P.S.Valsange / International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622 www.ijera.com
Vol. 2, Issue 6, November- December 2012, pp.513-522

Design Of Helical Coil Compression Spring" A Review

P.S.Valsange

Abstract

This report is a review of fundamental stress distribution, characteristic of helical coil springs. An in depth discussion on the parameters influencing the quality of coil springs is also presented. Factors affecting strength of coil spring, F.E.A. approaches by the researchers for coil spring analysis are also studied. Reduction in weight is a need of automobile industry. Thus the springs are to be designed for higher stresses with small dimensions. This requires critical design of coil springs. This leads to critical material and manufacturing processes. Decarburization that was not a major issue in the past now becomes essential, to have better spring design.

Keywords: Decarburization; Inclusion; Delayed quench crack; Coil spring; Spring relaxation.

INTRODUCTION

Introduction

Helical compression springs are used widely all over the world. It has different type of applications in different areas. According to that the design considerations are to be made which are discussed in this chapter. The chapter further discusses about basic phenomenons like stability of spring, surge in spring, spring relaxation, fatigue loading, strain energy. and basic design procedure of the helical compression springs.

1.1 Concept of spring design.

The design of a new spring involves the following considerations:

• Space into which the spring must fit and operate.
• Values of working forces and deflections.
• Accuracy and reliability needed.
• Tolerances and permissible variations in specifications.
• Environmental conditions such as temperature, presence of a corrosive atmosphere.
• Cost and qualities needed.

The designers use these factors to select a material and specify suitable values for the wire size, the number of turns, the coil diameter and the free length, type of ends and the spring rate needed to satisfy working force deflection requirements. The primary design constraints are that the wire size should be commercially available and that the stress at the solid length be no longer greater than the torsional yield strength. Further functioning of the spring should be stable.

1.1.1 Stability of the spring (Buckling).

Buckling of column is a familiar phenomenon. Buckling of column is a familiar phenomenon. We have noted earlier that a slender member or column subjected to compressive loading will buckle when the load exceeds a critical value. Similarly compression coil springs will buckle when the free length of the spring is larger and the end conditions are not proper to evenly distribute the load all along the circumference of the coil. The coil compression springs will have a tendency to buckle when the deflection (for a given free length) becomes too large.

Buckling can be prevented by limiting the deflection of the spring or the free length of the spring.

The behavior can be characterized by using two dimensionless parameters, critical length and critical deflection. Critical deflection can be defined as the ratio of deflection (y) to the free length (Lf) of the spring. The critical length is the ratio of free length (Lf) to mean coil diameter (D).The critical deflection is a function of critical length and has to be below a certain limit. As could be noticed from the figure absolute stability can be ensured if the critical length can be limited below a limit.

For reducing the buckling effect following condition must be satisfied

$L_f < 4D$ ….1.1

The crippling load can be given by

$W_{cr} = K \times K_B \times L_f$ ….1.2

Where,

K = spring rate
K_B= buckling factor

1.1.2 Spring Surge and Critical Frequency.

If one end of a compression spring is held against a flat surface and the other end is disturbed, a compression wave is created that travels back and forth from one end to the other exactly like the swimming pool wave. Under certain conditions, a resonance may occur resulting in a very violent motion, with the spring actually jumping out of contact with the end plates, often resulting in damaging stresses. This is quite true if the internal damping of the spring material is quite low. This phenomenon is called spring surge or merely surging. When helical springs are used in applications requiring a rapid reciprocating motion, the designer must be certain that the physical dimensions of the spring are not such as to create a natural vibratory frequency close to the frequency of the applied force.

1.1.3 Fatigue Loading

The springs have to sustain millions of cycles of operation without failure, so it must be designed for infinite life. Helical springs are never used as both compression and extension springs. They are usually assembled with a preload so that the working load is additional. Thus, their stress-time diagram is of fluctuating nature.
Now, for design we define,

$$Fa = (Fmax - Fmin)/2$$
$$Fa = (Fmax + Fmin)/2$$

Certain applications like the valve spring of an automotive engine, the springs have to sustain millions of cycles of operation without failure, so it must be designed for infinite life. Unlike other elements like shafts, helical springs are never used as both compression and extension springs. In fact they are usually assembled with a preload so that the working load is additional. Thus, their stress-time diagram is of fluctuating nature. Now, for design we define,

Then the stress amplitude and mean stress values are given by: if we employ the Goodman criterion, then
The best data on torsional endurance limits of spring steels are those reported by Zimmerli. He discovered the surprising fact that the size, material and tensile strength have no effect on the endurance limits (infinite life only) of spring steels in sizes under 10mm(3/8 inches). For all the spring steels in table the corrected values of torsional endurance limit can be taken as: = 310 Mpa (45.0 kpsi) for unpeened springs= 465 Mpa (67.5 kpsi) for peened springs.
The stress amplitude and mean stress values are given by:

Twisting moment :-

$$\mathbf{Fs1 = Kc}\left(\frac{\mathbf{8PD}}{[\mathbf{\Pi d3}]}\right) \qquad ....1.3$$

Direct shear stress :-

$$\mathbf{Fs2} = \frac{\mathbf{W}}{[(\frac{\Pi}{4})\mathbf{d2}]} \qquad ....1.4$$

Max shear stress :-

$$F_{s=} \frac{\mathbf{8KWD}}{[\mathbf{\Pi d3}]} \qquad ....1.5$$

1.1.4 Strain energy

Traditionally the springs are made up of materials. The main factor is to be considered in design of spring is the strain energy of material used. Strain energy in materials can be expressed as.

$$U = б^2 / (\rho\times g) \qquad ....1.6$$

This indicates that the material with lower young's modulus (E) or density (ρ) will have relatively higher strain energy under same stress (б)

1.1.5 Spring Relaxation:

Springs of all types are expected to operate over long periods of time without significant changes in dimension, displacement, or spring rates, often under fluctuating loads. If a spring is deflected under full load and the stresses induced exceed the yield strength of the material, the resulting permanent deformation may prevent the spring from providing the required force or to deliver stored energy for subsequent operations. Most springs are subject to some amount of relaxation during their life span even under benign conditions. The amount of spring relaxation is a function of the spring material and the amount of time the spring is exposed to the higher stresses and/or temperatures.
Static springs can be used in constant deflection or constant load applications. A constant deflection spring is cycled through a specified deflection range, the loads on the spring causing some set or relaxation which in turn lowers the applied stress. The spring may relax with time and reduce the applied load. Elevated temperatures can cause thermal relaxation, excess changes in spring dimension or reduced load supporting capability. Under constant load conditions, the load applied to the spring does not change during operation. Constant load springs may set or creep, but the applied stress is constant. The constant stress may result in fatigue lives shorter than those found in constant deflection applications.

In many applications, compression and extension springs are subjected to elevated temperatures at high stresses which can result in relaxation or loss of load. This condition is often referred to as "set". After the operating conditions are determined, set can be predicted and allowances made in the spring design. When no set is allowed in the application, the spring manufacturer may be able to preset the spring at temperatures and stresses higher than those to be encountered in the operating environment.

A highly stressed spring will set the first several times it is pressed. Relaxation is a function of a fairly high stress (but usually lower then that required to cause set) over a period of time. Creep in the spring may lead to unacceptable dimensional changes even under static loading (set). A spring held at a certain stress will actually relax more in a given time than a spring cycled between that stress and a lower stress because it spends more time at the higher stress. The amount of spring relaxation over a certain period of time is estimated by first determining the operating temperature, the maximum amount of stress the spring sees and how long the spring will be exposed to the maximum stress and the elevated temperature over its lifetime.

1.2 Design considerations

The long-established compression spring design theory involves over simplification of the stress distribution inside the wire so called un-wound spring as shown in Fig. 1.1. 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, a: Pitch angle, s: Shear stress, R: Coil radius, and d: Wire diameter. The torsion is then calculated as PR cos a, the bending moment as PR sin a. the shear force as P cos a, and the compression force as P sin a. 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 a = PR, the following should be valid:

$$\tau = \frac{16PR}{\pi \cdot d^3}. \qquad \text{....1.7}$$

The shear stress here is usually called uncorrected shear stress. The total length l is 2pRn, where n is the number of active coils. Using the fact that c = s/G, it can be rewritten as 16PR/(p _ d3G), and the total angular torsion u becomes

$$\varphi = \int_0^{2\pi Rn} \frac{2\gamma}{d}\mathrm{d}x = \frac{32PR}{\pi d^4 G}\mathrm{d}x = \frac{64PR^2 n}{Gd^4}, \qquad \text{....1.8}$$

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

$$\delta = R\varphi = \frac{64PR^3 n}{Gd^4} = \frac{8PD^3 n}{Gd^4}. \qquad \text{....1.9}$$

The spring rate therefore becomes:

$$k = \frac{P}{\delta} = \frac{Gd^4}{8nD^3}. \qquad \text{....1.10}$$

It is still commonly used to estimate the spring rate by suspension designers. As opposed to the uncorrected shear stress in Wahl 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, sa, is obtained by multiplying the uncorrected stress with a correction factor K, which depends upon the spring index D/d. Fig.1. 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:

$$\tau_a = \frac{32xPR^2}{\pi \cdot d^4 (R - d^2/16R - x)} \qquad \text{....1.11}$$

