CAPITAL UNIVERSITY OF SCIENCE AND TECHNOLOGY, ISLAMABAD

Design and Analysis of Composite Leaf Spring

by

Mawiz Muaz Tariq

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To my parents

Mr. Tariq Mahmood

Mrs. Atiya Naz Tariq

My Grandparents(Late)

My wife Tooba Mawiz

And

It is with my deepest gratitude and warmest affection that I dedicate this thesis to my Supervisors "Dr. Waqas Akbar Lughmani" who have been constant source of knowledge and inspiration for me in this whole period.

CERTIFICATE OF APPROVAL

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Abstract

The mechanics of composite material laid the foundation for the utilization of composites in various fields. The demand of composite materials has been highly augmented in vehicle industry. In this work, I investigated the use of composite material for leaf spring of light weight vehicles. For this purpose, I selected the leaf spring of Suzuki Mehran car for the study. The main goal of this effort is to obtain optimal geometry for leaf spring with light weight and that can sustain external loading without failure. The analytical and numerical analysis were performed on the leaf spring under static loads. The values of stress and deflection obtained from numerical analysis were verified by analytical analysis while optimization was performed on the basis of the results obtained from static stress analysis to decrease the weight of the composite leaf spring while conserving its strength. Also, the dynamic analysis was performed to determine maximum cycles of failure under dynamic loads. Moreover, stresses against each dynamic load were also determined. The composites used to design such leaf spring were Kevlar/epoxy and E-glass/Epoxy. The results of values of stresses and deflection obtained from analytical and numerical analysis for composite leaf spring were compared and are in compliance. The results of optimization showed a linear variation of width with length and linear variation in thickness along the length from eyes to the axle seat. The proposed optimization method can be efficiently used to reduce the weight of the springs. However, the modal analysis showed that natural frequency of the composite leaf springs was higher than the steel leaf spring. Finally, the result of dynamic analysis shows that Kevlar/Epoxy leaf spring has better tendency to sustain dynamic loads up to 90000 cycles at the speed of 120 km/h as compared to E-glass/Epoxy leaf spring which can sustain dynamic loads up to 60000 cycles at the speed of 120km/h. Furthermore, comparison based on strength and weight among both conventional and composites-based leaf spring was conducted. As compared to steel, a weight economy of 89.1% was achieved for optimized leaf spring using Kevlar/epoxy composite and weight minimization of 88.2% was accomplished. with E-glass/Epoxy composite leaf spring. The stresses produced in composite leaf springs was calculated to be 268.51 MPa for Kevlar/epoxy material

and 145.38 MPa for E-glass/epoxy material which is less than the stress found in steel leaf spring which was calculated to be 965.2 MPa.

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Chapter 1

Introduction

In automobiles, the leaf spring are crucial components to provide comfortable ride and stability to the vehicle. The requirement to replace leaf spring with more robust and durable leaf spring is the major concern in transport and automotive industry. With this datum it was expected that the vehicle would be more reliable, comfortable and faster. In recent years multiple researches, on metallic and composite materials were carried out to study the application of leaf springs and revealed that the use of composites in vehicle suspension can have accountable significance. This chapter explain briefly about composite and conventional leaf spring. The research aims and objectives are also presented in this chapter.

1.1 Background of the Study

The suspension system is significant mechanism to provide strength and protection to the frame of vehicles. However, the suspension system is the area of concern for manufacturers to decrease the load of an automobile by reducing its weight of suspension. The suspension system weights accounts for 15-20 percent of un-sprung weight. Furthermore, the suspension systems of automobiles are also developing regularly to provide protection from impact loading, prevent chassis distortion and damage [1].

