Evaluating Fatigue as a Failure Mechanism in Zigzag Pipelines

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ABSTRACT

Straight pipelines supported by unstable soil can fail by localized buckling induced by thermal stress when transporting heated products. A zigzag instead of a straight geometry can avoid such problems by forcing the pipeline to bend in a predictable manner. However, the bending stresses in zigzag pipelines can be quite high, inducing significant elastic-plastic strain loops near the root of corrosion pits or similar unavoidable defects which can, consequently, lead to fatigue cracking due to the heating-cooling cycles. Since this problem is not addressed in welded pipeline design codes, a methodology to calculate the initiation and propagation lives is proposed and applied to evaluate the potential effect of this failure mechanism.

INTRODUCTION

A zigzag instead of a straight geometry is an uncommon but interesting solution to avoid localized buckling in pipelines designed to work with heated products. The pumping of these products warms the pipeline, introducing compressive thermal stresses whenever there is a restriction on its free expansion, for example when it is rigidly connected to the soil. Such a good support can generally prevent any pipeline buckling. However, if a long enough stretch of the pipeline is not so well supported (e.g. when it is buried in unstable soil or leaned on any other unsuitable foundation), those compressive thermal stresses can induce that part of the pipeline to fail by localized buckling. Such failures can lead to excessive localized plastic strains and eventually to ductility exhaustion, which can cause the pipeline to crack and possibly to spill its content. The zigzag design can avoid such severe buckling problems by forcing the heated pipeline to bend in a predictable and controlled manner.

On the other hand, the maximum bending stresses in the purposedly curved shape of a properly designed zigzag pipeline can be quite high, close to the yield strength of the material. Since the number of heating-cooling cycles is normally in the order of a few per month, such localized high stresses do not appear in principle to be a big problem for the very tough steels normally used in pipelines. However, those cycles can induce significant hysteresis loops of elastic-plastic strains near the roots of surface defects (such as corrosion pits, e.g.) on the pipeline, if there are any in the highly stressed regions. In fact, the heating-cooling cycles can lead to fatigue cracking in a number of cycles much smaller than the anticipated for a flawless pipeline, because of the stress concentration effect of the defects.

Since there is no way to warrant that all the critical regions of the pipeline are flawless, its designer or analyst should assume that a defect might be there for structural integrity evaluation purposes. However, the initiation and propagation of fatigue cracks from pits is usually not considered a major risk in the design of oil pipelines, even when their dominant failure mechanism is assumed to be corrosion. Therefore, it appears to be wise to consider the possible effect of the combination of both mechanisms when dealing with high-amplitude low-cycle loads under long periods of time.

This type of consideration can also be applied in structural integrity evaluations of old pipelines. For example, several Brazilian pipelines have been in service since the 1960's, carrying heated oil products under variable temperatures and pressures. All of them were designed as nominally straight at that time, using segments with a very large radius of curvature when needed. However, some stretches of these veteran pipelines have been locally bent or curved due to soil settling or accidents. These slight perturbations in a nominally straight geometry do not cause any important decrease in the pipeline static safety factor, in the sense that their burst pressure is virtually unaffected by them, due the very ductile and tough steel used in their manufacturing.

However, as in the zigzag case, these curves or bends induce significant localized cyclic bending stresses when the pipelines are heated and cooled. Since no bending stresses were assumed in their design, and since there is a non-negligible chance to find corrosion defects near to or at their most stressed regions, it seems reasonable to consider fatigue cracking at these possible defects when evaluating the residual pipeline service lives. In the following, a methodology is proposed and applied to predict the fatigue crack initiation and propagation lives of zigzag or bent pipelines.

FATIGUE ANALYSIS FUNDAMENTALS

Fatigue is a type of mechanical failure primarily caused by the repeated application of variable loads. Fatigue failures are

characterized by the stable generation and/or propagation of a crack, which gradually grows until eventually causing the structure to fracture. These phenomena are progressive, cumulative and localized.
The crack generation usually starts from a notch root, and depends primarily on the range of the (local) stress $\Delta\sigma$ or strain $\Delta\varepsilon$ acting on that critical or most loaded point of the structure. For design purposes, or whenever the classical (macroscopic) stress analysis techniques are used, $\Delta\sigma$ and $\Delta\varepsilon$ should be calculated in a volume large in comparison to the microstructural parameter which characterizes the material anisotropy (e.g., the grain size in metals). When the cyclic loads are small (i.e., macroscopically elastic), the material fatigue strength S_F can usually be assumed to depend on the rupture strength S_U. But when these loads are large (causing macroscopic cyclic yielding), the main material fatigue controlling parameter is its ductility.

Under macroscopic stresses greater than the material cyclic yielding strength S_{Yc}, the so-called **εN** method is the traditional procedure for designing against fatigue crack initiation [1-4]. This is a *local* design method, in the sense that it only requires the strain history at the *critical point* of the structure to predict the fatigue crack initiation life using the material fatigue properties (i.e., the stress and strain fields at the remaining of the structure are not *required* for the **εN** modeling process).

The **εN** method correlates the number of cycles **N** to *initiate* a fatigue crack in any structure with the life (in cycles) of small specimens that should be tested having: (i) the same fatigue strength (hence, the same material and superficial details) and (ii) under the same strain history that loads the structure critical point (generally a notch root) in service. This method can only be applied to predict crack initiation (since it does not explicitly deal with the presence of cracks), but it recognizes macroscopic elastic-plastic hysteresis loops at the notch roots and uses the local strain range $\Delta\varepsilon$ (a parameter more robust than $\Delta\sigma$) there to predict the fatigue crack initiation life **N**.

