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Design and failure modes of automotive suspension springs

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Abstract

This paper is a discussion about automotive suspension coil springs, their fundamental stress distribution, materials characteristic, manufacturing and common failures. An in depth discussion on the parameters influencing the quality of coil springs is also presented.

Following the trend of the auto industry to continuously achieve weight reduction, coil springs are not exempt. A consequence of the weight reduction effort is the need to employ spring materials with significantly larger stresses compared to similar designs decades ago. Utilizing a higher strength of steel possesses both advantages and disadvantages. The advantages include the freedom to design coil springs at higher levels of stress and more complex stresses. Disadvantages of employing materials with higher levels of stress come from the stresses themselves. A coil's failure to perform its function properly can be more catastrophic than if the coil springs are used in lower stress. As the stress level is increased, material and manufacturing quality becomes more critical. Material cleanliness that was not a major issue decades ago now becomes significant. Decarburization that was not a major issue in the past now becomes essential.

To assure that a coil spring serves its design, failure analysis of broken coil springs is valuable both for the short and long term agenda of car manufacturer and parts suppliers. This paper discusses several case studies of suspension spring failures. The failures presented range from the very basic including insufficient load carrying capacity, raw material defects such as excessive inclusion levels, and manufacturing defects such as delayed quench cracking, to failures due to complex stress usage and chemically induced failure. FEA of stress distributions around typical failure initiation sites are also presented.

Keywords: Metallurgical failure analysis; Decarburization; Inclusion; Delayed quench crack; Coil spring

1. Introduction

A mechanical spring is defined as an elastic body which has the primary function to deflect or distort under load, and to return to its original shape when the load is removed. The long-established compression spring design theory involves over simplification of the stress distribution inside the wire. One of the simplest

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approaches available is referenced here [1]. The so called un-wound spring as shown in Fig. 1 is commonly used. It is based on the assumption that an element of an axially loaded helical spring behaves essentially as a straight bar in pure torsion. The following notations are typically used: P: Applied load, α: Pitch angle, τ: Shear stress, R: Coil radius, and d: Wire diameter. The torsion is then calculated as $PR\cos\alpha$, the bending moment as $PR\sin\alpha$. the shear force as $P\cos\alpha$, and the compression force as $P\sin\alpha$. Traditionally, when the pitch angle is less than 10°, both the bending stresses and the compression stresses are neglected.

Assuming that the shear stress distribution is linear across the wire cross section, and $PR\cos\alpha = PR$, the following should be valid:

$$\tau = \frac{16PR}{\pi \cdot d^3}. \tag{1}$$

The shear stress here is usually called uncorrected shear stress. The total length l is $2\pi Rn$, where n is the number of active coils. Using the fact that $\gamma = \tau/G$, it can be rewritten as $16PR/(\pi \cdot d^3 G)$, and the total angular torsion φ becomes:

$$\varphi = \int_0^{2\pi Rn} \frac{2\gamma}{d}\mathrm{d}x = \frac{32PR}{\pi d^4 G}\mathrm{d}x = \frac{64PR^2 n}{Gd^4}, \tag{2}$$

where G is the modulus of rigidity. The total deflection caused by the angular torsion is:

$$\delta = R\varphi = \frac{64PR^3 n}{Gd^4} = \frac{8PD^3 n}{Gd^4}. \tag{3}$$

The spring rate therefore becomes:

$$k = \frac{P}{\delta} = \frac{Gd^4}{8nD^3}. \tag{4}$$

Eq. (4) is still commonly used to estimate the spring rate by suspension designers. As opposed to the uncorrected shear stress in Eq. (1), Wahl [2] proposed corrected shear stress. The uncorrected shear stress neglects a great many factors which modify the stress distribution in actual helical springs. The corrected shear stress, τ_a, is obtained by multiplying the uncorrected stress with a correction factor K, which depends upon the spring index D/d. Fig. 2 shows the typical corrected shear stress distribution.

Furthermore, by taking x as the distance from the cross point where the shear stress is zero, Wahl proved that the following equation holds:

$$\tau_a = \frac{32xPR^2}{\pi \cdot d^4 (R - d^2/16R - x)} \tag{5}$$

With the introduction of the spring index $c = D/d$, the maximum shear stress at the inner side of the coil, where $x = d/2 - d^2/16R$, becomes:

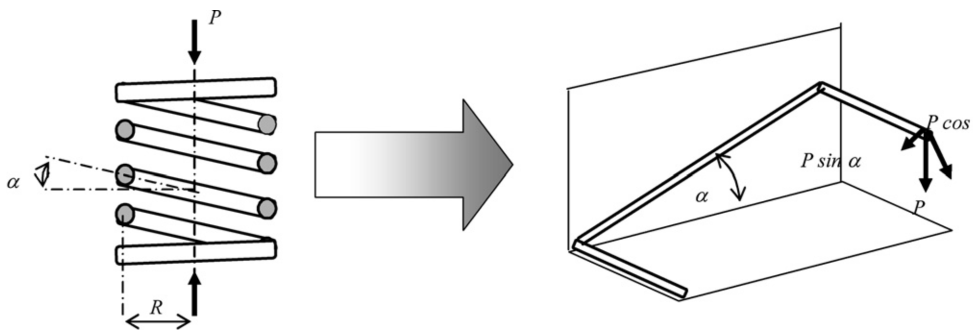

Fig. 1. Wound and un-wound coil springs.