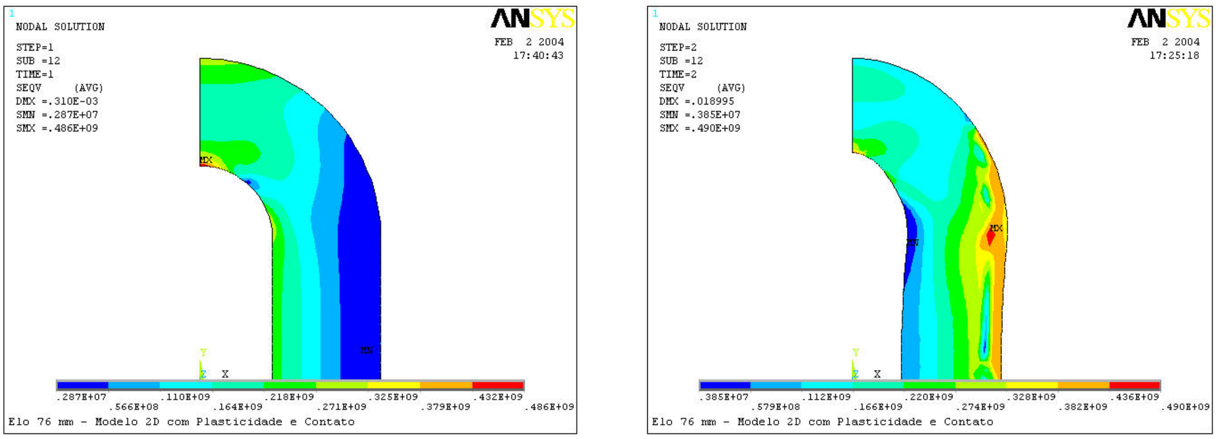


Figure 4. *von Mises* residual stress distribution. Elastoplastic model.



(*a*) (*b*)

Figure 5. *von Mises* equivalent stress for the operational loading - (*a*) Elastic and (*b*) elastoplastic models.

5. FATIGUE LIFE

The API2SK Recommended Practice (API, 1995) furnishes a methodology based on a *T-N* curve (tension *x* number of cycles) that is normally used to predict the fatigue life of mooring line components. This methodology is exclusively based on the load applied on the component and does not consider the stress-state of the component promoted by the applied load. The API2SK furnishes the following *T-N* equation:

$$N\,R^{M} = K \tag{6}$$

where N is the number of cycles, M and K are material parameters. For common chain links, the recommended practice indicates values of 3.36 and 370 for M and K, respectively (API, 1995). R is the ratio of tension range to nominal breaking strength as showed at the following:

$$R = \frac{T_{\max} - T_{\min}}{T_{BL}} \tag{7}$$

where T_{max} and T_{min} are the maximum and minimum values, respectively, of a sinusoidal loading pattern; T_{BL} is the nominal breaking strength. According to IACS-W22 (IACS, 1993) a nominal breaking strength of 4884 kN is adopted.

In order to perform a direct comparison between the mooring line tension-cycle (*T-N*) and traditional mechanical components stress-cycle (*S-N*) approaches, the fatigue *S-N* method (using the modified Goodman diagram to consider the mean stress component) is employed to estimate the fatigue life of chain links (Shigley and Mischke, 2001). For the mechanical component analyzed, an endurance limit (S_e) of 160 MPa is obtained using the following equation (Shigley and Mischke, 2001):

$$S_e = k_a\, k_b\, k_c\, k_d\, k_e\, (0.504\, S_u) \tag{8}$$

and considering the following modification factors: surface factor k_a = 0.52 (hot-rolled), size factor k_b = 0.86 (round bar in bending not rotating), load factor k_c = 1 (bending), temperature factor k_d = 1 (room temperature) and miscellaneous-effects factor k_e = 1 (Shigley and Mischke, 2001). The predicted fatigue life (N, in cycles) for all models are obtained employing the following expressions:

$$N = \left[\frac{\sigma_a^{von\,Mises}}{a} \left(1 - \frac{\sigma_m^{von\,Mises}}{S_u} \right)^{-1} \right]^{1/b}$$

$$a = \frac{(0.9\, S_u)^2}{S_e}\,; \qquad b = -(1/3)\log\left(\frac{0.9\, S_u}{S_e} \right) \tag{9}$$