A well calculated nominal stress/strain history or a properly positioned strain gage plus an appropriate stress concentration factor can provide all the loading information required to apply the **εN** design method. In this way, it is normally necessary to quantify the elastic-plastic stress and strain ranges $\Delta\sigma$ and $\Delta\varepsilon$ induced by the nominal stress $\Delta\sigma_n$ and strain $\Delta\varepsilon_n$ ranges at the critical notch root whose geometrical (or linear-elastic) stress concentration factor is K_t. In practice, $\Delta\sigma$ and $\Delta\varepsilon$ can be reasonably well estimated using Neuber's rule (mainly when the notch root is under a plane-stress controlled state [3]), which states that the product of the stress concentration $K_\sigma = \Delta\sigma/\Delta\sigma_n$ and the strain concentration $K_\varepsilon = \Delta\varepsilon/\Delta\varepsilon_n$ factors is constant and equal to the square of the geometric stress concentration factor K_t

$$K_t^2 = \frac{\Delta\sigma \cdot \Delta\varepsilon}{\Delta\sigma_n \cdot \Delta\varepsilon_n} \tag{1}$$

Note that K_t can only be used to amplify elastic stresses, referred to the point of the structure where the nominal stress is calculated or measured. Therefore, when the nominal stress amplitudes are lower than S_{Yc} (i.e. when $\Delta\sigma_n < 2 \cdot S_{Yc}$), it still is a regrettably common practice to model them as Hookean and to use Neuber's equation in the simplified form

$$K_t^2 = \frac{\Delta\sigma \cdot \Delta\varepsilon \cdot E}{\Delta\sigma_n^2} \tag{2}$$

Ramberg-Osgood is one of many empirical relations that can be used to model the cyclic elastic-plastic stress-strain behavior of structural materials. Its main limitation is not being able to recognize a purely elastic behavior, and its main advantage is its mathematical simplicity. It can be used to describe the stresses and strains at the notch root by

$$\Delta\varepsilon = \frac{\Delta\sigma}{E} + 2\left(\frac{\Delta\sigma}{2H_c}\right)^{1/h_c} \tag{3}$$

where E is the Young's modulus, H_c is the hardening coefficient and h_c is the hardening exponent of the cyclically stabilized $\Delta\sigma\Delta\varepsilon$ curve. Eliminating $\Delta\varepsilon$ from Equations (2) and (3), $\Delta\sigma_n$ can be directly related to $\Delta\sigma$ by

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

However, the above equation is logically incongruent, since it treats the same material by two different models: Ramberg-Osgood at the notch root and Hooke at the nominal region. Moreover, this procedure can generate significant numerical errors even when the nominal stresses are much lower than the material cyclic yielding strength. To avoid the errors induced by the classical (or Hookean) Neuber approach, it is necessary to use the Ramberg-Osgood model to describe not only the stresses at the notch root, but also to describe the nominal stresses, writing

$$\frac{\Delta\varepsilon_n}{2} = \frac{\Delta\sigma_n}{2E} + \left(\frac{\Delta\sigma_n}{2H_c}\right)^{1/h_c} \tag{5}$$

In this case, given $\Delta\sigma_n$, the stress range at the notch root $\Delta\sigma$ can be calculated from Equations (1), (3), and (5), giving

$$K_t^2\left(\Delta\sigma_n^2+\frac{2E\Delta\sigma_n^{(h_c+1)/h_c}}{(2H_c)^{1/h_c}}\right)=\Delta\sigma^2+\frac{2E\Delta\sigma^{(h_c+1)/h_c}}{(2H_c)^{1/h_c}} \tag{6}$$

To quantitatively account for the errors induced by the Hookean modeling of the nominal stresses, a comprehensive study has been performed on measured properties of 517 different structural steels [5,6]. It has been found that the Hookean simplification in Neuber's rule can lead to significant prediction errors in $\Delta\sigma$ and $\Delta\varepsilon$, even when the nominal stress ranges $\Delta\sigma_n$ are much smaller than $\mathbf{2{\cdot}S_{Yc}}$. And due to the highly non-linear nature of Coffin-Manson's **εN** curve, $\Delta\varepsilon/2 = (\sigma_f'/E)(2N)^b + \varepsilon_f'(2N)^c$, which correlates the strain range $\Delta\varepsilon$ with the fatigue crack initiation life **N**, relatively small errors in calculating $\Delta\sigma$ and $\Delta\varepsilon$ can induce much higher *non*-conservative errors in life prediction. Depending on the considered material, even nominal stress amplitudes as low as $\mathbf{0.1{\cdot}S_{Yc}}$ can result in significant non-conservative life prediction errors, see Figures 1 and 2.

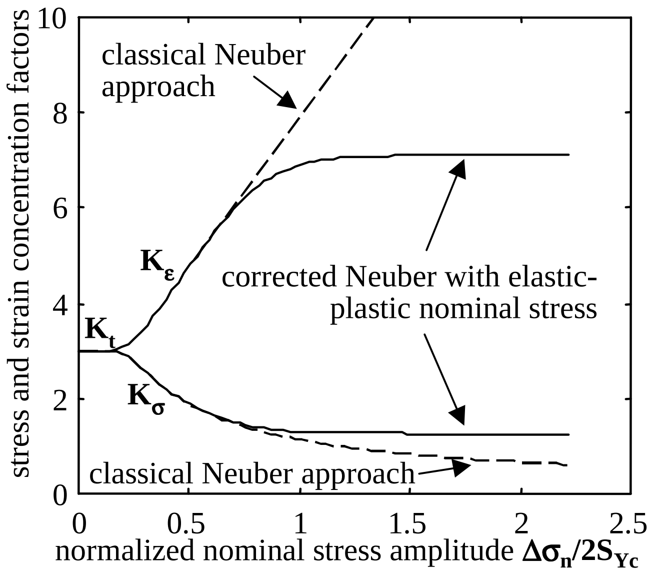

Fig. 1: Stress K_σ and strain K_ε concentration factors calculated by both Neuber approaches (SAE 1009, K_t = 3) [6].

