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Experimental validation of multiaxial fatigue theories to estimate fatigue life of helical compression spring

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Abstract

High-stress amplitudes and mean stress cycles are expected to be endured by helical compression springs utilized in a two-suspension wheeler's system. Fatigue failure of all these springs brings enormous eventual repair and replacement costs. A fatigue test is conducted to ascertain fatigue strength. The spring test plan is rather extensive due to the limited time and numerous test spring versions. Efforts were made to anticipate the helical spring's fatigue life. Multiaxial fatigue theories are examined in this work. The research paper aims to scrutinize the appropriate techniques that spring manufacturers should employ in the planning stage to calculate the fatigue life of helical compression springs. The criteria of Wang & Brown, Fatemi & Socie, Mitchell, Baumel & Seeger, and Smith & Watson are used in the present investigation. The results of the experiment and the predicted life are compared. Spring fatigue life is overvalued by the Wang-Brown criterion, while the Fatemi-Socie model offers a precise forecast of fatigue life.

Keywords: *baumel & seeger; fatemi & socie; fatigue life; fatigue criteria; helical compression spring; wang & brown.*

Nomenclature

Symbol	Parameter	Symbol	Parameter
σ_{max}	*maximum normal stress*	k	*normal stress sensitivity coefficient*
σ_y	*cyclic yield stress*	$\Delta\varepsilon_n$	*normal strain*
τ'_f	*shear fatigue strength*	$\Delta\gamma$	*shear strain range*
γ'_f	*shear fatigue coefficients*	$\sigma'f$	*fatigue strength coefficient*
b_0	*shear fatigue strength exponents*	b	*fatigue strength exponent*
c_0	*shear fatigue ductility exponents*	$\varepsilon'f$	*fatigue ductility coefficient*
N_f	*fatigue cycles to failure*	c	*fatigue ductility exponent*
G	*shear modulus*		

1. Introduction

Recent years have seen new difficulties in vehicle design and development, especially for two-wheelers. When riding a motorcycle, the customer expects greater safety and comfort. It is common for these motorcycles to be used beyond their intended load and lifespan. The constant cyclic stress has an impact on the vehicle's suspension spring. This leads to the helical spring failing, which worsens back problems and, consequently, passenger comfort. It also affects vehicle handling, which could cause a major accident. These failures of spring maintenance and repair costs have been steadily increasing for a multitude of reasons. But because performance is important, several worn-out springs are taken out and replaced (Kumar, Anikivi, & Deshpande, 2019).

Fatigue is the progressive, persistent structural change in one area that takes place in a material over time as a result of varying loads and stresses. After a significant number of cycles, it develops cracks or a complete fracture. Fatigue has emerged as a significant design issue for varied forms, including those used by aircraft, bridges, trains, vehicle suspensions, and automobile chassis. Failures from fatigue are frequently disastrous. They strike suddenly and could result in serious property damage as well as fatalities (Pastorcic, Vukelic, & Bozic, 2019).

During the sample stage, the springs must be assessed for their fatigue life. To verify helical compression spring's fatigue lives, fatigue testing equipment is used. The International Standard IS: 4454 establishes a 2 million-cycle limit on fatigue. The fatigue testing equipment requires more than three months for testing, making it a very time-consuming process (Geilen, Klein, & Oechsner, 2020).

An expert designer who can precisely foresee the fatigue is required to save time and money. Researchers have proposed various fatigue-life prediction approaches. The fatigue life could be determined using the approach that produces the most reliable results.

Referring to the applied loading state parameter, three categories of multiaxial fatigue failure are characterized as stress, strain, and energy-based criteria. As part of a critical plane concept-based set of criteria, conversion of a multi-axial stress to an analogous uniaxial is required (Karolczuk, & Papuga, 2019).

Kamal, and Rahman (2018) investigated cutting-edge research activities related to the building of models that estimate fatigue life. According to the author, there are several ways to estimate fatigue life, including the critical plane deviation and the customized Wöhler curve.

Sajith, Murthy, and Robi (2020) compared the forecasted life to the investigational fatigue life and concluded that the models of Irwin and Tanaka accurately forecast the combined-mode fatigue life comparable to the empirical observations.

Vijayanandh et al., (2020) used Ansys Fluent 16.2 to assess the fatigue life of propellers using a variety of materials in different ocean environments (Vijayanandh et al., 2020).