The stress levels obtained from the two models developed (analytic linear-elastic and bidimensional elastoplastic finite element models) are considered for the *S-N* approach adopted in this analysis. The developed stresses were accomplished in three sections (*A*, *B* and *C*) shown in Figure (6). The region near the load application is not considered in the analysis. The analysis is performed considering the variation of loading parameters T_{min}/T_{BL} and T_{max}/T_{BL} in the following ranges: $0 \le T_{min}/T_{BL} \le 0.1$ and $0.2 \le T_{máx}/T_{BL} \le 0.4$.

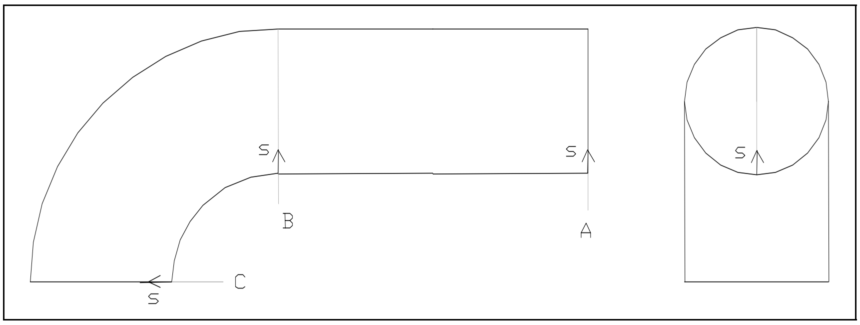
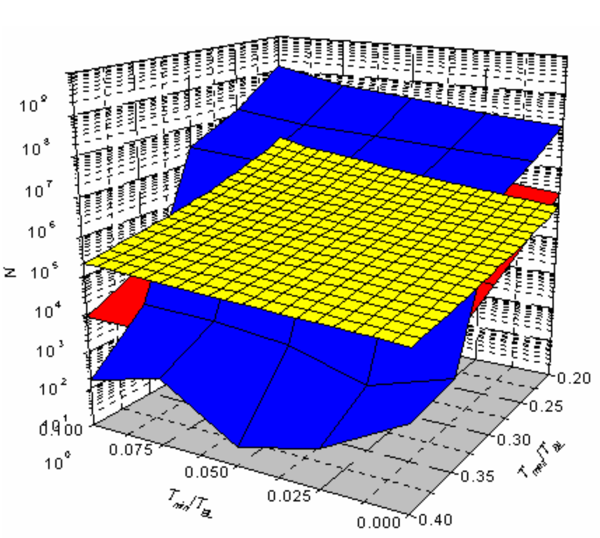
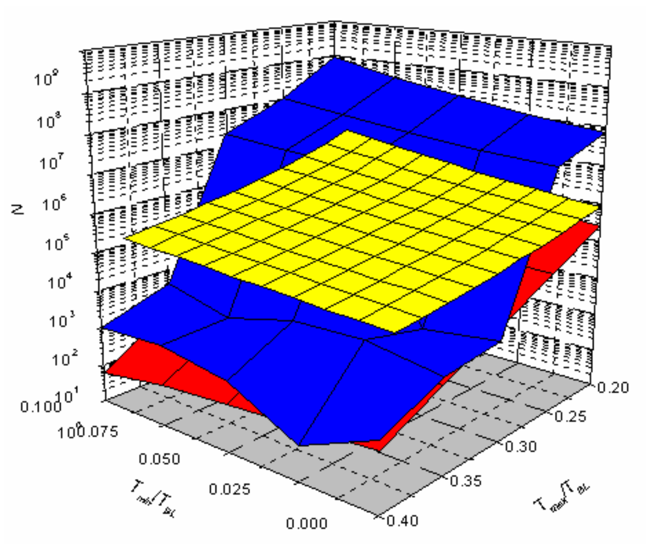


Figure 6. Sections *A*, *B* and *C*.

