MODELING, SIMULATION AND EXPERIMENTAL VERIFICATION OF VARIABLE PARAMETER OF SUSPENSION SPRING

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ABSTRACT

Helical spring plays an important role in many applications such as machines, and vehicles. When a helical spring is under a static load, it is important to know their static displacement characteristic. The common static model for helical spring is the Wahl's factor equation, which assumes all design parameters such as spring radius, wire diameter, helix radius and pitch angle are constant. In this research, several mathematical models are used to the static displacement characteristic of automotive suspension helical spring with design parameters. These design parameters include pitch angle, wire diameter and helix radius. Finite element analysis and experimental simulation by Autodesk Simulation Multiphysics (Algor) were performed to verify and support the accuracy of the formulated model with the actual test of the helical spring results.

ABSTRAK

Spring heliks memainkan peranan penting dalam banyak aplikasi seperti mesin, dan kenderaan. Apabila spring heliks dikenakan beban yang statik, perkara penting yang perlu dikenalpasti adalah untuk mengetahui ciri-ciri statik spring tersebut. Persamaan yang biasa digunakan untuk model statik ini adalah persamaan faktor Wahl's, yang menganggap semua parameter seperti diameter luar, diameter gegelung, jarak/sudut antara gegelung adalah sama. Dalam kajian ini, beberapa persamaan matematik digunakan untuk spring statik yang mempunyai parameter yang pelbagai. Parameter reka bentuk termasuk diameter luar, diameter gegelung, jarak/sudut antara gegelung. "Finite Element Analysis" dan simulasi eksperimen oleh Autodesk Simulasion Multiphysics (Algor) telah dijalankan untuk mengesahkan dan menyokong ketepatan model yang dirumuskan dengan ujian sebenar keputusan spring heliks.

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LIST OF SYMBOLS

d	Wire diameter
Di	Internal diameter
De	External diameter
Ls	Solid length
Lf	Free length
α	Pitch angle
Ds	Spring diameter
c	Spring index
R	Coil radius
P	Applied load
φ	Torsion
G	Modulus of rigidity
δ	Deflection
k	Spring rate
τ	Shear stress
τ_{max}	Maximum shear stress
τ_{min}	Minimum shear stress
K_w	Wahl's factor

LIST OF ABBREVIATIONS

FEA	Finite Element Analysis
FEM	Finite Element Method
CATIA	Computer Aided Three-dimensional Iterative Application
MES	Mechanical Event Simulation

CHAPTER 1

INTRODUCTION

1.1 Project Background

Suspension system is one of the most important and basic systems in a vehicle. The major purpose of any vehicle suspension system is to maximize the friction between the road surface and the tires to provide the stability steering and good handling of the vehicle. To achieve the stability and rides comfort, there were three important principles must be resolved which is road isolation, road handling and cornering. Numerous studies have been conducted in other to achieve stability and rides comfort.

Vehicle suspension system consists of 3 elements which are wishbones, spring and the shock absorber [1]. These 3 elements are to filter and transmit forces exerted between the vehicle body and the road. The spring is important as it carries the body mass and isolates the vehicle form uneven road surface. This contributes to drive comfort. Furthermore, damper system also contributes to safety as it absorbs the damping of the body and wheel oscillations.

1.2 Problem Statement

Most of the spring fail is due to the fatigue, in other word, they have sustained much compression-extension cycle, which causes the metal to become brittle and then breaks. If the amplitude of these cycles is large, the fatiguing process is accelerated [2]. Springs tend to be highly stressed because they are designed to fit into small spaces with the least possible weight and lowest material cost. At the same time they are required to deliver the required force over a long period of time. The reliability of a spring is therefore related to its material strength, design characteristics, and the operating environment [3]. The same goes for the car suspension spring whereby after some period of time, the car spring will have irregular and unstable stiffness. To overcome this problem, this research investigates the different parameters of an automotive suspension spring that affect the static characteristic (displacement)

1.3 Project Objectives

- To investigate a static mathematical model for helical spring with variable design parameters.
- To create a finite element analysis (FEA) model of the spring and simulate the model using variable design parameters.
- Fabricate and setup an experimental apparatus to collect the spring static displacement-force data, and to verify the actual experimental test with simulation results.