Gates, and Fatemi (2018) concentrated on estimating multiaxial variable amplitude fatigue life for unnotched samples. Estimates of these fatigue life calculated using the critical plane were compared to investigational data acquired with an un-notched test sample.

Mulla (2016) estimated the fatigue life of a helical coil compression spring using a finite element analysis in the time domain. To calculate spring life with respect to varying amplitude loading, Morrow's and Smith-Watson Topper's equations were applied.

Al Musalli, Ali, and Esakki (2020) concentrated on using finite element analysis (FEA) to examine the fatigue performance of helical springs using stress-strain-life techniques. To anticipate the fatigue life and calculate the greatest shear stress, FEA findings were compared to theoretical calculations based on stress and strain life. It was reported that the strain-life methodology was better able to forecast fatigue life than the stress-life technique.

Hamzi, Singh, Abdullah, and Rasani (2022) attempted to examine the fatigue life parameters of a coil spring exposed to an arbitrary strain load with respect to time. The Goodman, Brown-Miller, Fatemi-Socie, and Wang-Brown fatigue life approaches were used to evaluate fatigue life. Brown- Miller's fatigue life was maximum.

Nya, Abdullah, and Singh (2022) examined the fatigue-life predictive model based on a condition monitoring system for coil springs in car shock absorbers. The Coffin-Manson, Morrow, and Smith-Watson-Topper models were used to assess fatigue life. As per a fatigue-life prediction stability analysis, the Morrow model predicted a safe zone of a life for the different road surfaces.

Abdullah et al., (2019) sought to decide the consistency evaluation of the anticipated fatigue behavior of leaf springs with unplanned strain loads. Strain-life approaches were used in the Coffin-Manson, Morrow and Smith-Watson-Topper (SWT) models to project fatigue life. When evaluating dependability based on fatigue life, the SWT model has the shortest fatigue life in contrast to the Coffin-Manson model.

Mozafari, Thamburaja, Moslemi, and Srinivasa (2021) built a new model to envisage multiaxial fatigue with combined loading and small-strain micro-plasticity. In the fatigue calculation, pre-full-yield micro plastic deformations were presumed. The fatigue life was related to the total plastic dissipation's accumulated (micro) plastic work. It was found that the generated theory and its application to finite elements can more precisely foresee experimental fatigue life under multiaxial loading.

Zhu, Yu, Correia, De Jesus, and Berto (2018) compared critical plane principles for the examination of ductile/brittle materials, comprising Fatemi-Socie, Wang-Brown, modified Smith-Watson-Topper, and the proposed modified generalized strain energy concept (MGSE). According to the findings, criteria with supplementary material constants gave more accurate life calculations for various materials. The shearing and uniaxial fatigue criterion was applicable to both brittle and ductile components, while MGSE was surpassing others within ductile and brittle materials and MSWT in brittle materials.

Deng, Zhu, He, Li, and Carpinteri (2022) evaluated recent research on the critical plane method with multiaxial variable amplitude loading. This study summarizes the four major features of multiaxial fatigue, the technique for determining the critical plane, the cycle measurement and the damage buildup principle.

Liao, Zhu, and Qian (2019) investigated the combined critical plane approach and critical distance theory for fatigue examination of notched parts subjected to multiaxial state. They adopted the Fatemi-Socie model for several coupling sequences of the critical plane and critical length ideas.

2. Objectives

Helical compression springs utilised in a two-wheeler suspension system are expected to withstand a significant cycle with high stress amplitude and mean stress. Fatigue failure of such a spring frequently results in significant consequential damage and repair costs. The fatigue test is applied to measure fatigue strength. The spring test plan gets extremely extensive due to the restricted time and high number of test spring variants. As a result, efforts were made to forecast the fatigue life of the helical spring, which will provide an exact prognosis of the spring's fatigue life. Several academicians have presented multiaxial fatigue criteria that are applicable to various materials and machine components under various stress conditions. Despite a wide range of requirements, there is no universally accepted model. Based on the literature and previous results, Wang, and Brown (1993), Fatemi, and Socie (1988), Mitchell (1992), and Smith (1970) criteria are evaluated and compared extensively.

According to the preceding, the research objectives are to investigate and validate various fatigue criteria for the fatigue life of the helical compression spring used in the Bajaj CT-100. The goal of this work is to discuss the right methodologies for spring producers to use during the design process to guess the fatigue life of a helical spring. The fatigue life of the spring is estimated with the help of different approaches and compared with experimental results.