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 -d2/16R, becomes:

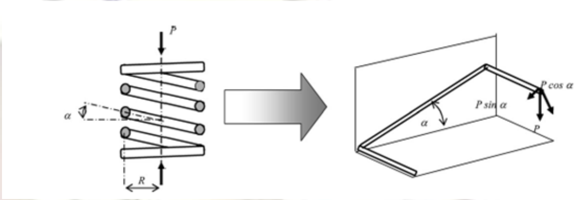

Fig.1.1. Wound and unwound coil spring [4]

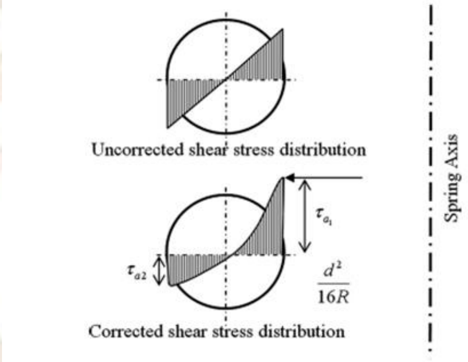

Fig.1.2. Uncorrected shear stress vs corrected shear stress distribution [4]

$$\tau_{a_1} = \frac{16PR}{\pi \cdot d^3}\frac{4c-1}{4c-4} \qquad \text{....1.13}$$

Additional shear stress caused by the neutral surface of a cantilever of circular cross section loaded by force P, the term 4.92P/pd2 should be added to obtain maximum shear stress. [4] Y. Prawoto.

$$\tau_{\max} = \frac{16PR}{\pi \cdot d^3}\left[\frac{4c-1}{4c-4} + \frac{0.615}{c}\right] \qquad \text{....1.14}$$

And minimum shear stress:

$$\tau_{\min} = \frac{16PR}{\pi \cdot d^3}\left[\frac{4c+1}{4c+4} - \frac{0.615}{c}\right] \qquad \text{….1.15}$$

Stresses in helical springs of circular wire:-

1) Twisting moment:- $\mathbf{Fs1} = \mathbf{Kc}\left(\frac{\mathbf{8PD}}{[\Pi \mathbf{d3}]}\right)$ ….1.16

2) Direct shear stress:- $\mathbf{Fs2} = \frac{\mathbf{W}}{\left[\left(\frac{\Pi}{4}\right)\mathbf{d2}\right]}$ ….1.17

3) Max shear stress:-$F_{s=}$ $\frac{\mathbf{8KWD}}{[\Pi \mathbf{d3}]}$ ….1.18

$Fs = F_{s1} + F_{s2}$ (for inner edge)
$Fs = F_{s1} - F_{s2}$ (for outer edge) and
$\mathbf{K} = \left[\frac{\mathbf{4C-1}}{\mathbf{4C-4}}\right] + \left[\frac{\mathbf{0.615}}{\mathbf{C}}\right]$ ….1.19

1.3Objectives

- To know the different parameters which affect design of helical compression spring.
- To know the relation between different parameters.
- To know the F.E.A. approach applied to design of spring.

2. PARAMETERS AND THEIR EFFECTS

Introduction

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.

2.1 Parameters affecting spring strength

Following are some parameters shown on cause and effect diagram which affect the spring strength.

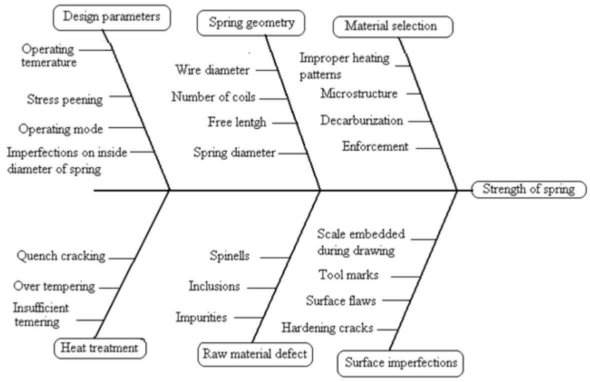

Fig.2.1. Different parameters affecting strength of spring

2.1.1 Material selection

Raw material selection is always the most important decision in obtaining the best quality of any product, including coil springs. The selection of the raw material usually includes the enforcement of cleanliness, microstructure, and decarburization inspection.

The sub parameters in material selection are improper heating patterns, microstructure, decarburization and enforcement. [4] Y. Prawoto.

