AN OVERVIEW ON FATIGUE PROBLEMS IN THE CAR INDUSTRY

A. BIGNONNET

The automotive industry is confronted with the fatigue problem for most car components.

At the design stage, the service loading is processed using a cumulative damage method to provide equivalent damage loading. The fatigue assessment calls for two types of fatigue criteria : high cycle fatigue criteria for components which are working in elasticity like chassis parts or crankshafts, low cycle fatigue criteria for component which may sustain inelastic strains such as cylinder heads or exhaust pipes. The methods used are illustrated with examples in the following.

INTRODUCTION

The automotive industry is confronted with the fatigue problem for most car components : chassis parts, engine and body. The fatigue assessment of an automobile security parts must cover the whole life of the vehicule because there is no possibility to monitor fatigue damage by regular control once the car is sold, as is the case in the aircraft industry for example.

The first difficulty is the knowledge of the loading history the vehicule will undergo during it is whole life and the scatter coming from the different usages and the different customer behaviour. The fatigue assessment will be based on a reliability approach taking into account this variability and the fabrication scatter.

At the design stage, the service loading is processed using a cumulative damage method to provide equivalent damage loading. The fatigue assessment calls for two types of fatigue criteria : high cycle fatigue criteria for components which are working in elasticity like chassis parts or crankshafts, low cycle fatigue criteria for component which may sustain inelastic strains such as cylinder heads or exhaust pipes. The methods used are illustrated with examples in the following.

Peugeot S.A., 78140 Vélizy, France

LOADING SPECIFICATION

Maximum loading and fatigue loading

The loading specifications and resistance criteria are based on the stress - strength reliability approach. From measurements performed on customer cars, loading histories at the wheel base are given in the three directions : longitudinal, lateral and vertical (X, Y, Z). A statistical analysis on a large population allows the definition of the loading distribution and of an "objective customer" with a known severity (mean, μ_c, plus a given number of standard deviations σ_c in the case of a gaussian distribution). The loading histograms corresponding to this objective customer yield the following :

- the maximum values of the loading in each direction, events that could occur a few thousand of times in the life of the vehicle.
- all the load cycles on the whole life of the vehicle (including the maximum values), which will produce the fatigue damage.

On the one hand, these loadings must be withstood without permanent deformation (the material remains in its elasticity range) and on the other hand, they must be withstood without fatigue degradation, (no crack can be detected). The acceptance of individual components, in validation testing or in predictive calculation is based on the stress - strength reliability approach, as shown in a further paragraph.

Equivalent fatigue loading

To analyse the loading and to provide useful information to the designers, the fatigue loading recorded on vehicles is transformed into an equivalent loading. The whole approach is based upon the calculation of an equivalent fatigue alternate loading of constant amplitude defined for 10^6 cycles which represent the number of kilometers covered by the car in its whole lifetime. This calculation is made using a Rain-Flow count, Whöler or Basquin normalized curves, the Gerber parabola and the Miners' rule [1]. This equivalent fatigue loading is established at the objective customer level, F_n, such that $F_n = \mu_c + \alpha\,\sigma_c$. The probability of finding a more severe owner is given by the normal law. For example if $\alpha = 4.1$, therefore $\mathrm{Prob}\left(\mathrm{severity} > F_n\right) = 1/50000$. The fatigue calculations and testing are performed with this level of severity.

The "stress-strength" method

Once the distribution of "Stress" and "Strength" are known, it remains the determination of the risk R of failure in service, in order to give the predictive reliability F = 1-R, figure 1.

For an easier analysis of the results a relative scatter parameter is introduced in each of the distributions : $\rho = \dfrac{\sigma_c}{\mu_c}$ and $q = \dfrac{s_r}{m_r}$, m_r and s_r being the estimation of the mean and standard deviation of the strength of the components), ρ is representative of the shape of the "Stress" distribution, while q is characteristic of a component family and its fabrication process. m_r and s_r are homogeneous to the applied forces and they can be normalized by the testing reference : $m_r^* = \frac{m_r}{F_n}$. Therefore the risk can be written :

$$R = \mathrm{Prob}\left(u < -\frac{\mu_Z}{\sigma_Z}\right) = f\left(\alpha, \rho, m_r^*, q, N, \gamma\right)$$

The estimation of the component strength scatter, s_r, is extremely important. An error of 20% on s_r could bring a factor of 10 on the estimation of the Risk *R*. The lower the number of tested components, the higher the value of σ_r and the greater the risk value. Therefore it is interesting to work with the parameter *q* which characterises the component and its fabrication process.

