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A METHOD TO EVALUATE THE SERVICE LIFE OF RUBBER SPRINGS IN ROLLING-STOCK

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Abstract: *The stress–endurance (S–N) method has been widely used in metal fatigue in the railway industry. When this method is applied to rubber-like materials, it seems not as successful as when it is applied to metallic materials. The continuum mechanics (total life) approach has been used to characterize the fatigue life in terms of the cyclic stress range (S–N curve). It is suggested that the effective stress,* σ_f*, can be used as a parameter (taking three principal stress ranges into consideration) to evaluate fatigue failure. The initial verification of this approach has been carried out on two types of rubber spring. The results have shown good agreement between the fatigue test of the components and the simulation based on this approach.*

1. INTRODUCTION

The stress–endurance (S–N) method has been widely used in metal fatigue in the railway industry. One of many successful case investigations is a dynamic real-time simulation of a bogie frame for metro vehicles [1]. This method seems to be applied less successfully to rubber-like materials than to metallic materials.

There are two ways of dealing with rubber fatigue caused by mechanical failure: continuum mechanics (total life) and fracture mechanics (defect tolerant) [2]. Roughly speaking, the total fatigue life in continuum mechanics is defined as the sum of the number of cycles to initiate a fatigue crack to some predefined size. Defect tolerance in fracture mechanics is based on the fact that there are inherent flaws in all engineering products. The useful fatigue life is defined as the number of cycles to propagate the dominant crack from this initial size to some critical dimension. The principal differences may be dependent on how the crack initiation and the crack propagation stages of fatigue are quantitatively defined.

The approach reported here attempts to link the complicated loading environment with a much simpler situation such as a uniaxial fatigue test. The basic principle is that the fatigue resistance of component A should be the same as that of component B when they have both undergone the same effective stress regime.

2. EFFECTIVE STRESS

To evaluate the fatigue life of components without reference to either geometry or loading conditions is necessary finding in a common base. Therefore, it is suggested that an effective stress, σ_f (a function of three principal stress ranges), be used as a parameter for this status to evaluate fatigue failure:

$$(1) \qquad \sigma_f = \sqrt{\sigma_1^2 + A\sigma_2^2 + B\sigma_3^2}, \ \sigma_1 > 0, \ \sigma_1 \geq \sigma_2 \geq \sigma_3, \ A \leq 1, \ -1 < B \leq 1$$

where: A, B - parameters of the effective stress; σ_f - effective stress; σ_1, σ_2, σ_3 - principal stress ranges.

The above definition assumes:

1. There is no fatigue damage when a point is under compression in all directions.
2. A is taken as positive when σ_2 is positive and B is taken as positive when σ_3 is positive.
3. The fatigue damage caused by stress in any one of the other two principal directions will not exceed that caused by σ_1.

This criterion includes all principal stress range components (σ_1, σ_2 and σ_3). Generally speaking, equation (1) describes an ellipsoidal failure envelope, as shown in Fig. 1. Any point on the envelope will give the same fatigue damage caused by repeated cyclic loading.

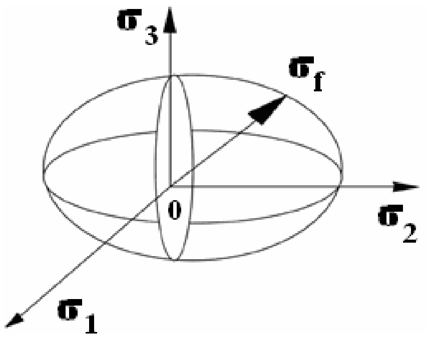

Fig. 1 Effective stress tensor

There are now some procedures under consideration to decide between components A and B. In one of the procedures it is suggested that, for a free surface, A and B be taken to have a maximum value of 1 for safety provided that $\sigma_2 > 0$, $\sigma_3 > 0$ and a value of 0 if $\sigma_2 \leq 0$, $\sigma_3 \leq 0$; i.e.