2.1.2 Raw material defect

A typical raw material defect is the existence of a foreign material inside the steel, such as non-metallic inclusions.Fig.2.2a. and Fig.2.2b. show the fracture surface and SEM fractograph, as well as the EDS spectrum of an inclusion located _1 mm below the surface. This particular coil failed early despite all other parameters being normal. [4]

Fig.2.2a. Fracture surface of a coil that failed early due to an inclusion and Its SEM appearance. [4]

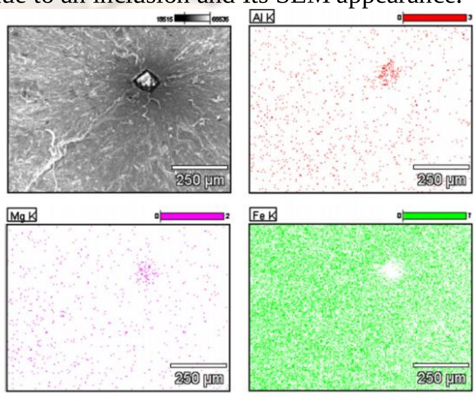

Fig.2.2b. Eds spectrum mapping of the inclusion.[4]

2.1.3 Heat treatment

Improper heat treatment can be easily overlooked since a temperature difference in heating does not relate directly to the hardness of the material. Extensive evaluations are usually needed to identify this problem. Fig .2.3. Shows a typical example of an improper heat treatment. Prolonged heating can cause the prior austenite grain size to grow significantly. Improper heat treatment can also result in the microstructure becoming pearlite instead of the required martensite. This type of defect is easier to identify due to the clear difference in hardness. [4] Y. Prawoto.

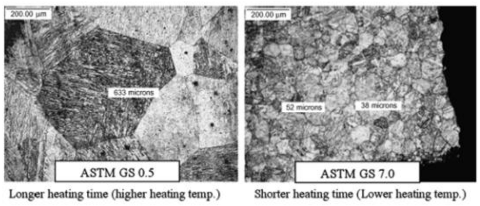

Fig.2.3. Identical raw materials heated with different heating patterns. [4]

2.1.4 Spring geometry

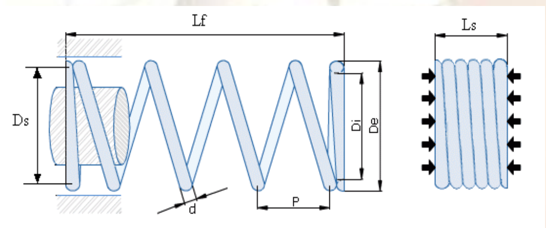

Fig.2.4.Spring geometry

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

2.1.5 Design parameters

The development of metal springs has continued during the past years and the concentration is on reducing the operating weight of spring. Which leads to smaller mounting space due to which the specific stress on spring are continuously increasing. Hence most care is to be taken for careful manufacturing of spring. With special respect to surface layer, hot presetting, 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 helical compression springs.

1) Operating mode

Depending on the application, a spring may be in a static, cyclic or dynamic operating mode. A spring is usually considered to be static if a change in deflection or load occurs only a few times, such as less than 10,000 cycles during the expected life of the spring. A static spring may remain loaded for very long periods of time. The failure modes of interest for static springs include spring relaxation, set and creep.

Cyclic springs are flexed repeatedly and can be expected to exhibit a higher failure rate due to fatigue. Cyclic springs may be operated in a unidirectional mode or a reversed stress mode. In one case, the stress is always applied in the same direction, while in the other, stress is applied first in one direction then in the opposite direction. For the same maximum stress and deflection between a unidirectional and reversed stress spring, the stress range for the reversed stress spring will be twice that of the unidirectional spring and therefore a shorter fatigue life would be expected.

Dynamic loading refers to those intermittent occurrences of a load surge such as a shock absorber inducing higher than normal stresses on the spring. Dynamic loading of a spring falls into three main categories: shock, resonance of the spring itself, and resonance of the spring/mass system. Shock loading occurs when a load is applied with sufficient speed such that the first coils of the spring take up more of the load than would be calculated for a static or cyclic situation. This loading is due to the inertia of the spring coils. Spring resonance occurs when the operating speed is the same as the natural frequency of the spring or a harmonic of the natural frequency. Resonance can cause greatly elevated stresses and possible coil clash resulting in premature failure. [2] William H. Skewis.