Leaf springs and helical springs are major components of suspension system used in vehicles, but in comparison to the helical springs, leaf springs are used more frequently due to their better tendency to absorb shock loads in the vehicles [2]. The leaf spring suspension systems are the key elements for reducing weight in automobiles due to this reason fuel efficiency and riding quality increases. The benefits of leaf spring include its simple and low-cost design. However, variety of leaf springs used in vehicles depending upon the gross weight of the vehicle [3]. The difference between several types of leaf spring depends on their shape. The parabolic leaf spring and normal leaf spring have different number of leaves piled together. Though, normal leaf spring requires a greater number of leaves than parabolic leaf spring. This is due to parabolic shape provides uniform stress distribution [4].

It is well acknowledged that the conventional steel leaf spring provides the necessary strength and stiffness to sustain the weight of the vehicle and appropriately performs its function. But it has some discrepancies during its period of operation. Initially the issue that arises is the weight and the ultimate strength of an ordinary leaf spring. Generally, the eye of the spring close to the shackle and also the curvature of the spring at the centre of the leaf spring gets weaken with the motion of the vehicle causing a reduction in comfort level thus resulting in catastrophic failure. However, this drawback of steel leaf spring can be overcome by enhancing the mechanical properties of steel. Thus, the enhanced material properties help in weight reduction and increased strength, which is also achieved by design optimization and improved manufacturing processes.

The most common failure in leaf spring is the fatigue failure which occur due to high dynamic loads. These dynamic loads significantly affect the leaf spring as it holds the whole mass of the vehicle. The dynamic loads are generated due to unevenness of the road resulting the wheel of the vehicle to move with irregular frequencies and producing repetitive loading on spring[5]. However, there are numerous ways to increase the fatigue life of the conventional leaf spring. Several surface processing techniques were also demonstrated by the authors among which shot peening practice is extensively used for increasing the strength of the steel leaf spring as it is inexpensive and simple method. To enhance the fatigue strength of conventional leaf spring by shot peening method which is a process of enhancing the fatigue strength of the spring by imparting residual stresses. Also, appropriate and effective conditions for shot peening method may be used [6]. Also, the residual stresses are handled carefully to enhance fatigue performance of the leaf spring. Moreover, fretting in due to inter-leaf contact in multi-leaf spring occurs among leaf surfaces resulting in miniature cracks owing to stress concentration. However, fatigue caused by fretting can be overcome by refining surface of leaves. This can be achieved by increasing hardness of material, impart compressive residual stresses and decreasing movement amplitude [7]. The material optimization is another method, in which experimentally verified materials are used for leaf spring design for given loading. The optimization techniques are also used to achieve optimum geometry. The geometry of the spring was first modeled and analyzed using numerical methods. The stresses at various points in design was calculated and optimized to get the optimal design.

P. Beardmore [8] revealed that the role of composite materials in leaf spring can have significant weight savings and can be efficiently used for lightweight vehicles because of their valuable properties over steel. Composites are the material that encompasses mixture of two or more constituents that have cumulative properties of the two materials combined, but it differs from its constituents at microscopic level. These constituent materials include matrix and reinforcement. However, their major function is to prevent mechanical or environmental damage and improve mechanical properties including strength and stiffness [9]. These materials possess good stiffness, high specific weight, high elastic strain energy absorption and specific modulus thus used in various structures in automobile, chemical, aero industries etc. Composite materials have replaced conventional materials used for manufacturing to conserve natural resources and to fulfil the demand of manufacturers by reducing weight.

The complex structural and manufacturing process of composite materials produce a various type of failure mechanisms such as; fiber-matrix debonding, matrix cracks, fiber failure, fiber buckling and delamination. However, it makes the material strength as the primary focus to execute the operation effectively. Therefore, the material behaviour must be studied and analysed, to address the real-time stress/strain state of the material. This behaviour may fluctuate with change in several parameters like material properties, mode of application and boundary conditions. Moreover, due to brittleness of composites, impact due to low velocity causes delamination under the surface while the upper surface remains intact. Such damage is barely visible which lead to substantial decrease in local strengths and can gradually grow further to reduce the stiffness and result into the biggest damage [10,11]. Thus, two issues need to be addressed i.e. to decrease the weight of the leaf spring irrespective of its strength and to reduce delamination beneath the upper surface under external loads.