3. Multiaxial fatigue criteria

Numerous fatigue models have been presented during the past few years to forecast the fatigue life under uni-axial stress conditions. Yet, in real life situations, investigators find it difficult to assess the challenge from the perspective of the uniaxial stress state due to complex geometries and loading patterns. As a result, numerous criteria and methodologies were proposed to portray and forecast the fatigue life of various materials under service conditions. These criteria predict fatigue life using multiple techniques. Multiaxial fatigue criteria can be grouped into four categories: stress, strain, energy-based and critical plane.

Equivalent stress methods are fatigue-specific extensions of static yield criteria. The maximum shear stress theory, the maximum principal stress theory, and the octahedral shear stress theory are the three most popular equivalent stress approaches for fatigue. The most popular equivalent stress criterion for multiaxial fatigue of materials with a ductile behaviour is octahedral shear stress criterion (von Mises). For multiaxial fatigue of brittle materials, maximum principal stress criterion is typically recommended. Strain-based aspects of the strain-life curve are used in circumstances where considerable plastic deformation is possible, such as in low cycle fatigue or at notches. Energy-based approaches quantify fatigue damage by utilizing stress and strain products. The physical nature of fatigue damage is represented in critical plane techniques.

Del Llano-Vizcaya, Rubio-González, Mesmacque, and Cervantes-Hernández (2006) investigated the appropriateness of multiaxial fatigue criteria for compression springs. The objective was to figure out which kinds of criteria, as well as the technique behind them, have the maximum chance of accurately predicting mechanical spring fatigue life. To assess the various criteria, foreseen lives were compared to the outcomes of investigations.

Assume a helical spring exposed to axial load F to evaluate the stresses induced in the spring, as shown in Figure 1(a). Assume that the spring is sectioned at some point, as shown in Figure 1(b). Internal forces are produced to keep the spring in equilibrium. There is a direct shear force F and a torque T. The following equation is used to compute the maximum shear stress on a wire (Shigley, & Mitchell, 1993; Carlson, 1978).

$$\tau \max = \pm \frac{T}{J} + \frac{F}{A} \quad (1)$$

where J denotes the polar moment of inertia, A denotes the area, and r indicates the wire radius. The first part describes torsion and the next part reflects direct shear stress.

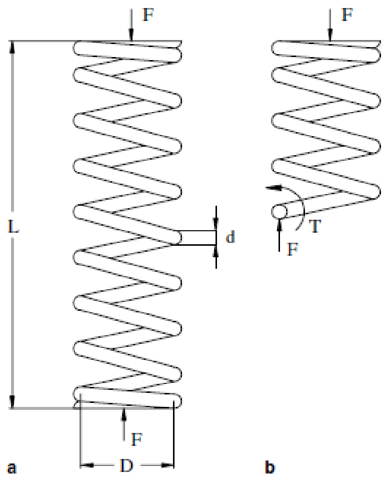

Figure 1 (a) Helical spring with axial load and (b) free body diagram. (Del Llano-Vizcaya et al., 2006)

The last equation can be simplified by assuming a wire with a diameter d:

$$\tau = K \ 8FD \ / \ \pi d^3 \quad (2)$$

where D represents the average coil diameter, K denotes the Wahl factor, and C signifies the spring index.

$$K = 1 + \frac{0.5}{C} \quad (3)$$

When the external load F changes, so does the induced stress. The mean stress τ_m and the amplitude τ_a are defined by:

$$\tau m = Ks \ 8FmD / \pi d^3 \quad (4)$$

$$\tau a = Kb \ 8FaD / \pi d^3 \quad (5)$$

where Fm and Fa are the mean load and load amplitude, respectively, and Ks and Kb are curvature-related correction factors (Shigley, & Mitchell, 1993). The Coffin-Manson equations were used to correlate the fatigue life under axial and torsional loading,

$$\frac{\Delta\epsilon}{2} = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \text{ for axial loading} \quad (6)$$

$$\frac{\Delta\gamma}{2} = \frac{\tau'_f}{G} (2N_f)^{bO} + \gamma'_f (2N_f)^{co} \text{ for torsional loading} \quad (7)$$

where $\triangle\epsilon$ and $\triangle\gamma$ are the strain range in axial fatigue and the shear strain range in torsional fatigue, respectively. σ'_f and ε'_f denote the axial fatigue strength coefficient and fatigue ductility coefficient, correspondingly; E represents the Young’s modulus and G is the shear modulus, respectively. These equations serve as the foundation for determining the fatigue parameter vs. life relationships for multiaxial fatigue (Del Llano-Vizcaya et al., 2006).

