FATIGUE VALIDATION PROCESS FOR A HELICOPTER-MOUNTED TURRET

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Author's abstract. The article concerns the fatigue qualification of a helicopter-mounted turret. The multi-scale process developed by Nexter from 1997 [1] onwards was applied. In this application case, it is based on measurements carried out in flight on the carrier and characterisations of the turret (Modal analysis, digital simulations, etc.).

1 INTRODUCTION

This publication describes the multi-scale process developed by Nexter from 1997 [1, 2] onwards making it possible to guarantee the mechanical strength of a helicopter-mounted Nexter Systems turret on the basis of a customised mechanical environment specification [3].
The first stage of the process consisted in adjusting the model on the basis of the experimental modal analysis results: this analysis performed for several stress levels made it possible to update behaviour non-linearities (evolution of natural frequencies as a function of the loading levels).
The second stage of the multi-scale process consists in evaluating the dynamic response of the structure followed by the location of the critical areas (weak point location, risk of cracking). The last stage concerns the thorough study of the risk areas by including precisely the details and the geometric singularities of the sensitive areas identified previously; this stage results in an estimation of the fatigue life crack initiation.

1.1 Abbreviations and terminology

The specific acronyms, abbreviations or terms used, which may not be understood by the reader, are listed here.

PSD	:	Power Spectral Density
A.M.E./A.M.A		Experimental/analytical modal analysis

1.2 Reference axes

The reference system used in this document is the carrier's:

- X axis: longitudinal axis in the forward direction of the helicopter,
- Y axis: transverse axis,
- Z axis: vertical axis, upwards.

The axes are such that (X, Y and Z) form a direct trihedron.

2 LIFETIME CALCULATION METHODOLOGY

2.1 Multi-scale approach

The search for the sensitive areas fits into the multi-scale calculation process put in place at Nexter Systems on the calculation of structures. This process begins with the quantitative description of the operational configurations of the mechanical systems and ends with the prediction of the lifetime of the structures. Three levels of scales make up the process:

- the scale of the mechanical system (vehicle, weapon system, etc.)
- the scale of the mechanical components or structures making up the mechanical system
- the scale of the structural details encountered in the mechanical components and structures (welds, geometric details, etc.).