In this work the composite materials were used instead of metallic materials to design leaf spring. For this purpose, the material chosen on the basis of strain energy to model the composite leaf spring are E- Glass/Epoxy and Kevlar/Epoxy which is compared with ordinary steel leaf spring based on weight saving and specific modulus. The dimensions of 'Suzuki Mehran' car leaf spring were used to model the composites-based leaf spring. However, in laminated composites, the delamination is the major cause of failure under buckling load. The delamination in composite materials occurred due to thermal and chemical shrinkage, manufacturing inadequacies, eternal impacts and higher concentration in the zone of material discontinuities resulting in reduced strength of material leading to the structural failure. The experimental data of mechanical properties were obtained from published data on the composite materials and ANSYS was used for numerical analysis and for comparing the results.

1.2 Statement of the Problem

Ultimately with the utilization of traditional materials the life of structural components is much lesser than those of composite materials and can fail rapidly. The main aim of this work is to design and analyze the composite leaf spring for light weight vehicle.

1.3 Research Scope

The scope of this research aims to:

1. Study of leaf spring subjected to static and dynamic loading.
2. Model leaf spring using composite materials.
3. Finite Element Analysis of composite leaf spring subjected to static and dynamic loading. Determining the fatigue life of composite leaf springs.
4. Comparing analytical and numerical results.
5. Weight optimization of composite leaf spring design.

1.4 Research Objective

Based on the above research scope following research objective have been established.

1. Mathematical modelling of selected composite material.
2. To use composite materials with appropriate fiber orientation and stacking sequence.
3. To perform the static stress analysis on composite leaf spring.
4. To perform the dynamic analysis on composite leaf spring.
5. To optimize the weight of composite leaf spring.
6. To compare the conventional and composite leaf spring.
7. To use same geometrical properties as for steel leaf spring.

1.5 Research Methodology

The research methodology is the foremost part in any research. The research methodology contained analytical analysis, finite element modelling and numerical analysis. The finite element analysis on composite leaf springs were performed to obtain optimum light weight structure for composite leaf spring. For this objective, a correlation was established between materials selection, properties of materials, composite fiber direction, fiber orientation, lay-up sequence and the mechanical behavior of the spring. Therefore, with the help of these essentials a mono composite leaf spring has been designed and analyzed under given loading conditions using FEA method. Also, validations for various models has been presented which are used for the designing of mono composite leaf spring. The numerical and analytical analysis were performed. The numerical analysis includes static analysis, optimization and dynamic analysis.

The numerical analysis was conducted to speculate the behavior of a composite leaf spring under static and dynamic loading conditions. To determine the behavior of the spring under the gross weight of the vehicle static analysis was performed. However, dynamic analysis was performed to determine its behavior for irregular road conditions as depicted by ISO road classification[12] under various speeds. The mode shapes of the composite leaf spring were determined through modal analysis. Finally, the results of conventional and composite leaf spring were compared.

1.6 Thesis Overview

This thesis comprises of nine chapters. Chapter 1, presented the background of leaf spring. It also includes aims and objectives of the research. Also, the methodology of research was sentenced in this chapter.

Comprehensive literature review is presented in Chapter 2, in which studies of last decade was presented, which includes new design, fabrication, FEA, manufacturing, design improvements and experimental investigation of composite leaf spring. However, a limited literature and lack of information regarding the correlation between material properties, materials selection, lay-up sequence and composite design structure was drawn from the previous research works. The literature on conventional and composite leaf spring was reviewed in this chapter.

Chapter 3, discusses the detail description of the methodology of the research conducted and also discusses the research design and methods used to conduct this research. The Chapter ends with a discussion based on the analysis performed to obtain the results from the current study. In Chapter 4, the analytical analysis of conventional and composite leaf spring was presented.

Material properties, leaf spring design model and spring design parameters are given in chapter 5. Whereas, the mathematical modeling and validations of various finite element models that are used to design the composite leaf spring are also presented in Chapter 5 along with the comparison of numerical and analytical results.