2) Imperfections on inside diameter of spring:

Helical compression springs respond to external compressive force with torsional stress caused by torsion of the active spring coils which, in a first approximation, may be estimated analogous to a straight torsion bar. Since the shear angle, however, is greater on the inner coil surface than on the 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 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. [4] This means:

- The maximum stress of helical compression springs occurs on the inner coil surface.
- Accordingly, fatigue fractures of helical compression springs generally originate from this area.
- Therefore, the spring's inner coil surface has to be shot peened with particular care, which depending on the spring geometry, constitutes a highly fastidious task. [4]

Following Fig.2.5. shows Correction factor k to describe the static stress concentration on the inner coil surface of a helical compression spring in dependence on the coil ratio w, factor k0 represents the effect of the stress concentration in the case of cyclic load

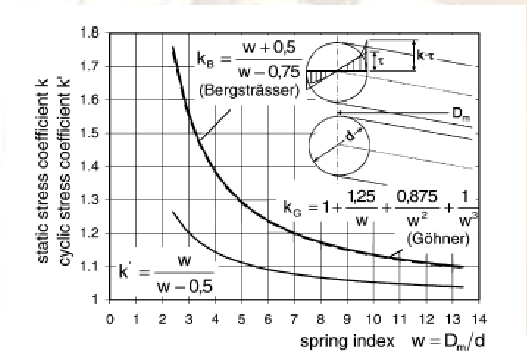

Fig.2.5.Coil ratio vs. stress concentration.[1]

3) Stress peening:

Shot peening is a standard technological procedure. Peening (in the technical context) is the interaction between a particle (with the necessary hardness) and the surface of a working piece. If the particles have a round shape, it is called shot peening1). In the surface layer (up to a depth of 0.5 mm), 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.

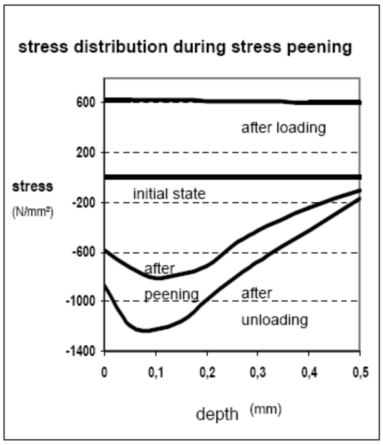

Fig. 2.6. The residual stress distribution

Here the working piece or component is stressed in the direction of the later loading. After this step, the original peening procedure is done, followed by the unloading. The compressive residual stress profile, which is now obtained, is significantly higher than gained by normal peening .The result depends on the (torsional-) preload (τks) σks during peening.

4) Operating temperatures:

Compression springs 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.2.7. This change is completely reversible.

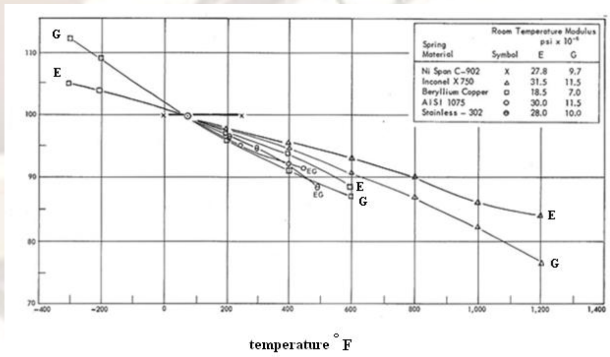

Fig.2.7. Change in modulus with temperature

Also, 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.2.8. 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 and elevated temperature environment.

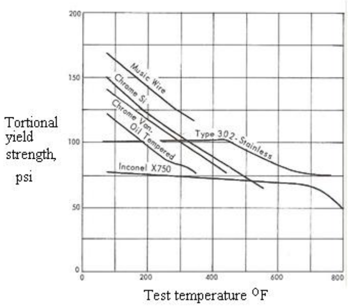

Fig.2.8. Torsional yield strength of spring wire at elevated temperatures

Maximum usable temperatures for spring materials are simply the temperature at which metallurgical change begins. Time dependent changes in springs occur when a sustained stress is applied at an elevated temperature. these changes will occur at room temperature if the stress is high enough. Increasing the temperature merely increases the rate of change. Normally the change occurs as a reduction in helical spring length under load or a reduction in spring load at a fixed length. This relaxation occurs rapidly at first and then at a decreasing rate over time. There is no apparent end point.

3.F.E.A. APPROACHES

Introduction

The errors of simplification of equations are reduced by F E A analysis. The most fundamental underlying concept of finite element method (FEM), or finite element analysis (FEA), is the piecewise approximation of solution of a known geometry for which the characteristics are well established.

Thus, the first requirement of FEM approach is descritization of the physical

domain for which appropriate type of element is required to be selected.

3.1 Designing coil spring using FEA

Reason to use the FEA in coil design is it reduces error caused by the simplification of equations. An FEA based design begins with the selection of the element type, how the model should be constructed, how accurate the results should be, and how fast the model should be run. The most accurate FEA results can be obtained by creating 3D parts of a coil spring and its seats, followed by meshing the parts with a 3D solid element. Finer meshing with higher order elements in general will produce the most accurate results.