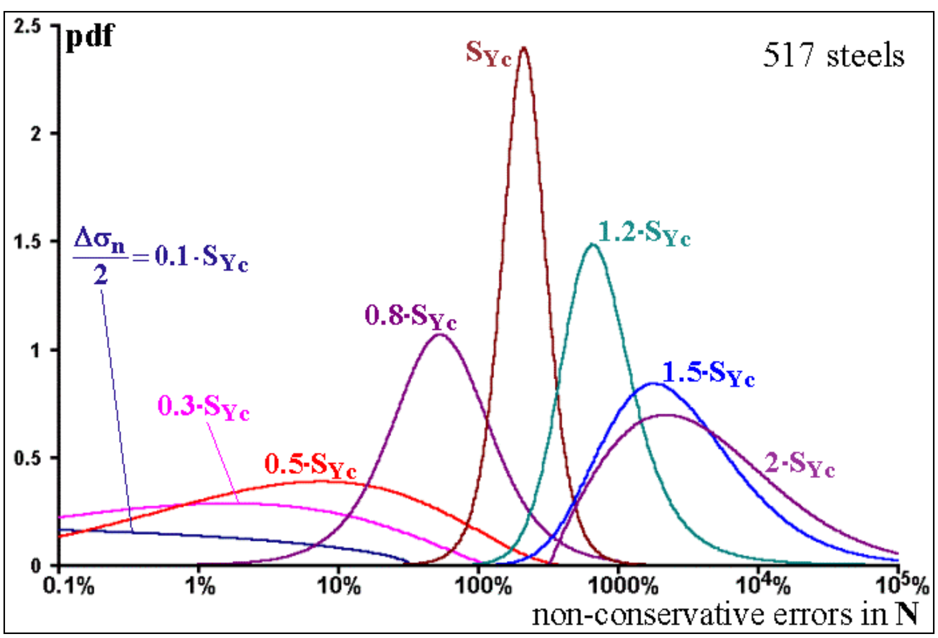

Fig. 2: Statistics of the non-conservative errors in crack initiation lives predicted by the classical (Hookean) Neuber approach, for several nominal stress amplitudes $\Delta\sigma_n/2$ (**517 steels**, K_t = 3) [6].

In summary, it should be mandatory to use Ramberg-Osgood to model both the nominal and the critical stresses and strains, as shown in Equation (6), otherwise completely wrong crack initiation life predictions may be obtained. This point is particularly important for the case discussed here, since the maximum nominal stresses in zigzag or bent pipelines is in the order of $\mathbf{S_{Yc}}$.

The remaining fatigue life of cracked structures can only be calculated using Fracture Mechanics concepts. The fatigue crack propagation rate **da/dN** is primarily controlled by the range of the stress intensity factor $\mathbf{\Delta K = \Delta\sigma \cdot [\sqrt{(\pi a)}] \cdot [f(a/w)]}$ (and *not* by the stress $\Delta\sigma$ or by the strain $\Delta\varepsilon$ ranges), which depends not only on $\Delta\sigma$ but also on the crack length **a** and on the geometry of the cracked structure, through the non-dimensional function **f(a/w)** that quantifies the effect of all geometric parameters that influence the stress field ahead of the crack front.

Several **f(a/w)** expressions can be found in the literature [7], in special for through cracks (which propagate in 1D). However, fatigue cracks commonly nucleate as part-through cracks, which propagate in 2D. The main characteristic of these cracks is a non-homologous fatigue propagation, because in general the crack front tends to change form from cycle to cycle, since $\mathbf{\Delta K}$ varies from point to point along the crack front. A few analytical expressions are available for the stress intensity factor of 2D cracks, such as surface, corner or internal cracks under combined tension and bending [8]. If the cracks have ellipsoidal fronts, and if they are built in a plate of width **w** or **2w** and thickness **t**, the stress intensity range $\mathbf{\Delta K = \Delta\sigma \cdot \sqrt{(\pi a)} \cdot f_\theta(a/c,\ a/t,\ c/w)}$ is a function of the stress range $\Delta\sigma$ and of the ratios **a/c**, **a/t** and **c/w** (where **a** and **c** are the ellipsis semi-axes) through a crack shape function $\mathbf{f_\theta}$ of the angle $\boldsymbol{\theta}$ defined in Fig. 3.

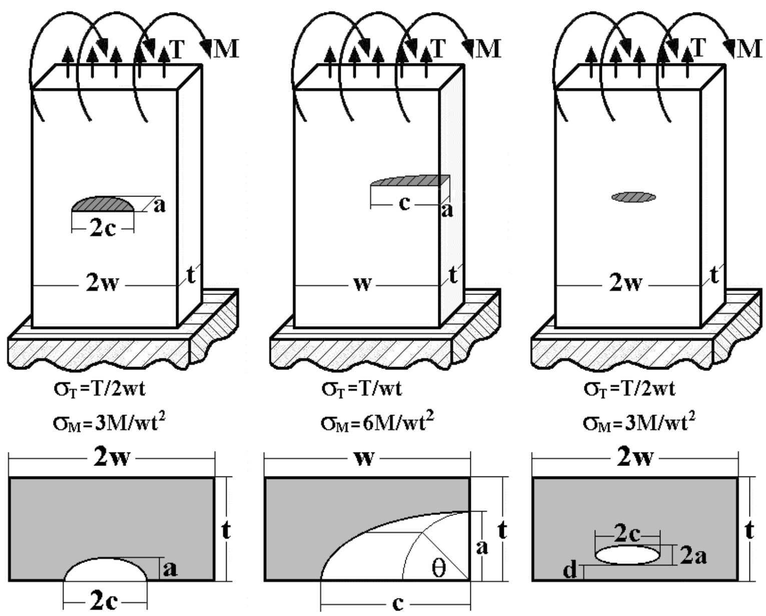

Fig. 3: Idealized surface semi-elliptical, corner quart-elliptical, and internal elliptical cracks of semi-axes **a** and **c** in a plate of thickness **t** and width **w** or **2w**, subjected to bending moments **M** and normal tensile loads **T** [4].