Chapter 6 disclosed the finite element analysis that was conducted on conventional and composite leaf spring. The boundary conditions and meshing details are discussed here. In Chapter 7, dynamic analysis performed on composite leaf spring was presented. Road profile generation, characteristic of vehicle and generation of dynamic loads are also presented. Moreover, modal analysis was included in this chapter.

The Chapter 8 deals with the weight optimization of a composite leaf springs. The methods used for optimization were also discussed in this chapter. This chapter also provides the information about design variables, constraints and objective function.

Chapter 9 summarizes the results of the finite element analysis and weight optimization. The results are compared, discussed and concluded. This chapter also outlines the future work in this area of the field.

Chapter 2

Literature Review

2.1 Introduction

In past three decades there has been extensive research on leaf spring. This chapter provides a summary of steel leaf springs and composite-based leaf springs. The papers that are examined made a valuable contribution for the completion of this work. This chapter is divided in to two sections. The section 2.2 examines the earlier research on leaf spring. Section 2.3 presents gist of composite materials and their application in leaf spring suspension system.

2.2 Leaf Spring

Leaf spring is a kind of automobile suspension system often used to provide protection to the vehicle from shock loads due to road irregularities. The load due to irregular road pavement stored as potential energy in leaf springs due to spring deflection. The stored energy is relieved slowly after the load is removed. The spring carries brake torque, lateral loads and driving torque in addition to shock absorption. [8], [13], [14]. Thus, leaf springs have great advantage over helical spring as it provides protection to the frame as it ends directed along a fixed path as it bends. Additionally, it also acts as a structural member and store energy during operation.

2.2.1 Types of Leaf Springs

Although there are two kinds of leaf spring- single leaf and multi-leaf spring but they can be divided into following types depending upon their shape:

1. Transverse
2. Quarter-Elliptic
3. Three Quarter-Elliptic
4. Semi-Elliptic
5. Elliptic

The single leaf spring consists of single layer of leaf having tapered ends and thick cross-sectional area at the center of the spring. These leaf springs are usually used for the light weight automobiles and do not carry heavy loads. The multi-leaf spring comprises of more than one leaf. This kind of leaf spring is used in heavy weight vehicles and carries greater loads. In this type of springs, the leaves are coupled together with the help of rebound clips at intervals to prevent the leaves from separation. A large bolt at the center of the spring was used to maintain the distance of the leaves along the length and prevent it to move off center when load is applied.

In some cases, a rubber padding or some insulating material are often used between the leaves of the multi leaf spring to reduce wear and any other damage. The interleave friction often causes the damage of the contact surfaces between the leaves due to which the leaves tends to deform under small loading. Therefore, insulator is widely used between the leaves to prevent the leaves from wearing and squeaking by reducing friction between each leaf.

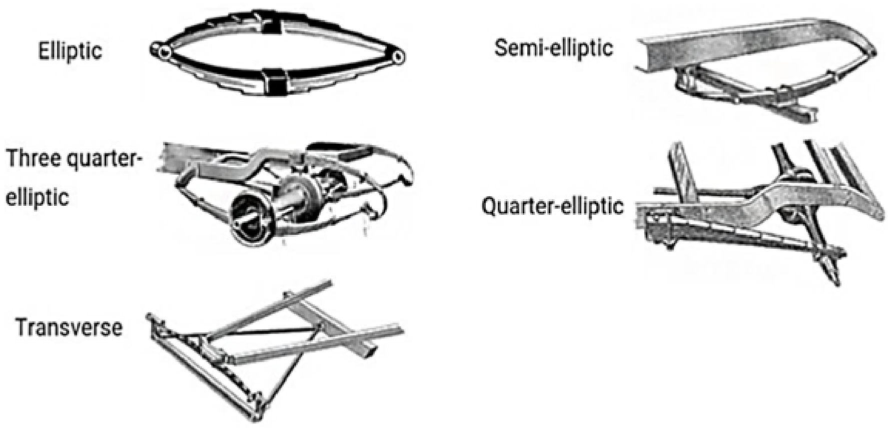


Figure 2.1: Types of Leaf Spring Based on Shape [15]

