Fatigue reliability analysis of mechanical components for airflow control in pneumatic solenoid valve

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Abstract. Pneumatic solenoid valves have been broadly used in vehicles to perform automatic control task. For safety reason, they have to be highly reliable and must have long fatigue lives. The failure of the mechanical components including a conical spring and rubber seal can induce airflow disorder, which renders the unreliability of the pneumatic solenoid valve. A method is presented to study the fatigue reliability of the mechanical components used for airflow control in a pneumatic solenoid valve. The statistics of the parameters used to derive the probability distributions of the stresses/strains in the mechanical components are obtained from the measurements of the parameter properties and static load testing of the components. The existing material S-N curves and stress/strain probability distributions are used to determine the reliabilities of the mechanical components. Based on the series system criterion, the fatigue reliability of the pneumatic solenoid valve with the consideration of the airflow disorder failure mode is computed for the life of 1 million cycles.

1. Introduction
A pneumatic solenoid valve is a device used to control the flow of gas in a mechatronic system. In general, a pneumatic solenoid valve is powered by the electromagnetic energy in the coil of a solenoid which converts electrical current into magnetic force. Pneumatic solenoid valves may have different configurations. As an example, the 3-port-2-way pneumatic solenoid valve considered herein belongs to a two-way style which has 3 ports, namely inlet, outlet, and exhaust air ports, as shown in Fig. 1. In the close state, the conical spring is pre-compressed to press the rubber seal against the orifice of the inlet port. On the other hand, in the open state, an additional compressive force generated by the solenoid is applied to the conical spring to remove the rubber seal away from the orifice. During operation, both the conical spring and rubber seal will be subjected to intermittent impulsive dynamic loads. Under such dynamic loads, the conical spring and rubber seal will experience fatigue which may lead to the failure of these components. Since pneumatic solenoid valves have been broadly used in vehicles to perform automatic control task, many research works have been devoted to study different aspects such as design, testing, applications, etc. of solenoid valves [1-3]. It has been pointed out that a pneumatic solenoid valve may have different failure modes during its lifetime [4-6]. One of the major failure modes of the solenoid valve is that it does not close the inlet port to stop airflow. In general, this failure mode is mainly due to the fatigue failure of any of the mechanical components such as conical spring and rubber seal used for airflow control. Therefore, for safe and reliable operation of a solenoid valve during its lifetime, the fatigue failure modes of the critical mechanical components must be mitigated and the uncertain parameters that can affect the fatigue reliability of the critical components identified and properly controlled. In this study, the fatigue stresses developed in the conical spring and rubber seal are determined in the finite element
analyses (FEA) of the components. The fatigue stresses are then used to estimate the fatigue reliability of the mechanical components for the life of $10^6$ cycles.

2. Stress Analysis of Mechanical Component

The stresses developed in the conical spring as shown in Fig. 2 are determined in the FEA. The dimensions of the spring are shown in Fig. 3. The finite element model of the spring is shown in Fig. 4 in which the dimensional triangular SOLID186 elements of the commercial finite element code ANSYS [7] are used to model the spring. The spring Young’s modulus is 198GPa. The spring may behave non-linearly due to the contact between neighboring coils when under higher loads. Therefore, the contact elements TARGE170 and CONTA174 are also included in the FEA. The experimental load-displacement curve has validated the correctness of the load-displacement curve obtained from the FEM as shown in Fig. 5. The maximum shear stress occurs at the bottom of the spring for different loading conditions as shown in Fig. 6. The maximum tensile stresses at the close and open states for four springs are determined in the FEA and listed in Table 1.