Many actual surface, corner, or internal cracks can be quite well approximated by these idealized 2D ellipsoidal cracks for fatigue propagation prediction purposes. Fractographic observations indicate that the successive fronts of those cracks tend to quickly achieve an elliptical form, and to stay approximately elliptic during their fatigue propagation, even when the initial crack shape is far from an ellipsis, see Fig. 4. Therefore, it can be assumed in the modeling that fatigue propagation just changes the shape of the 2D cracks (given by the ratio **a/c** between the ellipsis semi-axes, which quantifies how elongated the cracks are), but preserves their basic ellipsoidal geometry. The idea is then to maintain the fundamental hypothesis of the ellipsoidal geometry preservation, but accounting for the coupled growth in the depth (**a**) and surface width (**2c** for surface and **c** for corner cracks) directions of the 2D cracks.

In practice, fatigue problems generally occur under complex variable amplitude load histories. Generating reliable life predictions under these conditions is, to say the least, a laborious task when the load sequence is important. For example, plastic strains are history-dependent, and the fatigue life prediction model must preserve the load sequence information to describe in an appropriate manner the effects of stress/strain hysteresis loops at a notch root. Overload-induced crack retardation or arrest is another sequence-dependent problem. To solve problems such as these, a powerful academic program named **ViDa** (which means "life" in Portuguese, but also stands for <u>Vi</u>sual <u>Da</u>magemeter) has been developed to automate the fatigue design routines by all local methods, giving the user total control over the calculation procedures [9]. This software can calculate fatigue crack initiation by the **SN**, the **IIW** (for welded structures) and the **εN** methods, and crack propagation using the Fracture Mechanics-based **da/dN** methodology for any complex loading history. A comprehensive description (in English) of its major features, including its 110 page manual and several demo videos, can be downloaded from www.tecgraf.puc-rio.br/vida.

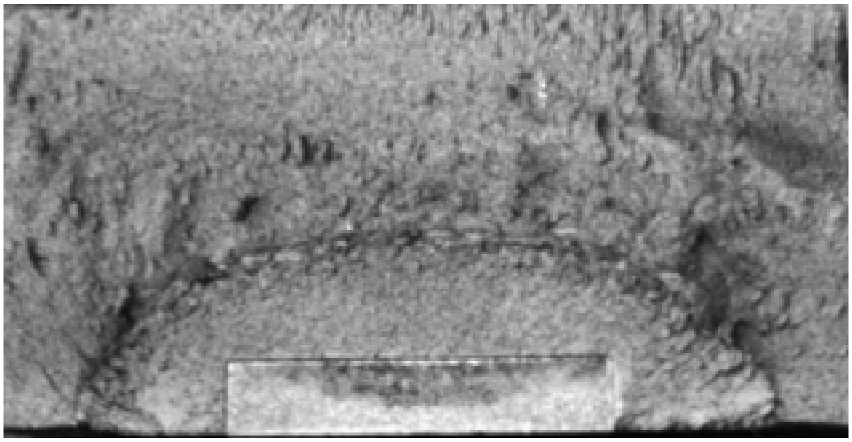

Fig. 4: An approximately semi-elliptical crack which started at a rectangular notch [4].

The developed program includes all necessary tools to perform the fatigue life predictions. It starts with an intuitive and friendly graphical interface in six languages, based on traditional engineering units and notation. The interface is integrated to several intelligent and expansible databases which contain most information required for the calculations, such as properties of over 13000 materials, hundreds of stress concentration and intensity factors, dozens of **da/dN** equations and several crack propagation models (to consider load interaction effects such as crack retardation or arrest after overloads), and more. Among other features stand out traditional and sequential rain-flow counters; graphical output for all material properties (like **σε**, **SN**, **εN** and **da/dN** curves) and computed results (including corrected elastic-plastic hysteresis loops, 2D crack fronts, damage progression, etc.); automatic adjustment of crack initiation and propagation experimental data; an equation interpreter which recognizes the BASIC syntax; and complete documentation.

These features allow, for example, crack growth to be calculated considering any propagation model and any ΔK expression. Moreover, it has safety features to automatically stop the calculations if, during any loading event, it detects that: (i) $K_{max} = K_C$, the material toughness; (ii) the crack has reached its maximum specified size; (iii) the stress in the residual ligament reaches the rupture strength of the material S_U; (iv) **da/dN** reaches **0.1mm/cycle** (for most engineering alloys, above this rate the problem is fracturing, not fatigue cracking); or else if (v) one of the borders of the piece is reached by the crack front, in the part-through crack propagation case (however, for some geometries, the software is able to model the transition from part-through to through cracks). Moreover, it informs the user when there is yielding in the residual ligament before the maximum specified crack size or number of load cycles is reached. In this way, the computed values can be used with the guarantee that the validity limits of the mathematical models are never exceeded.

Finally, another software named **Quebra2D** (meaning 2D fracture in Portuguese) has been developed to supplement **ViDa**'s features. This is an efficient interactive graphical program for simulating two-dimensional fracture processes using adaptive FE analyses and specialized crack tip elements. It has powerful automatic remeshing schemes which work both for regions with no cracks or with one or multiple cracks, which may be either embedded, surface breaking or branched. **Quebra2D** can calculate K_I and K_{II} in generic 2D structures along the (generally curved) crack path under 2D mixed-mode loading using three techniques (displacement correlation, modified crack-closure integral, or J-integral computed by means of the equivalent domain integral), and the crack incremental growth direction by means of the maximum circumferential stress, the maximum potential energy release rate, or the minimum strain energy density criteria. Further details are available elsewhere [10-12]. In the next sections, these tools are used to predict the fatigue life of pipelines.