2.3 Conventional Leaf Spring

Although conventional leaf spring are the earliest form of the suspension system widely used in vehicles. It consists of an arc joined to the axel from the center while the ends are attached to the frame with help of pins. The front end is fixed with the frame of the automobile whereas the rear end is attached with help of link known as shackle to provide necessary motion [14]. Whenever, the wheel moves over an uneven road, the force is transferred from the road to the spring of the automobile through an axel. The spring deflects resulting the front eye of the spring to rotate about the pin while the rear eye of the spring translates and rotates about its axis.

In actual practice, irrespective of road irregularities vertical loads are assumed to be dominant due to which vibrations produced in vertical direction are absorbed by deviation of spring from its mean position under loading. Hence, potential energy is stored in leaf spring is released slowly when the loads are vanished. This potential energy in leaf spring is stored as strain energy. These springs possess high density and high modulus due to which the specific strain energy tends to decrease resulting in reduced strength and life. However, over the decades various types of metallic leaf spring were manufactured. The interleaf friction in these springs are greater due to which the springs tends to lose its shape and can sag. Also, the weight of the metallic leaf spring is greater, and it accounts for the unsprung weight.

Conventional leaf springs are usually used in the automotive industry and can be manufactured using different types of metallic materials including Chromium-Nickel- Molybdenum steel, Chromium vanadium steel, plain carbon steel and Silicon-manganese steel. Some metallic leaf spring material includes; SAE-1080, 6150-60, 1095, 9250-60 and 5155-50 [17],[18]. Parabolic leaf spring and conventional leaf spring including mono-leaf and multi-leaf springs are manufactured using these materials. However, in order to increase the hardness of the thick leaf spring the alloy content needs to be higher.

In this work the steel leaf spring used in Suzuki Mehran Car was investigated and analyzed to replace it with much robust material leaf spring. The leaf spring of Mehran car consists of three steel leaves, two of them are full length leaves while one of the leaves is slightly smaller in length from the rest of the leaves. These leaves are piled up together with the help of large bolt pass through the center of the leaves and a metal clips to reduce the interleaf friction. The whole assembly is connected to the axel with the help of U-bolts.

2.4 Composite Leaf Spring

The demand of composite materials in automotive industry have been intensified in contrast to metallic materials due to significant advantages of composites like high specific modulus, high specific strain energy density, low weight, unrivalled corrosion resistance, better internal damping and longer life [12][19]. Due to these characteristics, the composite materials for automotive structures can be used either as a direct replacement of conventional components or integration of conventional and composite components into one component or structure.

In automobile, one of the crucial components i.e leaf spring is a major concern for automotive industries to replace it with composite leaf spring. Though, in recent years many studies were conducted based on the application of composites for leaf springs. The major concern of automotive industries from last few decades is to enhance the strength of metallic leaf spring as it accounts for unsprung weight. Therefore, with the benefit of composite materials and its practice in manufacturing of leaf spring made it possible to reduce the unsprung weight and provide good quality ride as composite materials possess high specific strength, long life and reduced weight compared to metallic materials. Besides, the strength of the composite leaf spring depends on the direction in which the fibers are laid and its orientation.

2.4.1 Material Selection for Composite-based Leaf Spring

In this section, classes and properties of composite materials and their application in industries are focused. There are numerous ways to define the composite material, defined in ref [9] [20], [21], [22], [23].

Conventional engineering materials such as steel and aluminium carry contaminants that can depict different phases of the same material and provide the comprehensive definition of a composite though not recognized as composites mainly due to identical elastic properties and strength of pure and impurity phase. In this text, composites are considered as two or more materials of dissimilar chemical and physical characteristics are combined to extract a material having properties better than from its constituent materials. Composite materials are usually combination of matrix and reinforcement.