![Figure 1. Pneumatic solenoid valve with two ports](image1)

![Figure 2. Conical spring](image2)

![Figure 3. Spring dimensions](image3)

![Figure 4. FE model of spring](image4)
The stresses/strains in the rubber seal are also determined in the FEA with the consideration of the contact behavior between the seal and the orifice ring. For the rubber seal, the Young’s modulus and Poisson ratio are 4.45MPa and 0.47, respectively; for the steel orifice, the Young’s modulus and Poisson’s ratio are 210GPa and 0.3, respectively. The diameter and depth of the seal are 4 and 2mm, respectively. The thickness of the hollow indenter wall is approximately 0.31mm. The FE model of the seal and orifice ring is shown in Fig. 7 in which the SOLID186 elements with 20 nodes are used to model the assembly. The accuracy of the FE model has been verified by the experimental indentation. For an applied load of 3.628N, the error between the experimental (0.072mm) and theoretical (0.074762 mm) indentations is less than 4%. The distribution of vertical normal strain in the vertical direction is shown in Fig. 8 which shows that the maximum vertical normal strain occurs at the bottom right corner of the indentation. The maximum von Mises stress is 0.6316 MPa.

**Table 1.** Statistics of spring parameter and tensile stress

| Specimen No. | Parameter       | Wire diameter d (mm) | Large circle inner diameter Dmax (mm) | Small circle inner diameter Dmin (mm) | Height h (mm) | Close state Tensile stress (N) | Open state Tensile stress (N) |
|--------------|----------------|----------------------|--------------------------------------|--------------------------------------|---------------|-------------------------------|-------------------------------|
| 1            | 0.95           | 10.12                | 6.40                                 | 13.830                               | 9.898         | 11.858                        |
| 2            | 0.95           | 10.12                | 6.37                                 | 14.115                               | 10.682        | 12.740                        |
| 3            | 0.95           | 10.15                | 6.49                                 | 14.355                               | 10.976        | 13.132                        |
| 4            | 0.95           | 10.14                | 6.48                                 | 13.930                               | 11.466        | 13.720                        |
| Mean         | 0.95           | 10.133               | 6.435                                | 14.058                               | 10.756        | 12.863                        |
| Variance     | 0              | 1.688×10^{-4}        | 0.003                                | 0.040                                | 0.324         | 0.458                         |
| Coefficient of | 0              | 0.001                | 0.008                                | 0.014                                | 0.053         | 0.053                         |

The stresses/strains in the rubber seal are also determined in the FEA with the consideration of the contact behavior between the seal and the orifice ring. For the rubber seal, the Young’s modulus and Poisson ratio are 4.45MPa and 0.47, respectively; for the steel orifice, the Young’s modulus and Poisson’s ratio are 210GPa and 0.3, respectively. The diameter and depth of the seal are 4 and 2mm, respectively. The thickness of the hollow indenter wall is approximately 0.31mm. The FE model of the seal and orifice ring is shown in Fig. 7 in which the SOLID186 elements with 20 nodes are used to model the assembly. The accuracy of the FE model has been verified by the experimental indentation. For an applied load of 3.628N, the error between the experimental (0.072mm) and theoretical (0.074762 mm) indentations is less than 4%. The distribution of vertical normal strain in the vertical direction is shown in Fig. 8 which shows that the maximum vertical normal strain occurs at the bottom right corner of the indentation. The maximum von Mises stress is 0.6316 MPa.
3. Material S-N curve

The fatigue data of the component materials obtained in room temperature have been reported in the literature [8, 9]. The S-N curve for relating the alternating stress and fatigue cycles is expressed as

\[ NS^m = C \]  

(1)

where \( N \) is fatigue cycles; \( S \) is alternating stress; \( m, C \) are constants. The above relation can be linearized in the log-log paper.

\[ \log C = m \log S + \log N \]  

(2)

For the fatigue reliability analyses of the components, the fatigue data are used to construct the S-N curves of the materials. The linearized experimental S-N curves of the conical spring and rubber seal derived from the fatigue data are shown in Figs. 9 and 10, respectively. Regarding the S-N curve of the rubber seal, \( W \) is the strain energy induced by uniaxial strain \( \varepsilon \) with \( W = E\varepsilon^2/2 \) where \( E \) is Young’s modulus.
4. Fatigue Reliability of Solenoid Valve Components