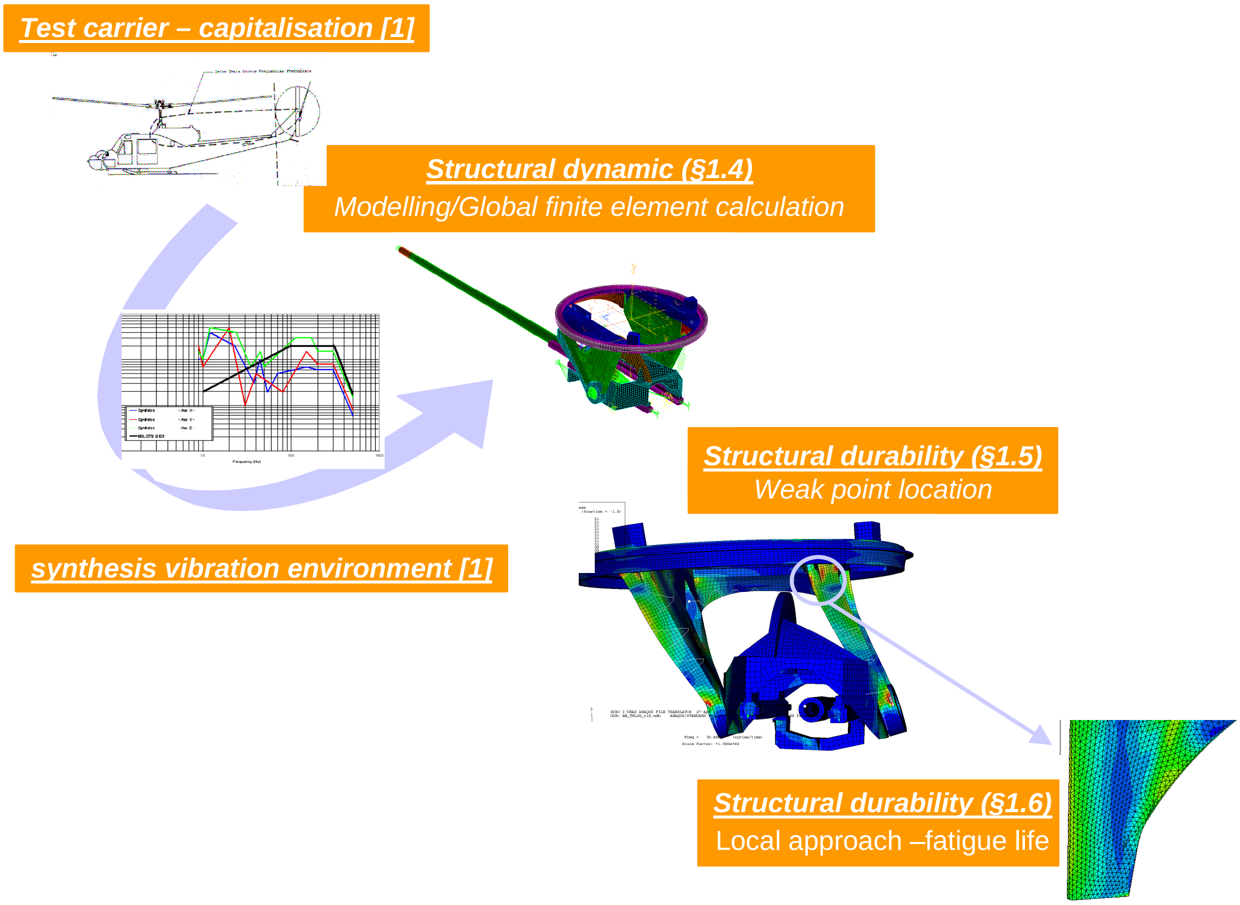


Figure 1: multi-scale process

2.2 Structure calculations – mapping of stresses

The objective of this study is to estimate the mechanical strength of the turret subjected to a random and harmonic environment. These signals are from the Nexter Systems customised specification utilising the actual levels identified on carrier and making it possible to dimension the structure to what is strictly needed [3]. The simulation consists in applying in the 3 directions a harmonic signal superimposed on a white noise type random signal. The time signal to be applied on the turret is produced by adding the time signals of sine waves to the time signal from the PSD; a tool developed by NS makes it possible to generate one or more time signals from one PSD;

Our structure is discretised and represented by a system with n degrees of freedom. The matrix system satisfies the following dynamic differential equation:

$$M\ddot{x}(t) + C\dot{x}(t) + Kx(t) = f(t)$$

where:
- M: mass matrix of the system (dimension N x N)
- C: damping matrix of the system (dimension N x N)
- K: stiffness matrix of the system (dimension N x N)
- $f(t)$: loads applied on the structure (dimension N x 1)
- $x(t)$: response of the structure in terms of movements and/or rotations according to the degrees of freedom.

The principle of the modal superimposition method is based on the projection of the equation of the motion on the basis of the natural modes of the structure.
In the case of resolution on the actual basis of the modes of the associated conservative system, there is not automatically diagonalisation of the damping matrix by projection on the actual modal matrix: on the other hand, if the assumption of decoupled modes is satisfied (no overlapping of the band width at - 3 dB) and if the damping is low (<10%), one will be able to disregard the extradiagonal terms of the generalised matrix of the dampings which will then be diagonal; the modal equations will then be decoupled which makes the search for a solution easier.

The solutions of the homogeneous system below correspond to the natural modes of the structure. In general, because of the cost of calculation, only *m (m<<N)* natural modes can be extracted.

$$\omega^2 M\varphi = K\varphi$$

i.e. *m* pairs consisting of *m* natural frequencies and *m* modal deformations.

By multiplying *M* and *K* on the right and on the left by respectively the truncated modal matrix (to the number of modes m considered) and by its transposition, one obtains diagonal generalised matrices, per orthogonality property of the modes:

$\varphi^T M\varphi = [m]$ diagonal matrix *(m, m)* of the generalised masses

$\varphi^T K\varphi = [k]$ diagonal matrix *(m, m)* of the generalised stiffnesses

One deduces from this the natural pulsation $\omega_\nu = \sqrt{\dfrac{k_\nu}{m_\nu}}$ for the mode η.