PREDICTION OF THE RESIDUAL LIFE OF AN OLD PIPELINE

The technique outlined above is applied to predict the residual life of an old 12" pipeline made of API 5L Gr. B steel and isolated by a 50mm concrete layer, in service for more than 40 years. Since its inauguration, this pipeline has carried several heated products under variable temperatures and pressures. The nominal stresses associated with this loading are quite low, and present no fatigue risk. However, due to soil settling or to operational accidents, in a few places the originally straight pipeline layout has been slightly bent. These defects have been recently discovered after an instrumented pig inspection, but there is no record of when they occurred. Due to this small change in the pipeline geometry, its service stresses have been much increased locally (peaks close to the yield strength of the steel have been FE calculated for the nominal stress). This huge increase is due to the thermal loads, which can induce significant bending in these now curved parts of the pipeline. To evaluate its actual fatigue risk, the elastic-plastic fatigue damage is calculated at the root of a functionally admissible notch or corrosion pit using the **εN** method, and the effects of surface semi-elliptical cracks in its internal (or external) wall is studied considering appropriate stress intensity factor expressions and the actual service loads.
The mechanical properties of the pipeline steel were measured (following appropriate ASTM standards) from test specimens machined from a section that was removed from service, resulting in the values shown in Table 1.

Table 1: Measured mechanical properties of the API-5L-Gr. B steel.

yield strength	S_Y **= 294 MPa**
ultimate strength	S_U **= 423 MPa**
reduction in area	**RA = 60%**
Young's modulus	**E = 208 GPa**
cyclic hardening coefficient	H_c **= 1229 MPa**
cyclic hardening exponent	h_c **= 0.24**
Coffin-Manson's elastic coefficient	σ_f' **= 964 MPa**
Coffin-Manson's elastic exponent	**b = 0.145**
Coffin-Manson's plastic coefficient	ε_f' **= 0.36**
Coffin-Manson's plastic exponent	**c = 0.55**
crack propagation rate (m/cycle, **R=0.1**)	$da/dN = 2.1\cdot10^{10}(\Delta K - 6.7)^{2.5}$ (ΔK in $MPa\sqrt{m}$)
crack propagation threshold (**R=0.1**)	$\Delta K_{th} = 6.7\ MPa\sqrt{m}$.

During its service, this pipeline has carried several products under operating temperatures between **60** and **80°C**, working under internal pressures between **6** and **15 kgf/cm²**. A Finite Element (FE) analysis of the duct was performed considering both temperature and pressure effects, but neglecting any surface defect. It was found that the thermal bending stresses were about 8 times higher than the stresses induced by the internal pressure. The calculated Mises nominal stresses (in **MPa**) due to the thermally induced bending and to the internal pressure stress history could be resumed by the sequence

$$\{2\times[0\to217\to177\to217\to0] + [0\to257\to0] + 4\times[0\to217\to177\to217\to0]\} \times 12 \text{ months} \times 31 \text{ years (from 1961 until 1991)} + \\ + \{15\times[0\to177\to0]\}\times 12 \text{ months} \times 10{,}33 \text{ years (since 1992)} \qquad (7)$$

Even the largest peaks in the load history above must be considered as *nominal* service stresses, because they do not account for the effect of small surface flaws that could be present on the pipeline wall. In fact, these defects are highly probable in a hot-rolled material operating for such a long time, and their effect must be considered in the residual fatigue life analysis. Thus, a pit-like stress concentration factor K_t has been postulated in the pipeline critical region. Since a semi-spherical corrosion pit in a semi-infinite solid has $K_t = 2.5$ [13], it has been assumed in the calculations that $2.0 < K_t < 3.0$.

Note that irregularly shaped pits may result in K_t values even higher than **3.0**, but for risk evaluation purposes that range was assumed as representative. On the other hand, it is probably too conservative to assume a corrosion pit from the very beginning of the pipeline service life, when its surface was still intact. Moreover, it is unlikely that the pipeline, when it was new, already had the defects recently discovered by the pig. However, there is no information on when these problems started. Since completely neglecting the bending and the stress concentration effects could lead to unacceptable non-conservative residual life predictions, it is a reasonable option to calculate a probably lower-limit value for this potential problem assuming a worst case scenario. These **εN** calculations have been performed using **ViDa**, and their results are presented in Tables 2 and 3.

Table 2: Fatigue damage predictions using the **εN** methodology according to the Coffin-Manson, Morrow Elastic, Morrow Elastoplastic and Smith-Topper-Watson (STW) equations (0 means no damage and 1.0 the initiation of a small fatigue crack).

Accumulated Damage since 1961				
K_t	Coffin-Manson	Morrow EL	**Morrow EP**	STW
1.0	0.005	0.007	**0.011**	0.037
2.0	0.077	0.092	**0.173**	0.340
2.5	0.175	0.202	**0.408**	0.681
3.0	0.334	0.378	**0.812**	**1.194**

Table 3: Residual life predictions until the initiation of a (small) fatigue crack on the pipeline.

Residual Life Predicted for the Pipeline (in years before the initiation of a fatigue crack)				
K_t	Coffin-Manson	Morrow EL	**Morrow EP**	STW
1.0	> 100	> 100	> 100	> 100
2.0	> 100	> 100	> 100	> 100
2.5	> 100	> 100	**83.4**	22.3
3.0	> 100	76.6	**13.2**	-

Note that there is a significant variation in the predictions according to the several strain-life models, since each equation is more appropriate to a specific combination of material-loading conditions. Since the mean plastic strain component is relatively high, probably the most accurate predictions are the ones obtained from the Morrow Elastoplastic (MEP) equation. These large differences explain why still today it is common to use very high safety factors ($\phi_N \geq$ **10** in the service life) when designing pipelines using fatigue design codes for welded structures [14]. These codes are based on the **SN** philosophy, and do not explicitly account for plasticity effects such as the one encountered in the case studied here.

