Stress evaluation of conical rubber spring system

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Abstract

The conical rubber springs manufactured by Trelleborg-Industrial AVS is a primary suspension layout used between the bogie frame and the axle box for railway vehicles. The system is subjected to both static and fatigue loads. Both FE analysis and experiments are required to assess the strength and durability of the metal parts of the system. Also the stress profile will indicate where the strain gauges should be located when actual experiments are conducted. The geometry of the component is imported from CAD system to CAE system. Care has been taken to provide a sufficient fine grid to give reasonable results when mesh of the part is generated. Static and fatigue load cases are evaluated separately. Von Mises criterion has been used to evaluate the stress state under the loading conditions. The critical locations of the metal parts were identified and suggestions made for the experiments. The prototype parts have successfully passed the fatigue tests and the components are ready for batch manufacture.

1 Introduction

Major areas of application of rubber springs are in railways, marine and other industries. Rubber springs are able to withstand large strains, and they can store more elastic energy per unit volume than metals. Also they possess some inherent damping, which is particularly beneficial when resonant vibrations are encountered. The conical rubber springs manufactured by Trelleborg-Industrial AVS are used as primary suspension layout between the bogie frame and the axle box for railway vehicles. The springs are easily fitted in the engineering applications and the rubber is loaded in combined shear and compression. The whole system is subjected to both static and dynamic loads. FE analysis and experiments are required to assess the strength and durability of the metal parts
of the system, especially FE results would indicate where the strain gauge rosettes should be located. Figure 1 shows the conical rubber spring system. It consists of inner metal, rubber section and outer metal. The loads applied are also shown in Figure 1.

![Figure 1](image_url)  
Figure 1  The conical rubber spring system

2 Constitutive models for rubber

There are several hyperelastic material models which are commonly used to describe rubber and other elastomeric materials based on strain energy potential. The mechanical behaviour of these materials is described in a form of an elastic, isotropic and approximately incompressible model. The governing constitutive equations are derived from strain energy potential. Strain energy potential can be expressed as the following polynomial series

\[
U = \sum_{i+j=1}^{N} C_{ij} (\tilde{I}_1 - 3)(\tilde{I}_2 - 3)^j + \sum_{i=1}^{N} \frac{1}{D_i} (J_{el} - 3)^{2i}
\]  
\(1\)

Where \(C_{ij}\) and \(D_i\) are temperature dependent material parameters, \(J_{el}\) is the elastic volume strain, \(\tilde{I}_1\) and \(\tilde{I}_2\) are strain invariants. If \(N=1\), the polynomial formulation represents the Mooney-Rivlin hyperelasticity model. The energy potential is as follows:

\[
U = C_{10} (\tilde{I}_1 - 3) + C_{01} (\tilde{I}_2 - 3) + \frac{1}{D_1}
\]  
\(2\)

The Mooney-Rivlin form (2) was used to model the rubber section of the spring components. The material constants in equation (2) were determined from experiments.
3 Finite Element Model

It is only necessary to model half of the rubber spring due to the symmetry of the structure and the loads applied. The geometry of the component is imported from a CAD system to a CAE system. The mesh is generated using MSC-Patran and finite element analysis is performed using ABAQUS. The metal part is modelled using 8-node linear brick elements and the rubber layer using 8-node linear brick elements with hybrid and constant pressure. The basis of the hybrid elements is that the purely hydrostatic pressure can be treated as an independent variable. Otherwise the solution can not be obtained based on the displacement history only, since a very small change in displacement generates large changes in pressure. Therefore the hybrid formulation allows the pressure stress as an independently interpolated basic solution variable, coupled to the displacement solution through the constitutive theory and the compatibility condition and implemented by a Lagrange multiplier. There are two models generated to test converge of the FE results. The results between the two models are less than 3 percent. The later refined mesh is shown in Figure 2. The number of total elements is 23,232 and the total degrees of freedom are 91,113.

![Finite element model](image)

Figure 2 The finite element model

4 Static stress analysis

Two static load cases are investigated. The results give the indication for both metal strength and locations of possible failure areas.