In general, the matrix of the dampings is not diagonalised by this operation: the full modal damping matrix $\varphi^T C\varphi = [c]$

By using the truncated modal basis (*m<N*) and assuming that the movement can be written as a linear combination of the modes, one has the following fundamental expression:

$$x = \sum_{i=1}^{m} \varphi^{(i)} \cdot q_i(t) = \varphi q$$

where: $q = \begin{bmatrix} q_1(t) \\ \mathrm{M} \\ q_m(t) \end{bmatrix}$ vector of the modal generalised coordinates and $\varphi = \left[\varphi^{(1)} \quad .... \quad \varphi^{(m)}\right]$ truncated modal matrix.

By premultiplying the equation of the motion by φ^T , *N* equations of the motion for the system studied are obtained:

$$[m]\ddot{q} + [c]\dot{q} + [k]q = \varphi^T F(t) = [f(t)]$$

with *f (t)* vector of the modal forces.

Under certain assumptions, it is possible to decouple the modal equations from the motion, that is, to have a diagonal damping matrix of the form:

$$\varphi^T C \varphi = [c_{ii}] = [2 m_i \omega_i \xi_i]$$

with ξ_i reduced damping factor of the mode i.

One is then called upon to process a system of *N* second-order differential equations independent of each other, each corresponding to a harmonic oscillator with 1 degree of freedom and a particular mode. For the equation corresponding to mode i, one has:

$$m_i \ddot{q}_i + c_i \dot{q}_i + k_i q_i = f_i(t)$$

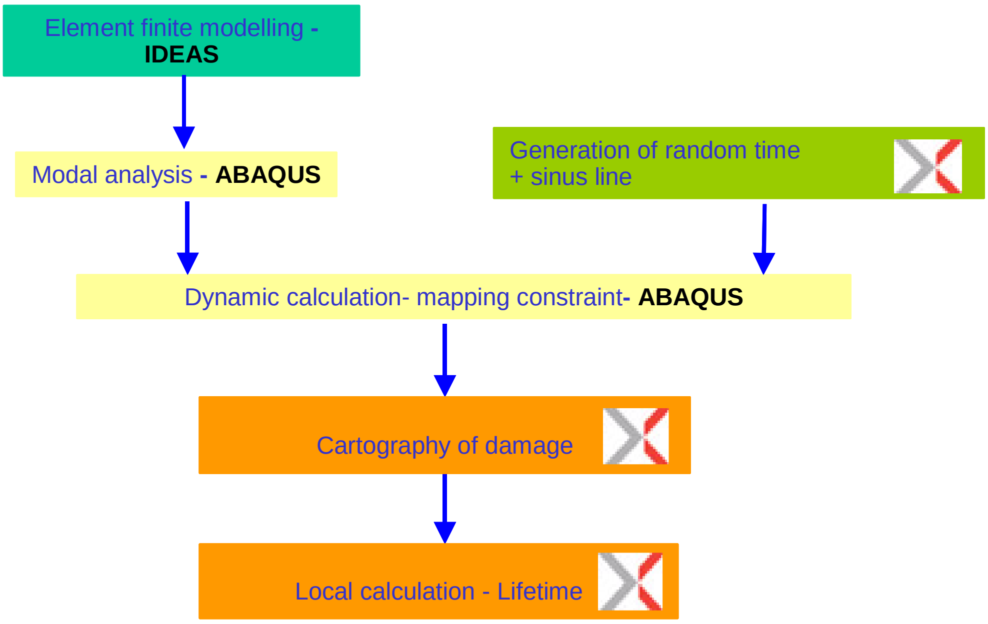


Figure 2: Basic diagram of the structure and fatigue calculations on the turret

2.3 Location of the areas at risk of cracking

The "global" calculations are performed by the fatigue software developed within Nexter Systems. The location method supplies a mapping of damage over the whole of the structure. At this stage, numerous parameters are not taken into account for the calculation of this damage (elasto-plastic correction, concentration of stresses in the specific areas), because the calculation times would then be too considerable; the damage is therefore estimated globally, it does not make it possible to determine a precise lifetime. Reading this mapping makes it possible to identify visually the areas at risk of cracking by fatigue where a thorough study will need to be carried out including precisely the details and the geometric singularities of the area to be studied.