Note also that there is a high dependence between the notch stress concentration and the predicted damage. If the corrosion pits in the structure are all such that $\mathbf{K_t < 2.5}$, then no fatigue crack will be predicted for at least another **100** years of service. In this case, fatigue may be neglected as a likely failure mechanism for this pipeline. However, in the presence of surface defects with a $\mathbf{K_t = 2.5}$, MEP predicts a residual life of **83.4** years, while STW method predicts **22.3** years. In the case of $\mathbf{K_t = 3.0}$, the residual life predicted by MEP is only **13.2** years, while STW predicts (probably conservatively) that a small fatigue crack is already present on the pipeline (since its predicted damage is greater than **1.0**).

To evaluate the sensitivity of this pipeline to semi-elliptical surface cracks on its internal wall, crack propagation is studied considering appropriate stress intensity factor expressions and the actual service loads. Conservative estimates of the residual propagation life are obtained neglecting any crack retardation effect, such as crack closure. Six configurations of 2D cracks have been considered in this study: one semi-circular (with $\mathbf{c_i = a_i}$) and two semi-elliptical (with $\mathbf{c_i = 2a_i}$ and $\mathbf{c_i = 4 \cdot a_i}$), considering in each case the crack depths $\mathbf{a_i}$ **= 2mm** and $\mathbf{a_i}$ **= 4.25mm** (half the pipeline thickness).

These chosen crack configurations were assumed to be representative because the smaller ones are of the order of the smallest detectable defects associated with good non-destructive inspection techniques, while the larger ones are defects likely to be found during a careful inspection of the pipeline. The **ViDa** program was used in the calculations, since it conveniently includes stress intensity factors for surface cracks at the inner wall of ducts, see Fig. 5. The results are presented in Table 4. From these results, one must note that even the largest assumed crack cannot induce leakage in the duct, even if it had been present since the beginning of its operational life. Such predictions, however, are highly sensitive to the assumed initial crack size.

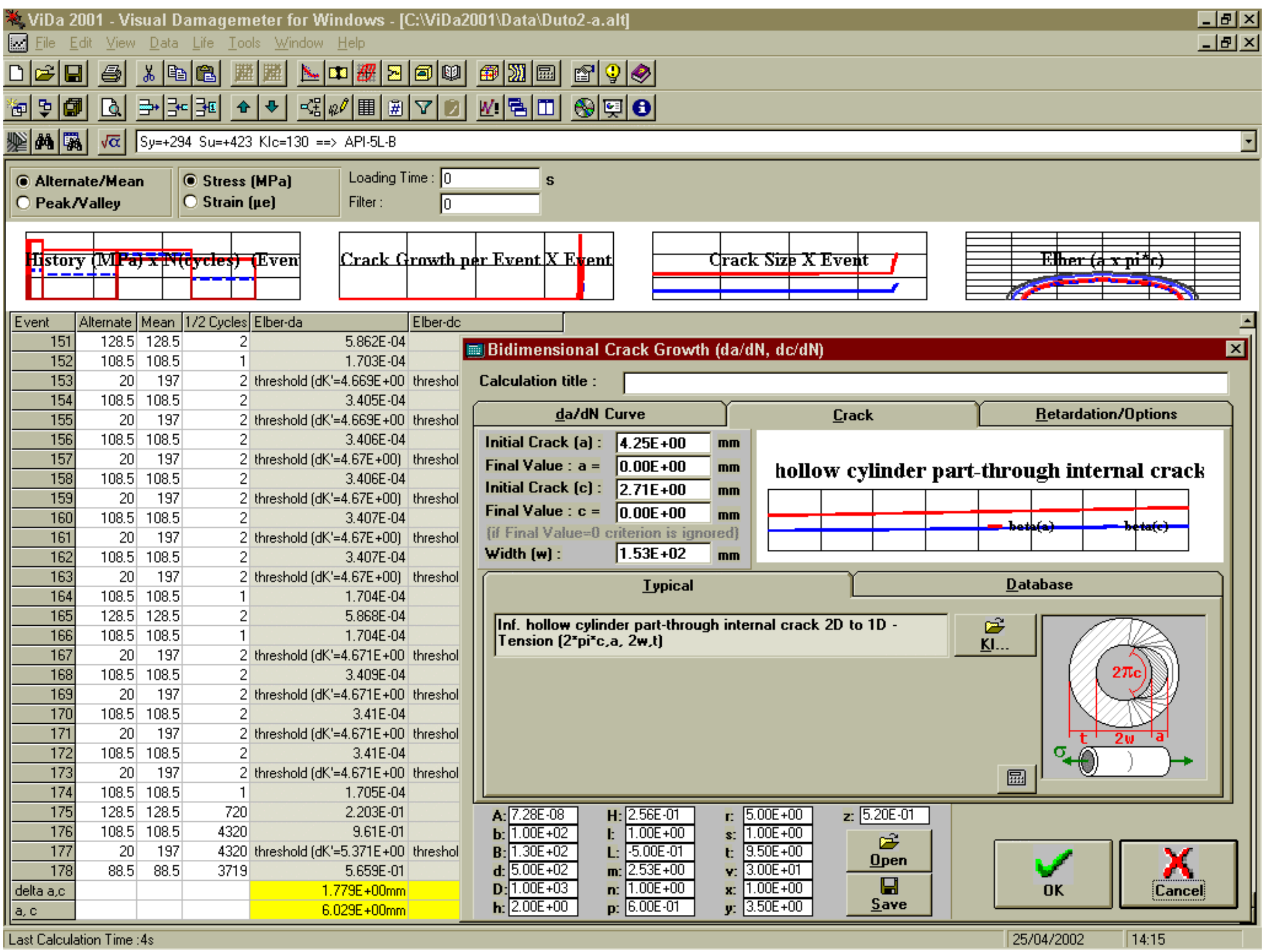

Fig. 5: Residual fatigue life prediction of a surface crack inside the pipeline using **ViDa**.