4.1 Axial load case

A typical static load case presented here is a bolt load 50 kN and an axial load 70 kN. Figure 3 shows the deformation of the spring under the loads. It is clear that the rubber section undergoes large change but it still keeps the reasonable shape. Stress distribution of the inner metal under bolt and axial load is shown in Figure 4. The maximum value of Von Mises stress is 284 MPa which occurs around the
neck of the metal. However this value is below the 0.2 proof stress value 310 MPa. This highest stress is mainly caused by the bolt load. The stress fluctuation at this location can be ignored as far as fatigue load concerned. Figure 5 shows the stress distribution of the outer metal under axial load. The Von Mises stress value is 60.8 MPa and appears at middle section where the geometry changes. Therefore for the axial fatigue loads strain gauge rosettes can be located at this area. For the inner metal the clear stress bands are hot spots to locate strain gauge rosettes.

Figure 3 Deformation under axial load

Figure 4 Von Mises stress distribution of the inner metal under bolt load + axial load
Figure 5 Von Mises stress distribution of the outer metal under bolt load + axial load.

4.2 Combined load case

Further radial load 39 kN are applied to the conical spring together with previous bolt load and axial loads. Figure 6 shows stress distribution of the inner metal under combined loads. Again the critical area is around the neck of the metal, and the value is 294 MPa < 0.2 proof stress value 310 MPa. Stress distribution of the outer metal under combined loads is shown in Figure 7. The highest value is 87.4 MPa < 310 MPa which occurs at the same area as that loaded in axial direction.

From above stress analysis the conclusion is:
A. The strength of the conical rubber spring system meets the design requirement.
B. The critical locations for the inner metal are the neck area. This stress concentration may be reduced if a larger radius is applied.
C. For the outer metal the higher stress area is around the middle section where the geometry change occurs.
Fatigue stress analysis

The rubber spring is subjected to complicated dynamic loading environment. There are several load cases to be considered in the component design. A detailed stress calculation and comparison reported here is for axial dynamic loads. The loading consists of a bolt load 50kN, an axial static load 58 kN and an axial dynamic load ± 22.5 kN. Figure 8 shows Von Mises stress profile of the outer metal under the bolt load and the lower axial load 37.5 kN. The maximum stress value is 30 MPa. Von Mises stress profile of the outer metal under the bolt load and an higher axial load 80.5 kN is shown in Figure 9. The maximum stress value is 70.4 MPa. The hot spots are located in the middle section that is the same area under axial static load. The stress range distribution can be found in Figure 10 where the maximum stress value is 40.4 MPa against material fatigue limit 70 MPa. Therefore the component meets the fatigue requirement under the axial load condition. From design and analysis point of view Figure 10 is much more important than Figure 8 and 9 since it shows the actual alternate stress change, which is the fatigue stress values to be assessed. Figure 11 shows Von Mises stress change in inner metal. The maximum value is 12.6 MPa, which is far below the fatigue limit 70 MPa.
Figure 9 Von Mises stress distribution of the outer metal under bolt load + axial load 80.5 kN

Figure 10 Von Mises stress range distribution of the outer metal under bolt load ± axial load 22.5 kN

Figure 11 Von Mises stress range distribution of the inner metal under bolt load ± axial load 22.5 kN
6 Comparison of stress calculation and experiment results

Strain gauge rosettes were placed in several locations in both inner and outer metals. Two results from strain gauge I1 for inner metal and O1 for outer metal are shown in Table 1. The comparison result for O1 is remarkable, which is about 10% difference. But the comparison for I1 is not desirable, which may be caused by higher stress gradient change around that area. Figure 12 shows the stress range profile at the strain gauge I1. The stress range changes from 2.2 MPa to 4.2 MPa in 5.5 mm. The effective areas of strain gauge sensor are 10×11.5 mm². Therefore it is understandable that the strain gauge gives an average measurement of stress over the covered area, which would be some different from the FE result.

<table>
<thead>
<tr>
<th>Table 1 Comparison between test results and FE calculations</th>
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<tbody>
<tr>
<td>Loading: 50 kN bolt load + 58 kN axial load ± axial load 22.5 kN</td>
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<tr>
<td>Rosette Reference</td>
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<tr>
<td>-------------------</td>
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<tr>
<td>I1</td>
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<tr>
<td>O1</td>
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Figure 12 Stress distribution around the strain gauge rosette I1 in inner metal

7 Conclusions

The FE calculation, coupled with the strength test including strain gauge rosettes, have shown the conical rubber spring meets the design requirement and the component is ready for a fatigue test. Recently a fatigue test was conducted by the customer based on the design and practice code. The test has demonstrated
that the conical rubber spring system displayed adequate strength and durability. Therefore the parts are ready for production manufacture.

References