Based on the literature review criteria suggested by Wang-Brown and Socie-Fatemi, the approaches of Mitchell, Smith-Watson, and Baumel-Seeger are investigated to find out the best approach that can give comparable results with the experimental observations.

3.1 Wang and Brown criteria

Wang, and Brown (1993) suggested a multiaxial fatigue criterion that assumes fatigue life has a non-linear relationship with strain. The plane having the greatest shear strain is the critical plane.

Only when proportional loading is assumed along with fixed primary strain orientations is this concept of the critical plane accurate. They suggested the following criterion to find out the fatigue life with respect to non-proportionality and varying strain amplitude.

$$(\Delta\gamma_{max}/2) + s.\varepsilon^*_n = ([1 + v + (1 - v). s](\sigma`_f - 2\sigma_n^0)/E)(2N_f)^b + (1.5+0.5s)\,\varepsilon`_f(2N_f)^c \quad (8)$$

Where S is the model coefficient. $\Delta\gamma$ is the maximum shear strain range amplitude. ε_n is the normal strain on the maximum shear strain plane. σ_f' and b are the fatigue strength coefficient and fatigue strength exponent, respectively. σ_n^0 is the mean normal stress. v_e and v_P are elastic plastic Poisson's ratio respectively (Wang, & Brown, 1993).

3.2 Socie and Fatemi approach

Fatemi, and Socie (1988) investigated fatigue fractures and discovered that opening cracks accelerate the fatigue crack process. This is due to normal strain acting in tandem with maximal shear strain. The force of friction between slip planes, preventing crack propagation, is reduced by crack opening. In this approach, the maximum shear strain amplitude and the maximum normal stress acting on the maximum shear strain magnitude plane are the parameters controlling fatigue life. Therefore, the following criterion incorporated the mean normal stress and greatest shear strain amplitude,

$$(\Delta\gamma/2) + (1+ \kappa(\sigma_{max}/\sigma_y)) = (\tau`_f/G)(2N_f)^{b0} + \gamma`_f(2N_f)^{c0} \quad (9)$$

where $\Delta\gamma$ represents the maximum shear strain range, σ_{max} is the maximum normal stress on the plane of $\Delta\gamma_{max}$, σ_y is the yield strength and k is an empirically calculated constant based on axial and torsional fatigue data (Fatemi, & Shamsaei, 2011).

Fatemi et al., (2011) observed that the Brown and Miller criteria, which solely consider strain values, exclude additional material hardening that takes place under asymmetrical loading. They altered the Brown and Miller criterion to account for this effect by swapping out the value of normal strain with the value of maximal normal stress. The crucial plane is where the shear strain amplitude γ_a is at its greatest. As a result, this model works well for materials where a significant proportion of the fatigue life is expended in crack development and small crack propagation along the maximum shear planes.

3.2.1 Material fatigue constants

Some studies have tried to determine fatigue properties from simple mechanical monotonic properties data. These estimating techniques involve empirical correlations between various collections of monotonic properties data. The competency to estimate fatigue properties using only monotonic properties offers a faster and less expensive method of obtaining fatigue characteristics. In the beginning phases of a design process, this is especially helpful. It will take a number of design reiterations to arrive at a design that can be pushed to the concluding design stage at this point because there are usually a lot of unknowns regarding the design.

Table 1 Axial and fatigue properties

Parameter	Axial	Shear
Fatigue strength coefficient	σ'_f	$\tau'_f = \sigma'_f/\sqrt{3}$
Fatigue strength exponent	b	$b_o = b$
Fatigue ductility coefficient	ε'_f	$\gamma'_f = \sqrt{3}\varepsilon'_f$
Fatigue ductility exponent	c	$co = c$

Manson first introduced a technique for estimating the strain-life relationship from linear characteristics, which Muralidharan later modified. Any metal can be processed using the technique, which is typically considered as the modified universal slopes technique. Tables 1 and 2 show correlations given by Mason and Muralidharan. e_f is true fracture strain. Rm denotes the ultimate tensile strength (Muralidharan, & Manson, 1988).