2.4 Estimation of the lifetime by local approximation [1]

At this stage, one endeavours to refine the estimation of the lifetime by a local calculation: the sensitive areas identified previously are meshed more finely incorporating for example structural details not taken into account in the global calculation: weld beads, fillets ... A finer stress mapping is then obtained by the structural zoom method (Abaqus): this consists in applying to the frontier nodes of the local model the movement field recorded on these same nodes at the time of the global calculation.

The local stresses are often greater than the limit of elasticity and it is necessary to calculate the plastic local response thanks to an elasto-plastic correction module. This local stress corrected for plastification is then used so as to determine a fatigue stress based on a Dang Van facet criterion: this stress facet criterion consists in calculating, on each facet of normal *n*, the maximum over the cycle of the linear combination of the amplitude of the shear stress $\tau_a^n(t)$ and of the hydrostatic stress $\sigma_H(t)$:

$$\sigma_{fat}^n = \underset{cycle}{Max}\left(\tau_a^n(t) + \beta\sigma_H(t)\right)$$

The cycles are extracted using the Rainflow method. Each recorded cycle is characterised by the minimal value and the maximal value of this shear stress, and by the cycle start and end times. This makes it possible to calculate the amplitude of the cycle. The combination with the hydrostatic stress calculated between the cycle start and end times then gives the criterion corresponding to the cycle extracted.

The damage is then calculated for each cycle extracted using the material properties (Basquin's law) and it is then accumulated over all of the cycles by a linear accumulation (Miner's law).

$$d^n = \sum \frac{1}{\left(\dfrac{\sigma_{fat}^n(n_i)}{\sigma_f^{'}}\right)^{1/b}}$$ where with *b* and $\sigma_f^{'}$ material parameters.

The damage is calculated on each of the facets. The final damage is then the maximum damage over all of the facets.

$$d = \underset{n}{Max}(d^n)$$

The lifetime N is obtained by inverting this damage over the duration of the loading: $N = \dfrac{t}{d}$

A flow chart makes it possible to summarise the calculation process.