(2) $A = 0$ when $\sigma_2 \leq 0$, $A = 1$ when $\sigma_2 > 0$
(3) $B = 0$ when $\sigma_3 \leq 0$, $B = 1$ when $\sigma_3 > 0$

The direction d of the normal vector to the crack plane can be easily obtained by the following procedure. Let there be three principal stress ranges:

$\sigma_1 = [\sigma_{1i}, \sigma_{1j}, \sigma_{1k}]$
$\sigma_2 = [\sigma_{2i}, \sigma_{2j}, \sigma_{2k}]$
$\sigma_3 = [\sigma_{3i}, \sigma_{3j}, \sigma_{3k}]$
where: σ_{1i} (1, 2, 3), (i, j, k) projection of the principal stress ranges on to a tensor system.

According to equations (2) and (3) the direction d (perpendicular to the plane)

$d = [\sigma_{1i}, \sigma_{1j}, \sigma_{1k}]$ when $\sigma_2, \sigma_3 \leq 0$
$d = [a, b, c]$ when $\sigma_1, \sigma_2, \sigma_3 > 0$

where
(4) $a = (\sigma_{1i} + \sigma_{2i} + \sigma_{3i})/M$
(5) $b = (\sigma_{1j} + \sigma_{2j} + \sigma_{3j})/M$
(6) $c = (\sigma_{1k} + \sigma_{2k} + \sigma_{3k})/M$
(7) $M = \sqrt{(\sigma_{1i} + \sigma_{2i} + \sigma_{3i})^2 + (\sigma_{1j} + \sigma_{2j} + \sigma_{3j})^2 + (\sigma_{1k} + \sigma_{2k} + \sigma_{3k})^2}$

3 FATIGUE TEST UNDER UNIAXIAL LOAD

The purpose of this fatigue test is to create a base for the failure resistance of rubber springs. Rubber fatigue tests under uniaxial load were carried out in NITIJT-Sofia, Bulgaria physical test laboratory. The results are represented in the form of stress range against fatigue endurance (S–N curve), as shown in Fig. 2. Any point of the curve represents the component failure at a load level (in the form of the stress range) under a certain number of cycles.

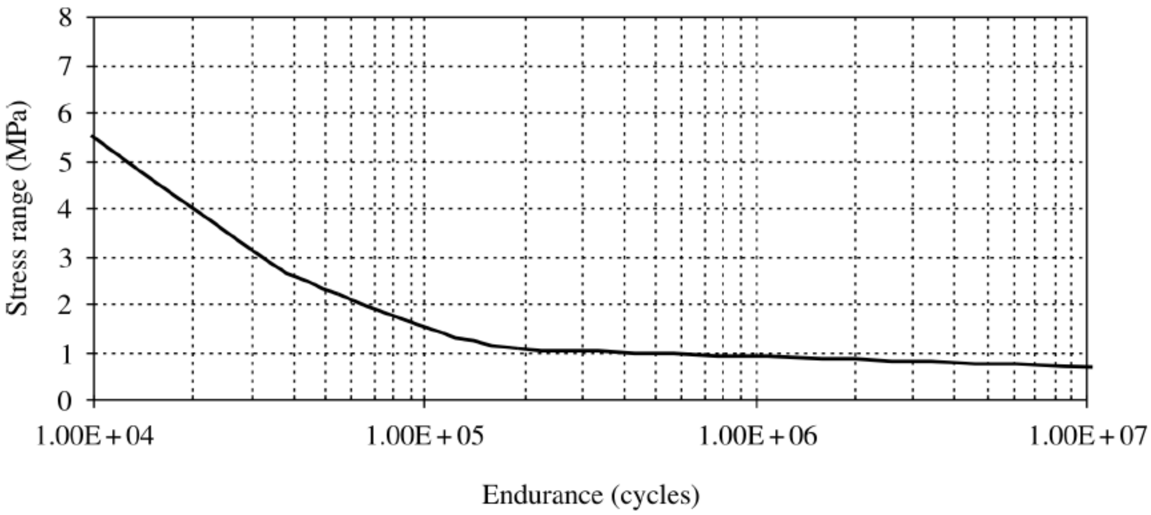

Fig. 2. S–N curve.

4 APPLICATIONS

Two case studies of rubber springs are reported here in order to validate the above methodology. One is a bearer spring and the other is a metal-rubber spring.