In acutal practice, composites are prepared using a matrix and fiber reinforcement to provide high performance. The type of reinforcements and matrices are shown in fig.1 and fig.2. Reinforcement provides a) length b) orientation c) shape [9],[21]. The fibers reinforcement increases the stiffness and provides high strength to the matrix. The matrix provides a) Compressive strength b) Transverse modulus c) Shear modulus d) High strength e) High coefficient of thermal expansion f) High thermal resistance [22].

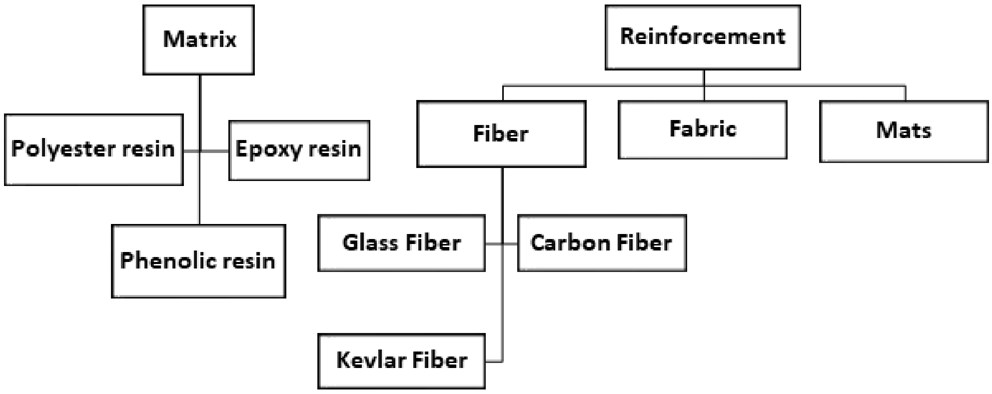


FIGURE 2.2: Different types of matrix and Reinforcement

2.4.2 Classification of Composites

The composite materials are of four types depending upon the type of the matrix used. These matrix includes; ceramic, polymer, carbon and metal matrix.. Different types of fiber materials can be mixed with a matrix to produce a variety of composite materials as given in Table 2.1. The strength of these composite materials depends upon the kind of the reinforcement and the matrix material. However, fiber volume fraction plays a critical role in controlling the strength of the material. Higher volume fraction of fiber increases the material properties. Moreover, fiber orientation and number of ply's is important concern in determining the strength of the material. The composite materials can be altered for different applications depending upon the load to be carried e.g. in case of longitudinal loading, the fiber oriented at 0 are enough to sustain the loading of the structure.But the maximum load applied should not exceed the tensile and compression strength.

TABLE 2.1: Types of composite materials [26]

Matrix	**Polymer**	**Ceramics**	**Metal**	**Carbon**
Polymer	✓	✓	✓	✓
Ceramics	✓	✓	✓	✓
Metal	✓	✓	✓	✓
Carbon	✓	✓	✓	✓
Ceramics		✓	✓	✓

Polymer Matrix Composites

They are essentially used in nearly low-temperature applications. It includes

1. Thermoplastic resins: polysulfone and poly-ether-ether-ketone
2. thermoset resins: polyester, polyimide and epoxy.
3. Reinforcements: Glass, boron, aramid (Kevlar) and carbon (graphite) fibers.

Metal Matrix Composites These composites possess high chemical inertness, high temperature stability and good shear strength. It consists of

1. metals or alloys: titanium, aluminum, copper and magnesium.
2. reinforcement: carbon (graphite), boron and ceramic fibers.

Ceramic Matrix Composites

They are extremely adapted for very high temperature applications. It consists of

1. ceramic matrices: glass-ceramic, silicon nitride, aluminum oxide, glass-ceramic, silicon carbide
2. reinforcement: ceramic fibers.