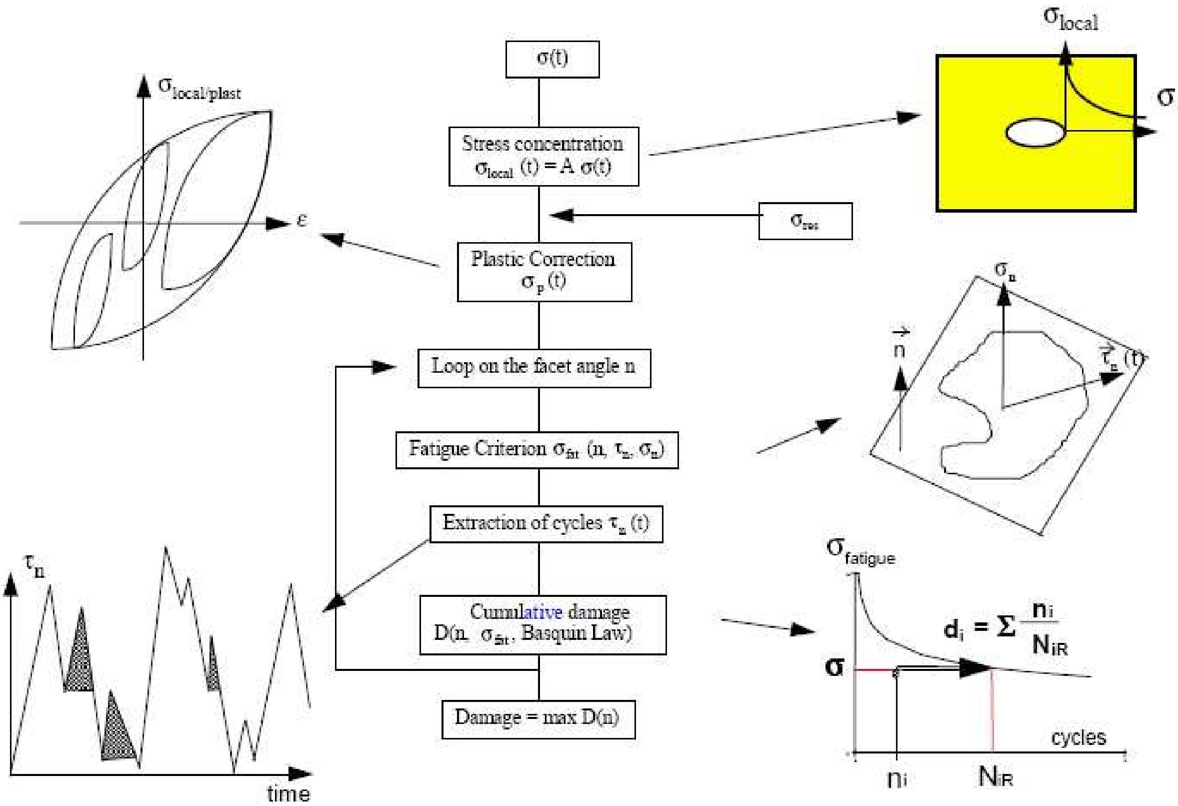


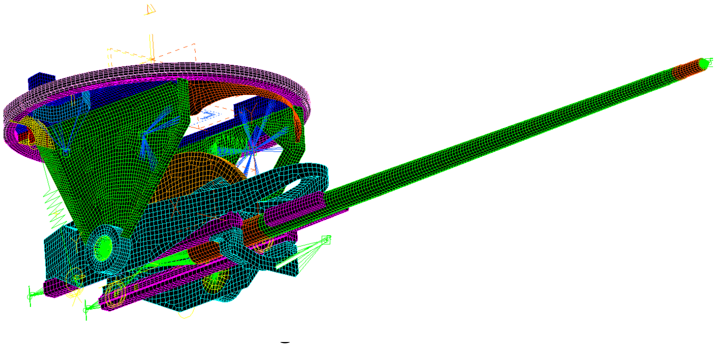
Figure 2: **Calculation of fatigue damage by local approach**

2.5 Safety coefficient

In view of the string of potential imprecisions over the whole of the process, a safety coefficient on the lifetime must be taken into account. This safety coefficient reflects the errors made on the estimation of the stresses resulting from the uncertainties/assumptions on the loads, the conditions at the limits, the modal dampings and the fatigue properties of the materials. For castings in aeronautics, one systematically assigns to stress tensors a coefficient of 3: this coefficient corresponds to the cast factor used in aeronautics for the dimensioning of fatigue parts: it makes it possible to take account of any inclusions or defects present in the material.

3 MODELLING

The main sub-structures are meshed, namely the fork, the ring, the cradle and the barrel. The meshing comprises shell elements (S3R and S4R) and volume elements (C3D6 and C3D8R). The stiffnesses of bearing, of laying drive unit, of the recoil mechanisms/links and of the rams/actuators are introduced into the model. The slide link is also represented.



The base excitation type calculation is performed by temporal modal superimposition; a sufficient number of modes is taken into account in the calculation and the modal dampings are estimated from results of trials. The finite element model of the turret is produced with Ideas NX5 and the calculations are performed with Abaqus 6.7.1.

4 MODAL ANALYSIS

4.1 Comparison of calculated modes / experimental modes

In the first instance, the natural frequencies and modal deformations of the turret calculated are adjusted in relation to the results of trials. Most of the analytical natural modes at 100 Hz are well correlated with those identified.