Table 2 MM parameters

Parameter	Modified universal slope(MM)
σ'_f	$0.623\, R_m^{\,0.823}\, E.^{0.168}$
b	-0.09
ε'_f	$0.0196\, e_f^{\,0.155}(R_m/E)^{-0.53}$
c	-0.56
K'	$\sigma'_f/\varepsilon'f^{0.2}$
n'	0.2

3.3 Mitchell approach

Mitchell (1992) proposed a method for calculating the parameters of fatigue life and suggested that fatigue strength and the exponent b are both functions of $\sigma_{ut.}$ Mitchell method is intended for steels with harnesses less than 500 HB. The actual fracture stress is supposed to be represented numerically by the fatigue strength coefficient, and the total strain at fracture is represented by the fatigue strain coefficient. Additionally, Table 3, Mitchell (1992) made the assumption that c = -0.6 applied to ductile materials while c = -0.5 applied to strong alloys (Mitchell, 1992).

$$\Delta\varepsilon/2 = (\sigma`_f/E)(2N_f)^{-(1/6)\log(2\sigma`f/\sigma ut)} + \varepsilon`_f(2N_f)^{-0.6} \quad (10)$$

Table 3 Mitchell Parameters

Parameter	Mitchell parameters
Fatigue strength coeficient	$\sigma`_f = R_m + 345$
Fatigue strength exponent	$b = -1/6 \log\{2(R_m + 345) / R_m\}$
Fatigue ductility exponent	$\varepsilon`_f = \varepsilon_f = \ln(100/100\text{-RA})$
Fatigue ductility coeficient	c=-0.6

3.4 Smith and Watson criteria

Smith (1970) offer a good approximation for materials that fracture in a tensile manner (Table 4). The maximum stress and strain are the parameters that control damage in this model, which can be utilized for in phase multiaxial loading conditions. The model assumes that cracks would develop at right angles to the greatest tensile stress (Smith, 1970).

Typically, the strain-life equation is solved using the Smith, Watson, and Topper technique as follows:

$$\sigma_{max}\varepsilon_a = \frac{(\sigma`_f)^2}{E}(2N_f)^{2b} + \sigma`_f\varepsilon`_f(2N_f)^{b+c} \quad (11)$$

Table 4 Smith and Watson parameters

Parameter	Smith and Watson
Amplitude Stress	$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$
Mean Stress	$\sigma_{mean} = \sigma_{max} + \sigma_{min}/2.$
Maximum Stress	$\sigma_{max} = \sigma a + \sigma m.$
Axial Strain Range	$\varepsilon_a = \Delta\varepsilon$

3.5 Baumel and Seeger approach

In order to approximate the material variables utilized in the local strain technique without using the comparatively expensive material test, Bäumel, & Seeger (1990) created the uniform material law (Table 5). It results in the equation shown below:

$$\Delta\varepsilon / 2 = 1.67(\sigma_{ut}/E).(2N_f)^{-0.095} + 0.35(2N_f)^{-0.69} \quad (12).$$

Table 5 Coefficients of uniform material law

Parameter	Baumel and Seeger
$\sigma`_f$	$1.5R_m$
b	-0.087
$\varepsilon`f$	0.59Ψ
c	-0.58
$K`$	$1.65 R_m$
$n`$	0.15

3.6 Fatigue Life Calculation

A fork damper spring is intended to cushion impacts from road surface defects in order to provide a comfy ride for motorcyclists. The first step in a fatigue study is to determine the internal loads on the component of interest. The next step is to decide the spring coil's worst-case stress location. The internal force must be determined after the reaction forces on the spring have been determined. A vertical shearing force and a moment are the internal forces acting on the coiled spring. It should be obvious from inspection that the worst-case stress occurs on the inside of the coil. As a result, the maximum stress magnitude occurs inside the coil. Following the determination of the stress in each case, the principle of superposition can be used to calculate the total stress acting on the inside of the coil. The worst case stress location was determined using the superposition principle. Finally, a stress concentration factor was developed to account for stress concentrations in springs. The fluctuating load is used in fatigue analysis. Following the determination of the minimum and maximum stresses, the mean and alternating stresses must be determined because these two quantities are plotted on a fatigue-life diagram (Juvinall, & Marshek, 2020).