Figure (7) shows a comparative tridimensional plot of the predicted fatigue life (*N*) for sections *A*, *B* and *C*, considering the two models developed (analytic linear-elastic and bidimensional elastoplastic fionite element models) and the *TN* methodology. The predicted fatigue life of the two models is calculated considering the critical point of the three sections analyzed.



(*a*)



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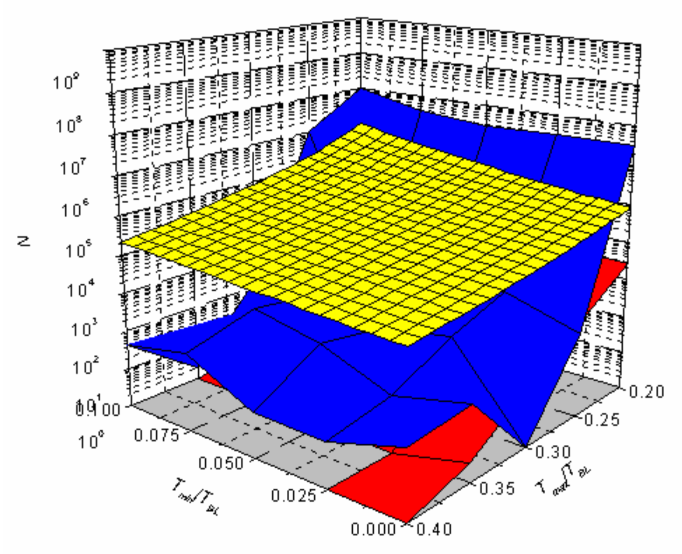

(*c*)

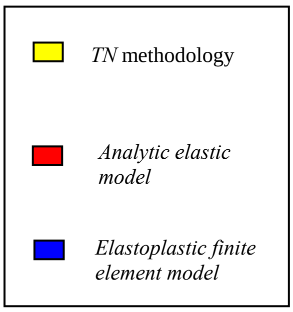

Figure 7. Number of cycles for the critical point in: (*a*) Section *A*, (*b*) Section *B* and (*c*) Section *C*.

Figure (7) shows that analytic linear-elastic model is the more conservative model (almost at every combination tested). The analytic linear-elastic model and elastoplastic finite element models are considerable affected by the mean loading (having shorter lives for larger mean loads).

It's worth to mention that in previous works (Pacheco *et al.*, 2002, 2003*a*, 2003*b*), the authors have shown that the material condition significantly influences the residual stress developed and have a preponderant effect on the structural integrity of chain links. Therefore, a more detailed study involving different material conditions must be addressed in future works.

6. CONCLUSIONS

This work presents a comparative study on fatigue lives prediction in studless chain links. The responses of a bidimensional_elastoplastic finite element model, a simple elastic analytic model and the *TN* methodology are compared considering a 76 mm studless chain link. The study addresses two important phenomena that can influence the stress distribution prediction: plasticity and mean stresses.

For typical operational loading conditions, numerical simulations show that the elastoplastic analysis furnishes a completely different stress distribution than the one obtained with elastic analysis, due to the high level of residual stresses present in a chain link. These results suggest that the plasticity must be considered in the analysis of chain links to support a reliable integrity analysis of this type of mechanical component. A simplified *S-N* method is used to estimate the fatigue life of the chain links in a comparative study using results obtained in the numerical simulations for two models.

It's worth to mention that an experimental program to measure residual and operational stresses must be established to confirm the present results.

7. ACKNOWLEDGEMENTS

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