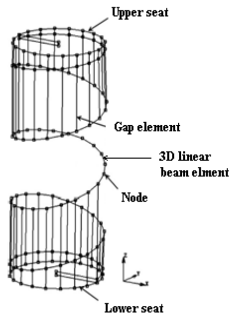

Fig.3.1.Finite element model [4]

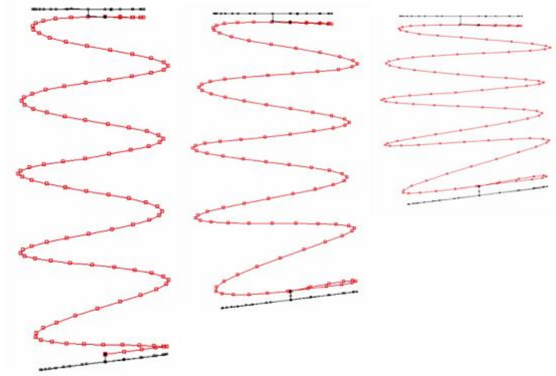

Fig.3.2. Example of compressed coil spring [4]

However, because of the higher number of elements and a non-linearity due to the contact between a coil spring and seat, or the coil itself, each analysis could take hours. While the accuracy of the result is important, the computational time must be reasonable to incorporate FEA into the coil spring design.

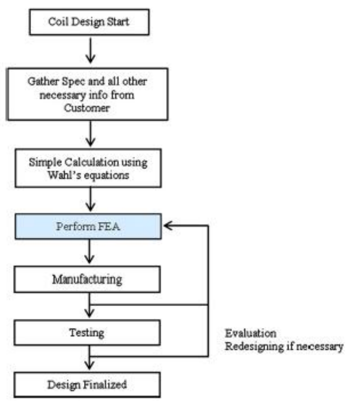

Fig.3.3. Flow chart of coil spring design [4]

To resolve lengthy computational time in a solid model, a 3D beam element is usually selected to model a coil spring and seat. The deformation of a seat under compression is very minimal. the material properties of seats are set very high to act as rigid. Contact between a coil and seat, or the coil itself, is detected by gap elements. A typical FEA model is shown in Fig. 3.1.Fig. 3.2 shows model of coils at free, normal, and compressed coils. Fig. 3.3 shows the steps of coil design. [4] Y. Prawoto.

3.2 Analysis

A finite element analysis was performed to check the local stress distribution around a given defect using a typical coil spring. First, the overall stress distribution was checked without any defect in the material, and then at the location where the highest stress was found, each defect was added. Since the size of the defect is significantly smaller than the whole model, a submodeling technique was used. This technique is used to study a local part of a model with refined meshing based on the FEA result of a global model with coarse meshing. Boundary conditions for the submodel will be automatically interpolated from the global model solution. As shown in Fig. 3.4, the submodeling technique was used twice for this study. Submodel 2 was modified to apply various defects. [4] Y. Prawoto,

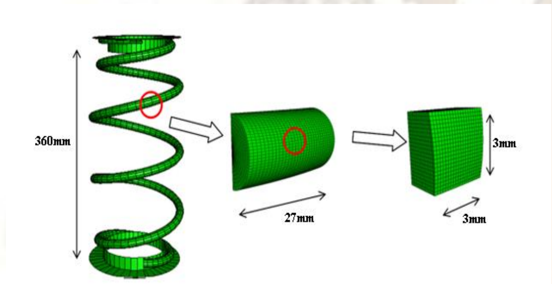

Fig.3.4. F.E.A. model of coil spring and its sub models [4]

3.3 FEA result of model without defect

For comparison with the defect model, Submodel 2 was analyzed first without any defects. Boundary conditions were interpolated from the result of Submodel 1 to the inner and side surfaces of Submodel 2. Fig. 3.5 shows the Von Mises stress distribution. The highest stress was found at the outer surface and the lowest at inner side of wire. The highest Von Mises stress was about 1715 MPa, which matches the stress level of the global model. The gray area around the outer edge shows a stress concentration, however this is ignored since it is where the boundary condition was applied. [4] Y. Prawoto,

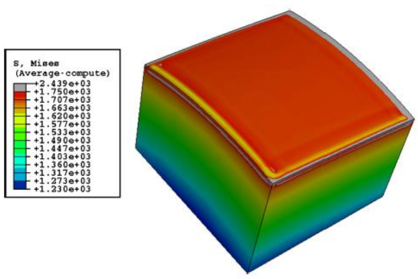

Fig. 3.5. Von Mises stress result of no-defect model. [4]

3.4 Defect FEA models and results

3.4.1Inclusion

A cubic hole was placed about 1 mm below the outer surface; its size is 50 lm (Fig. 3.6, red dot is the inclusion). Instead of using a foreign material for the cubic area, it was left as a hole for simplification. Since a higher stress concentration was expected around the inclusion area, a finer mesh was used at the center and coarser mesh was used at the outer area (Fig. 3.6b).