A data base derived from a large number of tests performed on components on specimens can provide a reliable value of the relative scatter parameter *q*. In this case, only the determination of the mean value of the "Strength" distribution is necessary. This can be carried out with a limited number of components. It is no longer necessary to take into account the number of tested components (m_r and *q* are considered to be representative of the whole components). The calculation of the risk *R* is more precise and is not penalized by a correction due to the number of tested components. The objective mean strength value which should be reached, depending on *q* and *N*, is given on figure 2. This figure shows how important the relative scatter of the process is. For example if the relative scatter $q=0,08$ is representative of the fabrication scatter, the design objective is reached with a mean strength value $m_r^* = 1{,}27$. It should be $m^*_r = 1{,}35$ if 20 components are tested.

HIGH CYCLE FATIGUE ASSESSMENT

Multiaxial design criteria

Problems arising from the industrial use of a fatigue criterion. Many components must resist high cycle fatigue, either in engines (crankshafts, connecting rods), or in chassis parts (steering knuckles, suspension arms, or wheels). Most of these components are subjected to a multiaxial state of stress. The origin of the multiaxiality is threefold : external loading, geometry of the structure, residual stresses, which by nature, are multiaxial

Materials considerations : limited number of parameters, easy to characterize.

Numerical considerations : computerized in a fatigue routine tied to a finite element code.

In the automobile context, it is clear from specifications, that any component which will resist high cycle fatigue cannot suffer cyclic plasticity at a macroscopic scale. In any case, the material will tend rapidly to an elastic state due to elastic shakedown of the structure.

Our approach is based on the theoretical work of Dang Van [2], the last developments of which have been presented in [3]. The expression of the criterion is :

$$\tau_a + ap(t) < B$$

where τ_a is the shear stress amplitude on the critical plane, p(t) is the maximum hydrostatic pressure reached during the cycle, a and B two material coefficients, which can be identified from two separate fatigue tests, such as reversed shear and alternate bending.

Fatigue assessment of a suspension arm. To illustrate the application of the Dang Van's criterion an example of fatigue design for a suspension arm is shown in figure 3.

The component is linked to the engine frame by two elastic articulations and connected to the steering knuckle with a ball joint. The equivalent fatigue loading is applied to the ball joint following X and Y directions : $F_x = F_a$ and $F_y = F_a (-0{,}5 + \cos\omega t)$.

Fatigue Life Prediction. The fatigue criterion requires that the fatigue cycle be described at several discrete times. In the present example, six values of ωt have been chosen $\omega t = 0, \pi/_2, , \pi, 3\pi/_2$ and two intermediate values, such that $F_X = F_Y$.

The corresponding stress tensors are then calculated by Finite Element Analysis, and stress results are analysed with the Dang Van's fatigue routine.

At the design stage, the component is accepted if, under the specified equivalent loading, the load path at the most critical point is below the “design line”, i.e mean value minus β x standard deviation (β comes from the reliability approach).

THERMOMECHANICAL FATIGUE ASSESSMENT

In spite of major advances in the understanding of fatigue phenomena at a local microscopic level and a large amount of experiments on specimens, robust criteria at a global macroscopic level to predict low cycle thermomechanical fatigue failure on structures are still missing. There are essentially two difficulties to attain this objective : a formulation of the material behaviour which gives access to an accurate description of the mechanical values which control the fatigue phenomenon in a structure and a failure criterion, derived from simple tests, which provides a reasonable estimation of the lifetime of the structure.

The constitutive law should permit computer simulations in a acceptably small computation time of elastoviscoplastic structure subjected to complex thermomechanical loadings. The fatigue criterion should be capable of explaining isothermal and anisothermal LCF failure for a large range of temperatures and for multiaxial loadings. These considerations have suggested a certain number of assumptions :

- the material behaviour can be modeled by a simple elastoviscoplastic constitutive law, which permits the representation of the cyclic response of the structure under thermomechanical loading,
- the fatigue criterion can be based on an intrinsic quantity permitting the prediction of failure under isothermal and anisothermal multiaxial loading. The dissipated energy per cycle, seems to give good results.