1.4 Project Scope

This project is focused on study the different parameter of the spring that will effect on the static characteristic of the linear spring. The purpose of the project is to compare the result from simulation software (Autodesk) with the actual result from the spring tester machine. The FEM analysis is perform in Autodesk software to define the characteristic of the spring.

1.5 Organization of Report

This thesis consists of five chapters. Chapter 2 presents the literature review while Chapter 3 discusses the methodology for the investigation. Chapter 4 presents the results from the experiment and the discussions regarding the results. Finally, Chapter 5 summarizes the study and provides recommendations for the study.

CHAPTER 2

LITERATURE REVIEW

2.1 Background of Spring

According to Wahl, [4]: “A mechanical spring may be defined as an elastic body whose primary function is to deflect or distort under load (or to absorb energy) and which recovers its original shape when released after being distorted”. Then he goes on detail to the functions of the spring where normally people do not thing of this general definition of the spring which is: to support a body or structure, to apply force, to absorb shock, or to provide load control. With these definitions of the spring, the aircraft wings, the body chassis of the car and even the shoes that we wear also will be considered as a spring. The same concept happen to this entire item which is all will depress under load and revert back to its original shape after load is released. A shoe sole will absorb the impress of the foot fall and the bending of arc of the foot and return to its normal state when the foot is removed. The aircraft wings must take the loading and unloading on take-off and landing of the plane in other to encounters the air turbulence.

2.2 Spring Geometry

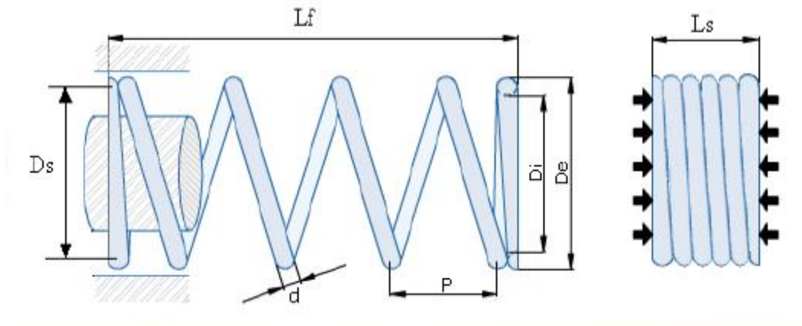

Figure 2.1: Spring geometry [5].

- $\boldsymbol{d}$ (wire diameter): This parameter describes the diameter of wire used as material for spring.
- $\boldsymbol{Di}$ (internal diameter): Internal diameter of a spring can be calculated by subtracting the doubled wire diameter from the external diameter of a spring.
- $\boldsymbol{De}$ (external diameter): External diameter of a spring can be calculated by adding the doubled wire diameter to the internal diameter of a spring.
- $\boldsymbol{Ls}$ (Solid length): Maximal length of a spring after total blocking. This parameter is shown in the picture on right.
- $\boldsymbol{Lf}$ (free length): Free length of compression springs is measured in its uncompressed state.
- $\boldsymbol{P}$ (pitch): Average distance between two subsequent active coils of a spring.
- $\boldsymbol{Ds}$ (Spring diameter): Spring diameter is mean diameter of spring. That is calculated by subtracting wire diameter d from external diameter $\boldsymbol{De}$ [5].

2.3 Shear Stress of Spring

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 [6]:

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

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 $\gamma = {}^{\gamma}/_{G}$, it can be rewritten as $16PR/(\pi \cdot d^3 G)$, and the total angular Torsion φ becomes [6]:

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

where G is the modulus of rigidity. The total deflection caused by the angular torsion is [6]:

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

The spring rate therefore becomes [6]:

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

Eq. (4) [6] is still commonly used to estimate the spring rate by suspension designers. As opposed to the uncorrected shear stress in Eq. (1), Wahl [4] 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. 2.2 shows the typical corrected shear stress distribution.