The stress distribution is shown in Fig. 3.7. As expected, a local stress concentration is observed at the inclusion area, and the highest Von Mises stress reached 2000 MPa, which is higher than the outer surface stress level. Stress on other areas, such as outer surface, was at the same level as the no-defect model. [4] Y. Prawoto,

3.4.2 Imperfection

The surface imperfection is inherited from the raw material. A crack (50 lm width, 500 lm depth) alongside of the centerline of the wire was applied to the Submodel 2 as shown in Fig. 3.8. The stress distribution is shown in Fig. 3.9. A high stress concentration is observed at the crack location, and the Von Mises stress exceeded 4000 MPa, which is much higher than the outer surface stress level. Therefore, the product would likely fail from this point. A stress concentration is also observed at the vertical edge, however this concentration occurred due to the boundary condition and should be ignored. Y. Prawoto,

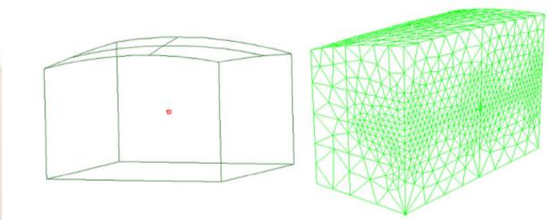

Fig.3.6. Part model with inclusion. (Left), FEA model with inclusion (display model is cut in half to show inside) (right).[4]

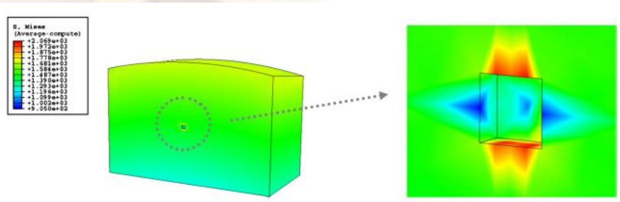

Fig.3.7. Von Mises stress result of inclusion model.[4]

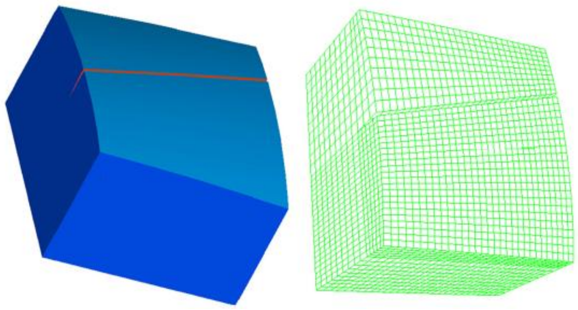

Fig.3.8. Part model with imperfection. (Left) and its FEA model (right). [4]

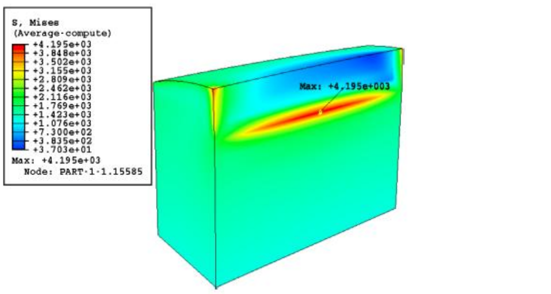

Fig.3.9. Von Mises Stress result of imperfection model (display model was cut at crack location). [4]

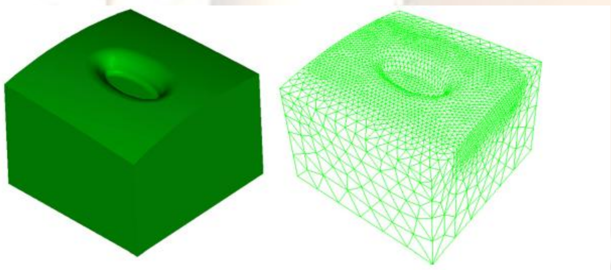

Fig. 3.10. Part model with corrosion (left) and its FEA model (right). [4]

3.4.3 Corrosion

Instead of modeling the actual corrosion part, a simple oval shape was removed from the outer surface to simplify the FEM model. Its size is approximately 300 lm in depth, 500 lm in height, and 1 mm in width. Finer meshing was used around the corrosion area since a higher stress concentration was expected there. The model is shown in Fig.3.10.