Carbon/Carbon Composites

They possess low thermal expansion, low density and can withstand relatively high operating temperatures. Their strength increases with rise in temperature [23]. It Its constituents are:

1. carbon or graphite matrix
2. reinforcement: graphite yarn or fabric.

Composites are also classified according to their structure and orientation of fibers. These includes the following:

1. Fibrous composites

   It incorporates group of long fiber or whiskers arranged within a matrix either in previously defined angle or in arbitrary angles at a definite temperature range. The fiber orientation can be continuous or discontinuous, straight or woven, shown in Fig 2.3.

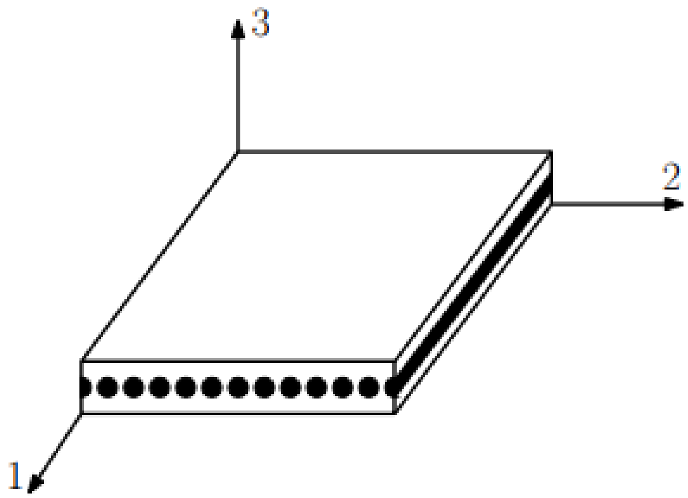
   

   FIGURE 2.3: Longitudinal fibers in a lamina [27]

   The disparity among whiskers and fibers can be characterized by their geometry. Though, fibers are approximately crystal-sized diameter with high length-to-diameter ratio whereas whisker exhibits same diameter as fibers with length-to-diameter ratio ranging up to hundreds. Generally, whiskers are short and thick perfectly aligned crystals and exhibits preferably higher properties than fibers.[24]

2. Laminated composites:

   It consists of two or more lamina bonded together and exhibits higher mechanical properties. However, the fiber orientation in laminate predicts the mechanical properties of composite [28].

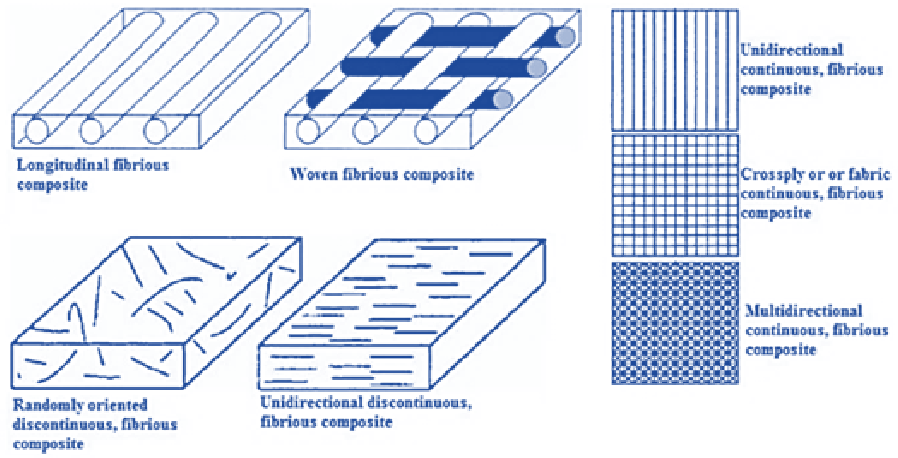


FIGURE 2.4: Classification of fibrous composites[29]

Laminated composites are in various forms, each exhibit different mechanical properties. Among them, sandwich structural form is one of them with higher mechanical properties. In this type of arrangement, thick core or lamina are sandwich between a thin facing. The facing provides resistance to the in-plane loads, bending moments and provide bending rigidity to the structure. It is also called as stressed skin construction. The core itself provides rigidity to the structure, separates the facings and spreads shear between them. This helps the structure to be effective at common neutral axis.