The alternating stress and normal strain at the critical locations in the conical spring and rubber seal, respectively, will become uncertain when the randomness caused by fabrication exists in the component dimensions and material constants. Such uncertainties may lead to unexpected failures of the mechanical components within the design lifetime of the solenoid valve. Therefore, the fatigue reliability of the mechanical components should be assessed to ensure that the fatigue failure probabilities of such components are negligible. In constructing the probability distribution of the alternating stress in the conical spring, a set of springs is subjected to static load testing to find the loads associated with the required spring shortenings at the open and close states, respectively. The loads are then used in the FEA to find the mean stress and alternating stresses of the springs. According to Goodman fatigue theory, the relation between mean and alternating stresses is

\[
\frac{\sigma_a}{\bar{\sigma}_a} + \frac{\sigma_m}{S_u} = 1
\]  

(3)

where \(\bar{\sigma}_a\) is the transformed alternating stress with zero mean stress, \(\sigma_a\) is the initial alternating stress, and \(S_u\) is ultimate stress. The alternating stresses are converted to those with zero mean stress via the following Goodman equation.

\[
\sigma'_a = \frac{\sigma_a}{\sigma_m - S_u} (-S_u)
\]  

(4)

The probability distribution of the alternating stress is assumed to be Gaussian as shown in Fig. 11 with mean = 131.97MPa and standard deviation = 7.35MPa. From the S-N curve in Fig. 9, the alternating stress corresponding to the life of one million cycles is 340.70MPa. The fatigue life is less
than 106 cycles when the transformed alternating stress is larger than 340.70MPa. Therefore, the failure probability of the conical spring can be expressed as $P_{fs} = P[\text{alternating stress} > 340.70\text{MPa}]$, ie,

$$P_{fs}(x) = \int_{\sigma_{\text{a}}}^{\infty} f(x)dx$$

(5)

The above integration can be performed using the mathematical code Mathematica which produces $P_{fs} = 1.6095 \times 10^{-177}$. Regarding the fatigue reliability analysis of the rubber seal, a set of rubber seals is first subjected to indentation testing under the load of 3.63N to find the possible indentations of the rubber seals. It is assumed that the Young’s modulus of the seal is a random variable. The applied load together with the indentations is then used in the FEA of the rubber seals to find the statistics of the Young’s modulus and maximum normal strain of the seal via an iterative approach. The probability distribution of the maximum normal strain which is again treated as a normal as shown in Fig. 12 with mean = 0.0671 and standard deviation = 0.004. From the S-N curve in Fig. 10, the normal strain for the life of one million cycles is 0.2288. The failure probability of the rubber seal is expressed as $P_{fr} = P[\text{normal strain} > 0.2288]$ which is equal to $2.2335 \times 10^{-211}$. Based on the series system theory, the fatigue reliability of the solenoid is defined as $P_{s} = [1 - P_{fs}][1 - P_{fr}]$ which is close to 1. The high fatigue reliability of the solenoid indicates that the mechanical components have been properly designed to prevent fatigue from occurring.

5. Conclusion
A method for evaluating the fatigue reliability of a pneumatic solenoid valve with the consideration of airflow disorder as the major failure mode has been presented. The mechanical components related to this failure mode have been identified as the conical compressive spring and rubber seal. The
probability distributions of the maximum alternating stress and normal strain in the conical spring and rubber seal, respectively, have been constructed using the results obtained from experiments and the FEA of the components. The S-N curves and the probability distributions of the components have been used to determine the component failure probabilities. The fatigue reliability of the solenoid valve has been estimated from the component failure probabilities based on the series system model. It has been shown that fatigue failure probability of the solenoid valve is negligible. The high fatigue reliability of the solenoid valve implies that the designs of the mechanical components for airflow control are acceptable.

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