‘+6jhdu4

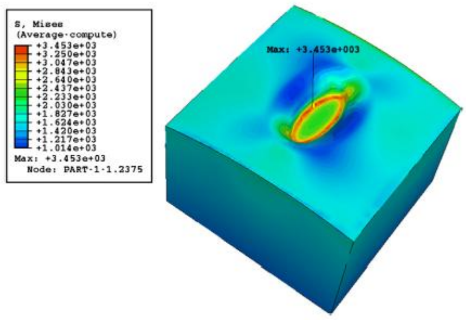

Fig.3.11. Von Mises stress result of corrosion model. [4]

The stress distribution is shown in Fig. 3.11. As expected, a local stress concentration is observed at the bottom edge of the corrosion area, and its Von Mises was about 3450 MPa, which is again much higher than the outer surface stress level. This high stress concentration will cause early spring breakage from this point. [4] Y. Prawoto,

3.4.4 Decarburization

The decarburization model is shown in Fig.3.12. Inner side material is the same as original except yield stress was specified this time for elastic–plastic analysis. Analysis results are shown in Fig. 3.13. The stress level on the decarburized layer reached the yield stress and remained that value because the material was assumed to be perfectly plastic, and the plastic deformation occurred on the decarburized layer. The rest of the part never reached the yield stress; therefore, no plastic deformation was observed inside of the 3*/ecarburized layer. [4] Y. Prawoto,

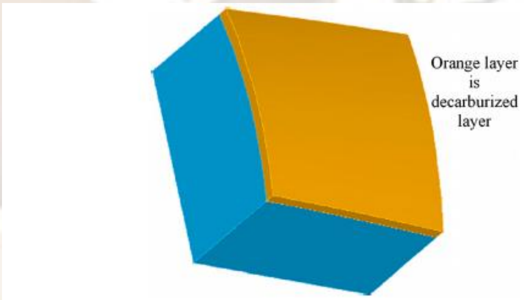

Fig. 3.12. Part model with decarburization [4]

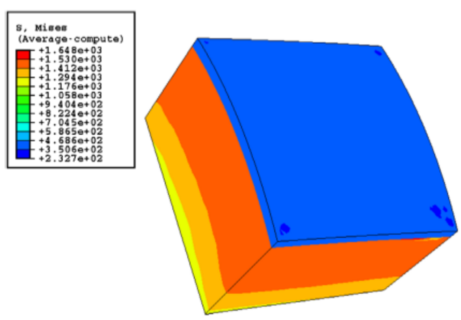

Fig.3.13. Von Mises stress result [4]

3.5 Analysis result summary

Table 1 shows the summary of analysis results. As expected, a local stress concentration was observed in the inclusion, imperfection, and corrosion defect models at each defect area, and those stress values were much higher than the model without any defects. These high stress concentrations will cause an early failure; hence the material needs to avoid these defects as much as possible. [4] Y. Prawoto,

Table 1.Analysis of stress concentration by F E A for different defects. [4] Y. Prawoto,

Defect	Summary
None	No stress concentration. The highest stress was found on the outer surface. Von Mises stress _ 1715 MPa. Max. Principal stress _ 1200 MPa. No plastic deformation occurred.
Inclusion	Stress concentration is observed at the inclusion area. Von Mises stress = 2069 MPa. Maximum principal stress = 1922 MPa.
Imperfection	Stress concentration is observed at the crack location. Von Mises stress = 4195 MPa. Maximum principal stress = 2670 MPa.
Corrosion	Stress concentration is observed at the bottom edge of corrosion surface. Von Mises stress = 3453 MPa. Maximum principal stress = 3286 MPa.
Decarburization	On decarburized layer, the stress reached the yield point, and a plastic deformation occurred.

4. CONCLUSION

- It is seen that major factors that affect the strength of springs are Design parameters, material selection, Raw material defect, Spring geometry and surface imperfection.
- It is seen that design parameters i.e. Operating modes, Operating temperature, shot peening and imperfections on inside the coil spring affect directly on fatigue life of spring, as we seen as temperature increases the modulus and torsional yield strength of spring material decreases.
  It is observed that if the inner side of the coil spring is shot peened the stresses on inside coil surface reduces and fatigue life of coil spring increases.
- It is also seen that presence of any impurity, inclusion in raw material reduces the strength of coil spring.
- In the F.E.A. model of corrosion the linear triangular element is used and for part model of imperfection the linear quadrilateral element is used.

REFERENCES

1) “Result of very high cycle fatigue test on helical compression springs”. C. Berger, B. Kaiser. 27th June 2006.
2) “Failure modes of mechanical springs”. William H. Skewis.
3) “Failure analysis of passenger car coil spring”. S.K. Das, N.K. Mukhopadhyay,B. Ravi kumar, D.K. Bhattacharya. 28th Feb 2006.
4) “Design and failure modes of automotive suspension springs”. Y. Prawoto, M. Ikeda, S.K. Manville, A. Nishikawa.21st feb 2008.