Furthermore, by taking x as the distance from the cross point where the shears 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:

$$\tau_{a1} = \frac{16PR}{\pi \cdot d^3}\frac{4c - 1}{4c - 4} \tag{6}$$

Additional shear stress caused by the neutral surface of a cantilever of circular cross section loaded by force P, the term $4.92P/pd^2$ should be added to obtain maximum shear stress:

$$\tau_{max} = \frac{16PR}{\pi \cdot d^3}\left[\frac{4c - 1}{4c - 4} + \frac{0.615}{c}\right] \tag{7}$$

And minimum shear stress:

$$\tau_{min} = \frac{16PR}{\pi \cdot d^3}\left[\frac{4c + 1}{4c + 4} - \frac{0.615}{c}\right] \tag{8}$$

Equations (7) and (8) are usually used by the design engineer for coil springs when neglecting the curvature. Also, since the equations were derived by over simplification, the larger the Pitch angle, the more error that will result. In reality, coil spring makers today use equations that are generally confidential, and therefore will not be discussed here. The equations require the design engineer to input the coil diameter, design height, design load, spring rate, etc. The equation will calculate the optimum possible shape and dimension of the coil. After this step, for more accurate stress distribution, it is usually too cumbersome not to use FEM to design.

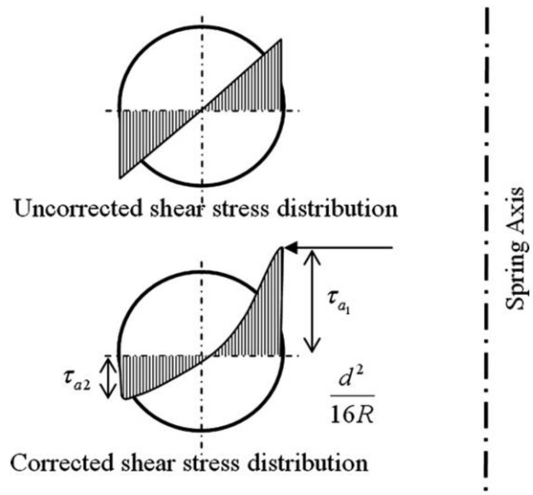

Figure 2.2: Uncorrected shear stress vs. corrected shear stress distribution [6].

2.4 Design Parameter of Coil Spring

The development of metal spring has continued during the past years and the focused is on reducing the operating weight of spring. The specific stresses on spring are continuously increasing which leads to smaller mounting space. Thus, the most care is to be taken for careful manufacturing of spring, focused on the surface layer, hot pre-setting and shot peening. The surface quality plays important part for operational durability of springs than material properties.

Following are the different design parameters which affect the design of mechanical coil spring.

2.4.1 Operating Mode

Spring is design to adapt many situations such as compression, extension, torsion, power and constant force. Depending on its function, a spring may be in static, cyclic or dynamic operating mode. The static condition for any spring is considered if change are happen in deflection or load occur only a few time, for example 10,000 cycle during expected life of cycle during the life of spring [3]. This condition is remaining loaded for a very long of time for a static spring. Spring relaxation, set and creep are the failure mode interest for this static spring.

Cyclic are spring expected to have higher failure rate due to fatigue compare to the constant spring as it is flexed repeatedly. Cyclic spring also can be operated in unidirectional mode or a reversed stress mode. In some cases, the stress is always applied in the same direction, while for the others, stress is applied first in one direction then in the opposite direction [3]. For the same maximum stress and deflection between a unidirectional and reversed stress spring, the shorter fatigue life would be expected since the stress range for the reversed stress spring is twice that of unidirectional spring.

Dynamic loading is referring to intermittent occurrences of a load surge such as a shock absorber inducing higher than normal stresses on the spring. Dynamic loading divided into three main categories: shock, resonance of the spring itself, and resonance of the spring /mass system. Skewis [3] asserted that shock loading is occur when the load is applied with sufficient speed as the first coil of the spring take up more of the load than would be calculated in static and cyclic spring situation. This loading condition occurs due to the inertia of the spring coil. When the operating speed is the same as the natural frequency of the spring or a harmonic of the natural frequency, the spring resonance is appears. Resonance can effect on greatly elevated stresses and possible coil clash resulting in premature failure.