The basic dimensions of the spring and the material's properties are taken into consideration when calculating the forces exerted on the provided helical spring. Different fatigue parameters are calculated using the above equations, and from this fatigue, the life of the helical spring is calculated for the various models described above. The calculated fatigue life and the outcomes of the experiment are then to be compared.

4. Experimental Methodology

4.1 Fatigue Testing Machine

Figures 2 and 3 depict the fatigue testing machine. The machine, made specifically by Solar Man Engineering Project (P) Ltd, is used to test front forks and rear shock absorbers with single or multiple pitch coil springs. It is based on the cam and follower mechanism principle. This system enables force action along the spring axis.

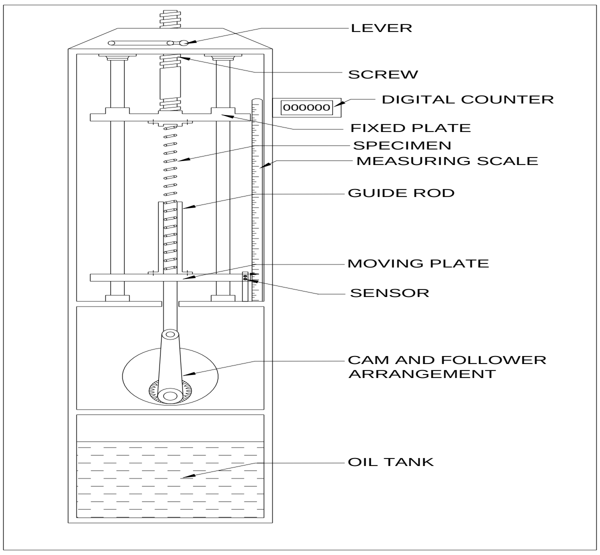

Figure 2 Fatigue testing machine layout

Figure 3 Fatigue testing machine

4.1.1 Experimental procedure

The wire is made of hard drawn MB, high carbon steel. The mechanical characteristics and chemical composition of spring material are shown in Tables 6 and 7. The springs were made by cold coiling and stress alleviation heat treatment. The parameters of the springs were shown in Table 6. These springs are employed in equipment for the automotive industry. The heat treatment is given for 30 minutes in an electrical furnace with an automated thermostat at 320°C.

Table 6 Specification of spring

Wire Diameter (d)	3.8 mm
Coil Diameter (D)	23.3 mm
Total Number of Coils	60.5
Free Length (L_o)	449 mm
Total Number of Active Coils	58.5
Stiffness (k)	4.905 N/mm
Ultimate Tensile Strength (Sut)	1490 N/mm^2
Modulus of Elasticity (E)	177 GPa
Modulus of Rigidity G)	80 GPa
Reduction in Area (RA)	40%

Table 7 Chemical composition of spring material

Carbon	0.60-85 %
Manganese	0.8 %
Sulphur	0.040 %
Phosphorous	0.040 %
Silicon	0.15 – 0.35 %
Cuprous	0.15 %

The number of cycles at which failure of the helical compression springs occurred was determined using a fatigue-testing apparatus. Cam and follower mechanism enables the spring axis to be forced. The conditions for the fatigue tests were a fixed mean stress τm = 875.85 MPa and a variable stress amplitude. Amplitude displacement was modified for every trial to acquire the required values of τ_m and τ_a in accordance with Eqs. of (τa) and (τm). The test was completed until the spring fractured. The number of cycles was recorded when the failure occurred, and a second test was performed using a fresh spring. The S-N curve was drawn as a result of this process. Table 8 shows the experimental observations.

Table 8 Experimental observations

Sr. No.	Maximum Load N	Minimum Load N	Amplitude Load N	Amplitude Stress (N/mm^2)	Experimental life Cycles
	(F_{max})	(F_{min})	(F_a)	(τ_a)	N_f
1	1070	10	530	714.032	25000
2	1000	80	460	619.725	151569
3	900	180	360	485.002	1.60E+06
4	800	280	260	350.279	6.90E+07
5	750	330	210	282.918	1.06E+09
6	700	380	160	215.556	2.36E+10

5. Results and discussion

With an aim of assessing the accuracy of various approaches with regard to the experimental life of helical compression springs, experimental and numerical studies on the application of multi-axial fatigue criteria for helical compression springs have been conducted. S-N diagrams with varying amplitude loadings were used to analyze the data. A view of the variation in fatigue lifetimes of helical compression springs according to various criteria was obtained from the study. Thus, the fatigue life has been predicted and compared to experimental data for conditions of varying amplitude loading.