Table 4: Fatigue propagation of semi-circular and semi-elliptical surface cracks in the duct.

Predicted Growth since 1961 for Several Semi-Elliptical Surface Cracks with Initial Depth a_i and Width $2c_i$					
a_i (mm)	**c_i (mm)**	**a_f (mm)**	**c_f (mm)**	**δa (mm)**	**δc (mm)**
2.00	**1.00**	**2.183**	**2.037**	**0.183**	**0.037**
2.00	**2.00**	**2.172**	**2.463**	**0.172**	**0.463**
2.00	**4.00**	**2.282**	**4.200**	**0.282**	**0.200**
4.25	**2.125**	**5.425**	**6.425**	**1.175**	**2.175**
4.25	**4.25**	**5.429**	**7.053**	**1.179**	**2.803**
4.25	**9.50**	**6.029**	**10.468**	**1.799**	**0.968**

Therefore, in the absence of notches or corrosion pits in the most stressed bent regions of this veteran pipeline, fatigue is unlikely to be an important failure mechanism. However, in the presence of surface flaws associated with stress concentration factors of the order of three, it is probable that a fatigue crack will initiate. On the other hand, if these fatigue cracks initiate at shallow corrosion pits and are small compared to the pipeline wall thickness, its propagation rate is very low. Probably, there is plenty of time to detect these cracks before they can cause oil leaks to the environment or compromise the structural integrity of the pipeline. It is necessary, of course, to use a precise inspection technology to detect small fatigue cracks, but since the points where they must be looked for are known beforehand, the detection problem is not a major one.

EVALUATION OF THE FATIGUE DESIGN OF A NEW ZIGZAG PIPELINE

A new 18" API-5L-X52 steel zigzag pipeline has been designed to transport heated products during a 40 year service life, considering fatigue as a possible failure mechanism following an up-to-date recommended practice for the fatigue design of welded structures [14]. For safety reasons, the methodology presented above was used to cross-check the predictions made based on the **SN** (linear elastic) traditional design philosophy used in that recommended practice.

Oligocyclic push-pull fatigue tests were carried out under axial strain control on **10mm** diameter specimens machined from a section of an API-5L-X52 steel tube, using a **100kN** computer-controlled servo-hydraulic testing machine. At least two specimens were tested at each strain amplitude, and to obtain the **εN** curve about fifty specimens were tested under deformation ratios varying from $\mathbf{R = -1}$ to $\mathbf{R = 0.8}$, see Fig. 6. The test frequency varied between 1 and 10 Hz, with a minimum of 500 points per cycle sampled by the data acquisition system, to avoid the interpolation problem when studying the elastic-plastic hysteresis loops. The module method (according to ASTM E 606-92 standard) was used to determine the fatigue life. The measured material properties are shown in Table 5. Note in that Figure that this X52 steel is almost insensitive to the deformation ratio, in special for short lives. Morrow's $\mathbf{\Delta\varepsilon/2 = [(\sigma_f' - \sigma_m)/E]\cdot(2N)^b + \varepsilon_f'(2N)^c}$ elastic strain-life equation, which includes the mean stress $\boldsymbol{\sigma_m}$ effect only in Coffin-Manson's elastic term, was found to best fit these experimental data. Morrow's fitted equation for $\mathbf{R = -1}$ (when it coincides with Coffin-Manson's equation, since $\boldsymbol{\sigma_m = 0}$ in this case) is plotted in Fig. 6. It is worth emphasizing that $\boldsymbol{\sigma_m}$ is the mean stress acting on the hysteresis loops, not the nominal mean stress.

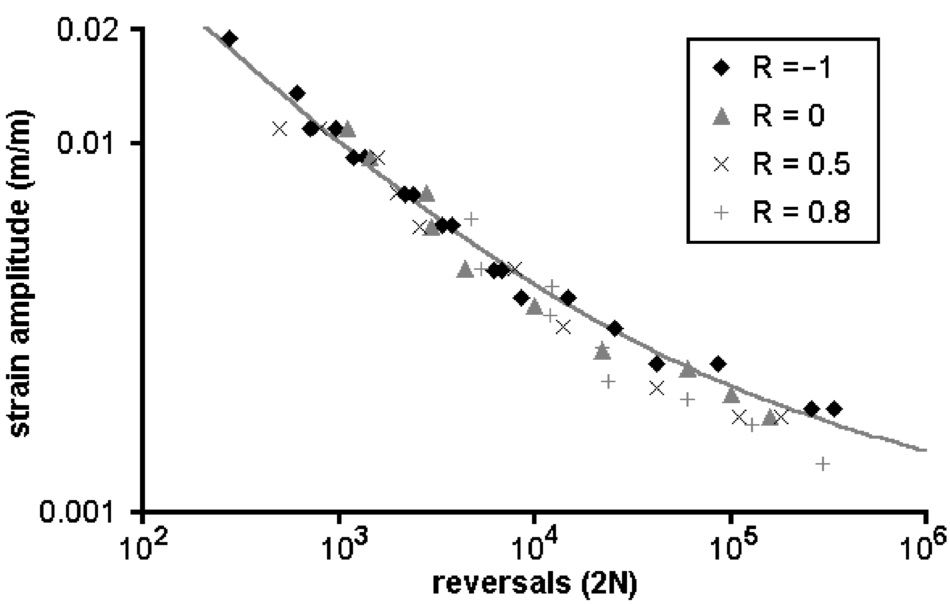

Fig. 6: Measured and fitted strain-life data for the API-5L-X52 steel.