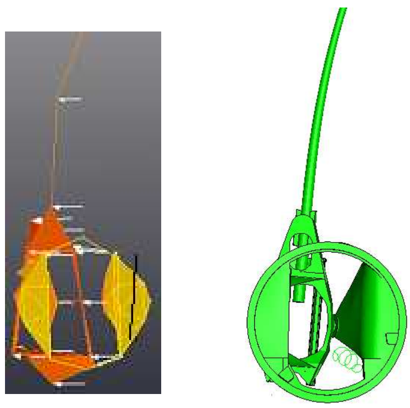


Figure 3: Experimental/analytical mode - Rocking of the cradle in the Y axis

A difference remains nevertheless in the cradle’s rocking mode : this is calculated at a frequency 10 Hz less compared with the measurement (Figure 4). At the time of the trials, a sine sweep performed under various levels of loading highlighted the frequency slip of this mode as a function of the stress level: it loses 7 Hz when the stress increases from 0.01g to 0.1g. This is the sign of a structural non-linearity due to the numerous plays present in this structure consisting of assembled parts.
Now, if one considers that the stresses specified are greater than these levels, and that the results of previous qualification trials (FRF) present a rocking mode similar to the current calculation, we can take it that our current model is representative of the rigidity of the turret in endurance trial.

Thus, the various models constructed in the course of the adjustment phase are valid for a specific use area : the choice of the model to be utilised for the calculations is suited to the level of stress actually applied on the structure.

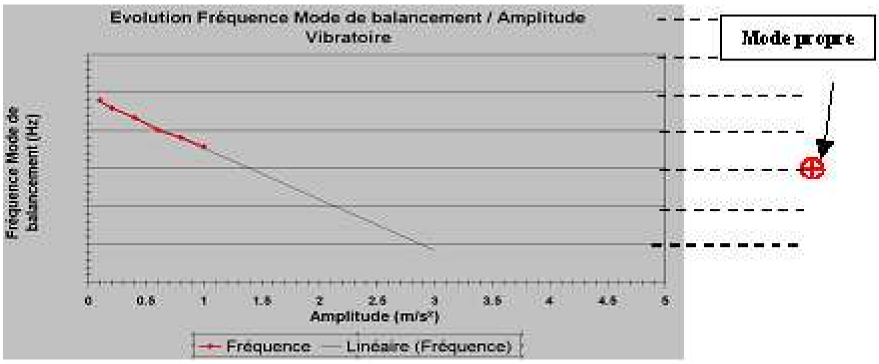


Figure 4: Evolution of the frequency of the rocking mode as a function of the stress level

4.2 Identification of damping

The A.M.E. indicates damping levels less than 5 %. One notes however on the FRFs from the sine sweep at various levels that the damping associated with the rocking mode increases with the stress level (Figure 5).

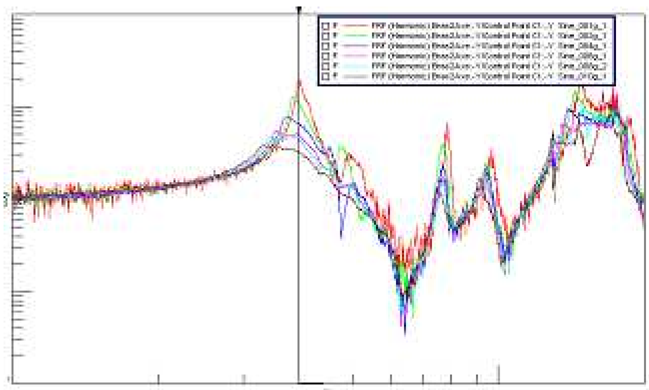


Figure 5 FRFs for the various stress levels (sine sweeps)

The damping is then estimated graphically from the previous results of endurance trials: this approximation is carried out on the basis of the resonant FRF on several sensors and rests on the following analytical relationship which links the overtension factor Q (evaluated as the ratio between the resonant frequency (or pulsation) and the band width at –3 dB) and the reduced damping factor ξ:

$$Q = \frac{\omega_0}{\omega_1 - \omega_2} \approx \frac{1}{2\xi}$$

This method is used only in the absence of damping data; it leads to an approximation of the modal damping factor. It is the most penalising value of the reduced damping which will be adopted for the calculations.

5 RANDOM AND HARMONIC SIMULATIONS

The fatigue calculations in a vibratory environment are produced along the 3 axes for the customised trial specifications over a reduced duration. For each calculation, the lifetime will be estimated without or with Cast Factor.

5.1.1 Global behaviour of the turret

Stressed transversely, the response of the turret consists in a rocking along the Y axis. The movement amplitude at the trunnions in the three directions as well as the deformation of the structure are given below.