An interesting interpretation of the cyclic behaviour of the material is presented in Skelton [4]. These results [4] suggest that the cumulated dissipated energy to the cyclic response stabilization can be used as a crack initiation criterion in low cycle fatigue. For design purpose, the basic assumption is that the failure of the representative elementary volume corresponds to the failure criterion. Using the ideas developed by Skelton, low cycle fatigue and thermomechanical fatigue tests on cast iron were analysed through the dissipated energy on the stabilized cycle [5]. The criterion itself has been established by linear regression between the dissipated energy and the number of cycles to failure.

The anisothermal behaviour of components is simulated using the elastoviscoplastic constitutive law. A typical FEM model for an exhaust pipe contains about 20000 hexaedric volume elements. The thermomechanical loading is given by the temperature variation and stresses and strains distribution were obtained in all the structure with a reasonable computation time. The computed dissipated energy distribution over the structure, indicates precisely the critical area and the number of cycles to failure.

CONCLUSION

It is important to point out that mechanical structures usually have complex geometries and that they operate under complex loading. Therefore the description of the behaviour of the structure must be as precise as possible and should provide the stress and strain elements necessary for the subsequent fatigue analysis. This fatigue analysis must be appropriate and take into account the structural behaviour and specificities of mechanical design (multiaxiality of the stress fields, complex geometries) and must be computerized with a fatigue routine linked to a finite element code. To that purpose it is not necessarily useful to describe in detail the damage mechanisms of the material which are anyway very complex. The relative simplicity of the application of the methods proposed here and the limited number of necessary material data allows their use in design offices by engineers and designers who are not fatigue experts.

REFERENCES

(1) Morel, F., Catherin, G., Bignonnet, A., and Petit, J., Proceed. of Sollicitations en service et comportement en fatigue, Paris 25-26 mai 1993. Pub SF2M.

(2) Dang Van, K., Sur la résistance à la fatigue des métaux. Sciences et Techniques de l'armement. 3ème fascicule, pp 647-722, 1973.

(3) Dang Van, K.,. Introduction to fatigue analysis in mechanical design by the multiscale approach. In CISM courses, Udine, sept. 1997 "High-Cycle Metal fatigue in the context of Mechanical Design". Ed. I.V. Papadopoulos, K. Dang Van. to be published Springer-Verlag.

(4) Skelton, R.P. Mat. Sc. Tech. Vol. 7, pp 427-439, 1991.

(5) Charkaluk, E., Constantinescu, A., Bignonnet, A. and Dang Van, K., "On a failure criterion for thermomechanical multiaxial low cycle fatigue and its industrial application". Int. Conf. on Low Cycle Fatigue. Garmisch Partern Kirschen, sept. 1998

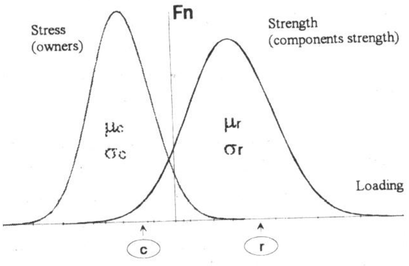

Figure 1 "Stress - strength" method illustration

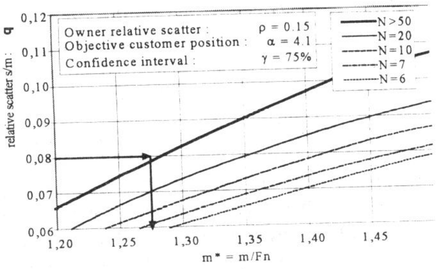

Figure 2 Determination of the objective mean stength from the relative scatter and the number of tested pieces.

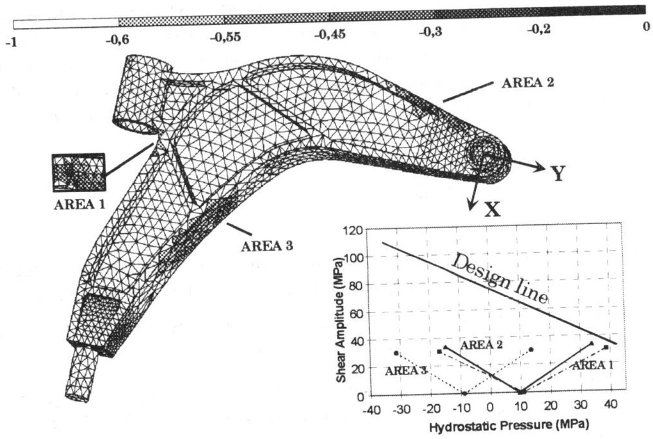

Figure 3 Fatigue assessment of a supension arm with the Dang Van criterion