Table 5: Measured mechanical properties of the API-5L-X52 steel.

E	**200 GPa**
S_u	**527 MPa**
S_Y	**430 MPa**
S_{Yc}	**353 MPa**
H_c	**925 MPa**
h_c	**0.155**
σ_f'	**720 MPa**
ε_f'	**0.31**
b	**0.076**
c	**0.53**
ΔK_{th} **(R = 0.1)**	**8.0** $MPa\sqrt{m}$
da/dN (R=0.1)	$2 \cdot 10^{10} \cdot (\Delta K - 8)^{2.4}$ **m/cycle**

The FCG tests of this API-5L-X52 steel were performed under LEFM conditions using C(T) specimens, **50mm** wide by **10mm** thick. Pre-cracking was made under $\Delta K = 20\ MPa/m^{1/2}$ until reaching $a_0 = 12.55mm$ ($a_0/w = 0.25$). Testing was conducted using the same equipment described above. Crack size was measured within a **20μm** accuracy by the Back Face Strain technique, using a 5mm 120Ω strain gage. The measured properties are also listed in Table 5.

The welded structure code used in designing this pipeline was the DNV RP-C203 [14]. It recommends a modified version of the traditional Miner's rule to account for the fatigue damage accumulation: $\Sigma n_i/N_i = \alpha$, where n_i is the number of cycles of the i-th load event applied to the structure, N_i is the number of cycles the structure could endure if it was loaded by that type of event only, and $\alpha \leq 1$ is a factor which depends on the consequences of an eventual failure. The (at least in principle) conservative value used in this design was $\alpha = 0.1$, which should correspond to an expected safety factor in life $\phi_n = 10$. Constant temperature was assumed for all pumped heated products, and in this way the pipeline was designed to endure **62400** identical thermal load cycles, which would correspond to a 400 year expected fatigue life assuming three heating-cooling cycles per week. To calculate the admissible stresses in the pipeline, fatigue SN curves **B1** and **F3** of the RP-C203 were used for the base material and for the welded joints, respectively. Curve **B1** is given by $N\Delta\sigma^3 = 3.26 \cdot 10^{12}$ and curve F3 by $N\Delta\sigma^3 = 1.40 \cdot 10^{11}$.

The admissible stress range in the welded joints is, therefore, $\Delta\sigma_{ad}(wj) \cong 131MPa$, which limited the operational temperature of the pipeline to $\theta_{max} = 75^oC$, a relatively low value. But this $\Delta\sigma_{ad}$ is probably too conservative even from an **SN** point of view, since a new zigzag pipeline could be constructed without any welds close to the points where the bending stresses in its curved parts are high. This could be achieved by forcing the longitudinal welds to be close to the neutral axis of the pipeline in those critical regions. In the base metal the admissible stress was limited by the nominal yield stress of the X52 steel, therefore $\Delta\sigma_{ad}(bm) \cong 360MPa$, a value which is, by the way, a little larger than the measured cyclic yield stress S_{Yc}. Since the thermal loading was assumed pulsating, this nominal S_Y limit is again probably too conservative, because to avoid any local yielding in the first thermal load cycle in an already yielded tube (and strain hardened by its cold work fabrication process) would do no significant harm. Controlled strain hardening and introduction of compressive residual stress are, indeed, two well known processes for increasing fatigue life.

To verify if the chosen $\Delta\sigma_{ad}(wj)$ would also be appropriate from an **εN** point of view, predictions of the fatigue crack initiation life were made considering:

(i) the measured properties of the API-5L-X52 steel,
(ii) the effect of the rough weld fillet surface on the *elastic* part of the **εN** curve,
(iii) that both high residual stress σ_{res} and a pit-like stress concentrator K_t could be present on the critical weld fillets, and
(iv) that this critical point in the weld fillet is subjected to a *nominal* stress range $\Delta\sigma_n = \Delta\sigma_{ad}(wj) = 131MPa$.

The value $\sigma_{res1} = S_Y = 430MPa$ could be chosen for simulating the high tensile residual stresses always present in non stress-relieved weld fillets but, assuming that the weld fillet and the base metal properties are similar, this value is probably too optimistic since the strain hardening of this X52 steel is non-negligible. Therefore, a second value $\sigma_{res2} = 500MPa$, which corresponds to an approximately **2%** residual strain in the fillet, seems a more realistic guess.

Even when a high but possible $K_t = 4$ is used for simulating the stress concentration effect, the life predicted is still **N = 20800 cycles**, which corresponds to more than 3 times the 40 year design life (but it is only around one third of the **SN** life actually used in specifying the $\Delta\sigma_{ad}(wj)$ for the pipeline, which was calculated to endure 62400 cycles).

Therefore, the conservative procedures (because of the huge **10x** factor used in the fatigue design life) based on **SN** design principles, which do not explicitly consider plasticity effects, do appear to be safe in this case. However, this is due to the very *low* sensitivity of this particular X52 steel to the mean load effects. If the material was better described by, e.g, the Smith-Topper-Watson model, its predicted life considering plasticity effects would be *less than half* the 40 year design life.

CONCLUSIONS

Fatigue is not a common problem in pipelines. However, fatigue cannot be ruled out beforehand as a potential failure mechanism in pipelines subjected to high but localized bending stresses when they transport heated products. These problems cannot be well simulated by the traditional (**SN**-based) welded structure codes or recommended practices, which do not explicitly consider possible cyclic plasticity effects at the roots of highly probable corrosion pits or similar surface defects. In this way, a methodology was proposed in this work to evaluate the residual life of zigzag pipelines from a more realistic point of view.

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