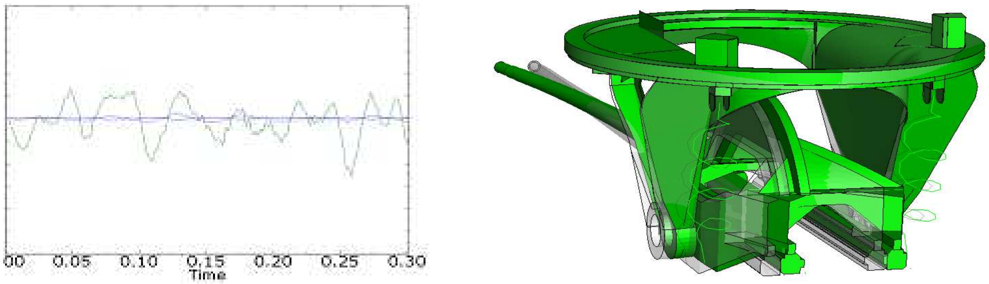


Figure 6: Movements of the trunnions and amplified view of the deformation (stress along Y axis)

5.1.2 Calculation of location of the hot spots

The initial global calculation makes it possible to obtain the distribution of the damage levels over the whole structure (Figure 7). This mapping therefore makes it possible to locate the areas at risk of crack initiation.

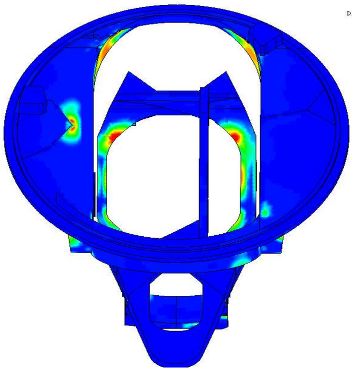


Figure 7 Location of the most stressed areas

5.1.3 Determination of the local stresses by the structural zoom technique

A 3D finite element model, representative of the local geometry of a risk area identified previously, is produced. The local calculation is therefore limited to the portion of the structure to be studied; the stresses are evaluated by applying at the limits of the 3D meshing the movements from the global shell model.
The following figures present the results in terms of Von Mises stresses for the global and local models. The local models were calculated on the basis of the movements of the global model at the time of the dynamic simulation of the turret.

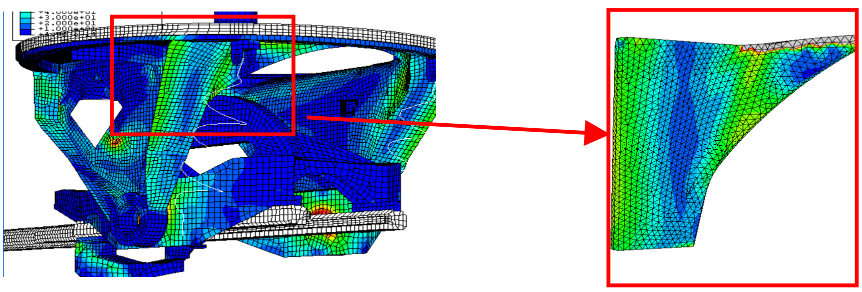


Figure 8: Structural zoom (right view) on the basis of the shell global model (left)

Thus, the local calculations show that the shell meshing describes stress levels very similar to the 3D models for the areas presented above (the difference does not exceed 10%).
The estimations of the lifetime in various areas indicate that:

- ✓ the lateral axis Y is the most penalising direction;
- ✓ the fatigue strength of the turret is guaranteed for the 3 axes, without and with Cast Factor type safety coefficient of 3;

6 CONCLUSION

The multi-scale process developed by Nexter from 1997 onwards was implemented on a helicopter-mounted Nexter Systems turret, the objective being to dimension the turret in keeping with its future environment. The stresses adopted in order to guarantee the mechanical strength are from a customised mechanical environment specification.
The lifetime estimation process is carried out in several stages. It is a matter first of all of constructing a model representative of the behaviour of the structure: adjustment in relation to the trials made it possible to define a customised model, suited to the level of the stresses. The next stages consist in locating the risk areas and in estimating the crack initiation lifetime for the environment specified.

7 REFERENCES

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