2.4.2 Imperfection on Inside Diameter of Spring

Helical compression springs respond to external compressive force with torsional stress caused by torsion of the active spring coil which, in a first approximation, may be estimated analogous to a straight torsion bar. Since the shear angle is greater on the inner surface than on outer surface, the peripheral torsional stress on the inner coil surface is higher than on the outer surface. This circumstance is described by using a correction factor of k which is dependent on the curvature of the wire. The curvature can be characterized by the quotient from the mean spring diameter and the wire diameter, the so-called coil ratio [6]. This means:

- The maximum stress of coil spring occurs on the inner coil surface.
- Accordingly, fatigue fractures of coil springs generally originate from this area.
- Therefore, the spring coil’s inner surface has to be shot peened with particular care, which depending on the spring geometry, constitutes a highly fastidious task [6].

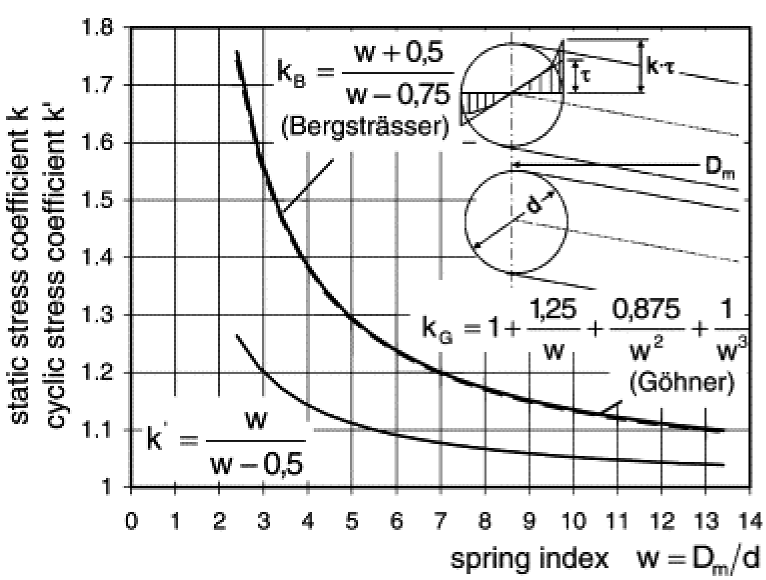

Figure 2.3: Coil ratio vs. stress concentration [5].

Fig. 2.3 shows the correction factor of k to describe the static stress concentration on the inner coil surface of coil spring in dependence on the coil ratio w, factor k_0 represent the effect of the stress concentration in the case of cyclic load.

2.4.3 Stress Peening

Shot peening is a standard technological procedure. Peening is the interaction between a particle (with the necessary hardness) and the surface of a working piece. If the particles are in round shape, it is called shot peening. In the surface layer (up to a depth 0.5mm), compressive residual stresses are induced. At a lower hardness of the working piece, an additional hardening is achieved. In order to obtain better results through the peening process, the so-called stress peening is used.

2.4.4 Operating Temperature

Compression springs are subjected to high temperature requires special attention to spring material selection and spring design. In elevated temperature service, advanced super alloys are required to give stable spring load characteristic. Increase in temperature will affect the elastic modulus and the elastic limit of most spring materials. The decreasing elastic modulus of spring alloys as temperature is increased is shown in Fig. 2.4. This change is completely reversible.

In addition, the rate of the spring will be changed in proportion to the modulus. The change in yield strength for several spring materials is shown in Fig. 2.5. The decrease in strength is not reversible. To avoid this, the designer needs to use a design with lower stresses when the spring is to work in an elevated temperature environment.