Figure 4 shows that under higher stress, the number of cycles to failure is low and as stress decreases, the number of cycles to failure increases. A fatigue limit, endurance limit, or fatigue strength is the stress amplitude value below which the material will not fail for any number of cycles. After, 150 N/mm^2, the curve becomes straight i.e. below which no failure takes place. Hence, the Wang, and Brown (1993) and Fatemi and Socie limit is assumed to be 150 N/mm^2. Experimental results show that the fatigue limit for the given material spring is approximately 215 N/mm^2.

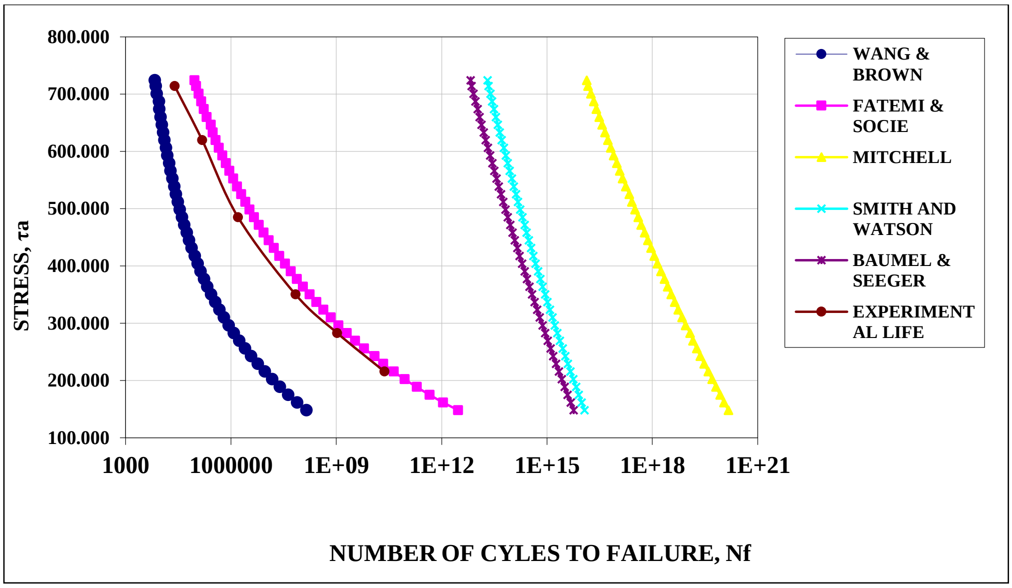

Figure 4 Stress vs. Number of cycle- combined s-n curve

The S-N curve for Smith and Watson, Baumel and Seeger Mitchell Multi-axial fatigue criteria curve is very steep, and follows a straight path, which means it is difficult to point out the fatigue limit. The S-N curves are far from the experimental results. As a result, it is unsuitable for predicting fatigue life.