4.1 Bearer springs

These springs support the carbody mass in compression. Bogie lateral and rotational displacements shear the rubber spring in the horizontal plane. The springs may be fitted directly between the body and the bogie frame or between the body and the bolster.

A validation procedure has been carried out, which involves the real component fatigue test and a finite element (FE) simulation (Fig. 3). A stiffness comparison between the test and FE simulation is shown in Fig. 4. After 31000 cycles, four cracks appeared on the short rubber profiles at the end of the component. The lengths of the cracks were in the range 2–15 mm. All these cracks were developed from the corners.

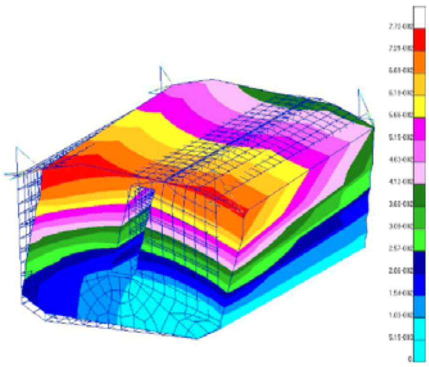

Fig. 3. FE simulation.

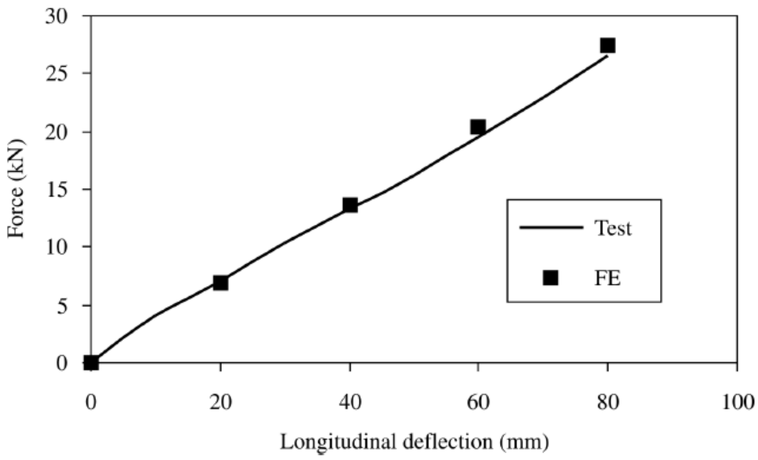

Fig. 4. Stiffness comparisons between the test and the FE simulation

The critical areas are located at the corners of both ends of the components, which are the same as the locations shown by the test. The effective stress value calculated from equation (1) is 3,86 MPa. The fatigue cracks are expected to appear around these areas at 20000 cycles based on the S–N curve shown in Fig. 2.

4.2 Metal-rubber springs

These springs the rubber is loaded in combined shear and compression. When used as primary springs, they are fitted, singly or in pairs, between the bogie frame and axle box (Fig. 5).

A fatigue test was carried out on one spring of this type. Several cracks were observed on the top rubber surface of the inner layer, after 0,5 million cycles under a specified fatigue load. The effective stress σ_f obtained from the finite element model is 1,2 MPa at the same location. From an S–N curve [3, 4] the fatigue crack will appear at about 0,3 million cycles. The direction of crack initiation can be determined by means of equations (4) to (7). The arrows indicate the direction of the effective stress σ_f. The observation from the actual cracks has verified this prediction.

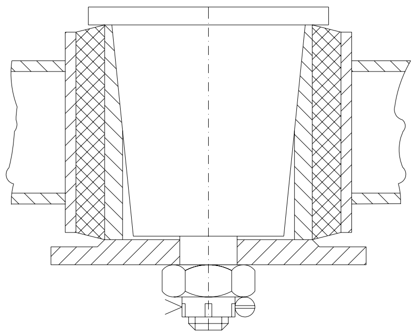

Fig. 5. Metal-rubber springs.

5 CONCLUSION

The method presented here offers a solution for industrial applications, especially for rubber components used in rail vehicle suspensions. The effective stress concept is taken as a function of all three principal stress ranges, which may enhance the reliability of fatigue design when a complex loading environment is involved. This initial verification has given some encouragement. However, further development and verification work is needed for more cases and for predicting fatigue failure inside the rubber section, which is not easy to observe.

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