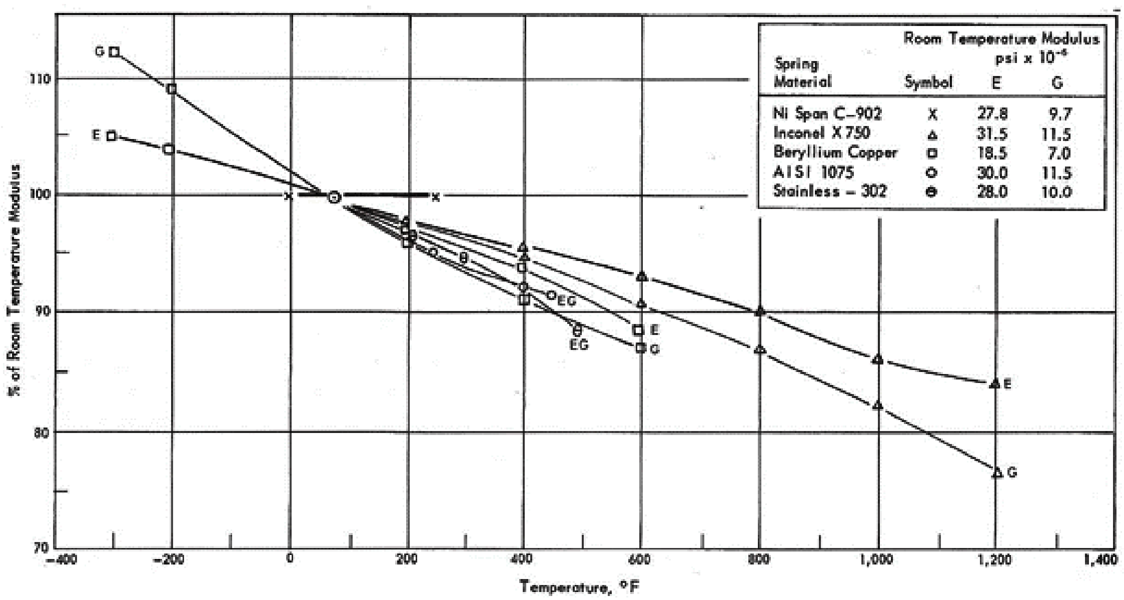

Figure 2.4: Change in modulus with temperature [5].

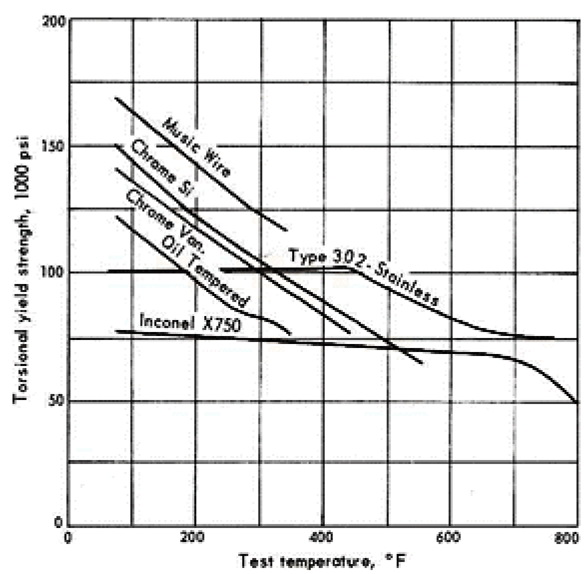

Figure 2.5: Torsional yield strength of spring wire at elevated temperature [5].

Maximum usable temperatures for spring material are simply the temperature at which metallurgical change begins. When a sustained stress is applied at an elevated temperature, the time dependent changes in spring occur. If the stress is sufficiently high, these changes will occur at room temperature. Increasing the temperature will proportionally increase the rate of change. Commonly, the change occurs as a reduction in coil spring length under load or a reduction in spring load at fixed length. This relaxation occurs gradually at first and then at a decreasing rate over time [5]. There is no apparent end point.

2.5 Failure Modes of Mechanical Spring

All type of spring are expected to undergo and operate for over long time without substantially any change in dimension, displacement or spring rate, often under changing loads. Concerning these requirements, potential failure modes include corrosion fatigue, fretting fatigue, relaxation, thermal buckling, yielding, fatigue, creep, and force-induced elastic deformation. The operating life of the mechanical spring arrangement is depending on the tendency of materials to corrosion and stress levels (static, cyclic or dynamic). Moreover, the most common failure mode for this mechanical spring is upon fatigue and excessive loss of load due to stress relaxation. By definition, the object that are loaded under purely oscillatory loads ($\sigma_{mean} = 0$) fail when their stress reach the material's fatigue limit, $\sigma_{fatigue}$. Otherwise, the load that purely static load ($\sigma_{alt} = 0$) fail when their stresses reach the material yield limit, σ_{yield}. The Soderberg Criterion provides s way to calculate a failure unit for spring that have a mixture of σ_{mean} and σ_{alt} stresses. Mean stress is plotted on one axis and alternating stress on the other. Figure 2.6 shows a typical Soderberg plot [3].