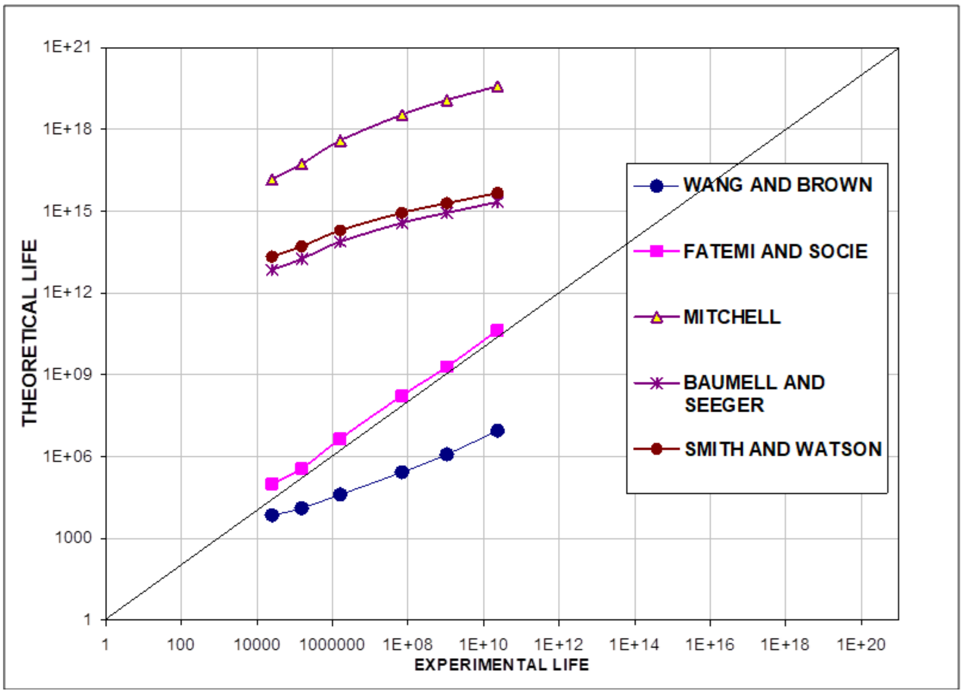

Figure 5 Theoretical life vs Experimental life

Figure 5 presents the results obtained from the experimental fatigue life based prediction criteria, enabling one to evaluate the predictive accuracy of estimated versus experimental fatigue characteristics. The slope of the curve obtained resembles the 45-degree line at its initial point and follows the same path as more readings are plotted. Figure 6 shows the predicted and experimental fatigue life calculation errors for the Fatemi–Socie and Wang-Brown, which indicate that the error percentage is less than 1 for the Fatemi-Socie life prediction model. Hence, the Fatemi and Socie approach gives better results compared to the Wang, and Brown (1993) approach.

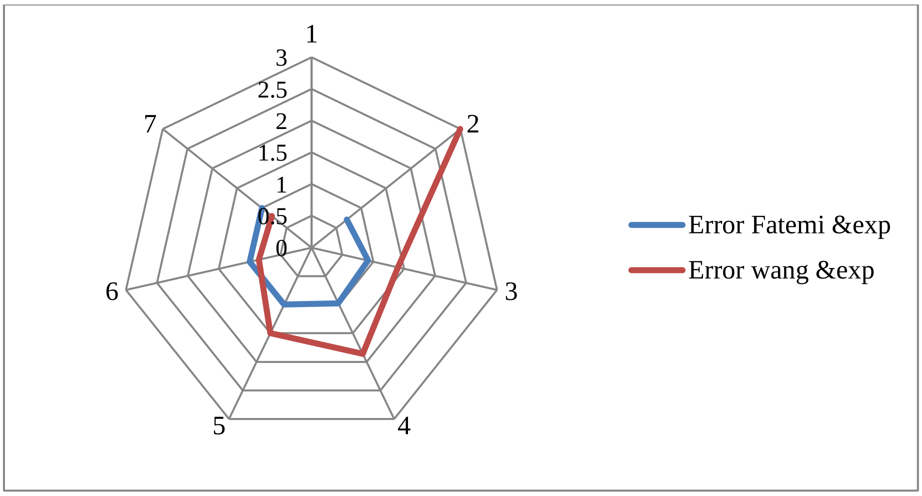

Figure 6 Experiemental vs predicted life error

The method suggested by Fatemi and Socie was successful in predicting fatigue life. Fatemi and Socie observed that the Brown and Miller criterion, which solely considers strain values, excludes additional material hardening that takes place under non-proportional loading. They changed the Brown and Miller criterion to account for this effect by substituting the maximum normal stress value for the normal strain value in the critical plane.

6. Conclusion

In order to assess the reliability of various multi-axial fatigue criteria for helical compression springs, with regard to the experimental life of helical compression springs, experimental and numerical studies have been conducted. Fatigue tests have been conducted on helical compression springs manufactured from AISI MB steel to evaluate the S-N curve at τ_m = 875.85 MPa fixed mean stress. With the Wang, and Brown (1993), Fatemi, and Socie (1988), Mitchell (1992), and Smith (1970) methods, exploratory fatigue lives were evaluated by comparing them to multiaxial fatigue criteria predictions. The results have been analysed using S-N diagrams, involving variable amplitude loading. By comparing all theoretical criteria's with investigational fatigue life, it can be spring fatigue life is exaggerated by the Wang-Brown criterion. The findings of this research are thought to be useful for spring fatigue design techniques, helping to choose the best fatigue model to examine fatigue in helical compression. This will save the manufacturer time and money.

7. Acknowledgments

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