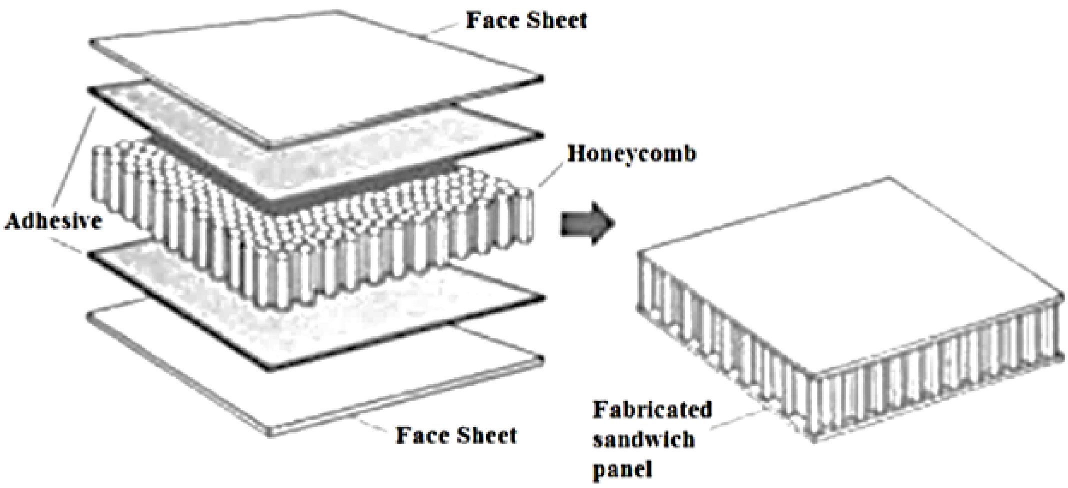


FIGURE 2.5: Honeycomb composite structure[29]

3. Particulate composites

These composites are obtained by combining particles and matrices together. These matrices include alloys and ceramics. These types of composites can be grouped into following arrangements.

1. Metallic particles in metallic matrix
2. Nonmetallic particles in nonmetallic matrix
3. Nonmetallic particles in metallic matrix
4. Metallic particles in nonmetallic matrix

These composites are isotropic in nature and have increased operating temperature, improved strength and increased oxidation resistance. Three types of such composites are shown in Fig.2.6

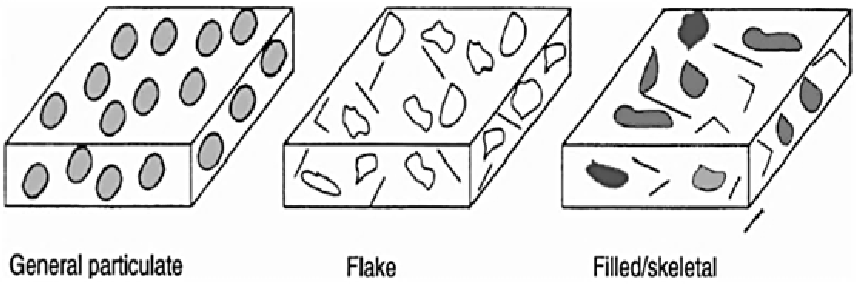


FIGURE 2.6: Three type of particulate composites [24]

Flake composite consist of flakes suspended in matrix. Though flakes provide large area to thickness ratio. Filled/Skeletal type of composite comprise of constant skeletal matrix occupied by additional material. Example includes additional insulting material in honeycomb core.

2.4.3 Properties of Composites

Properties of composites mainly depend upon:

1. Properties of constituent's materials
2. Interaction
3. Geometrical distribution and orientation

Advantages of Composite Materials

Composites are mainly preferred over natural and conventional materials due to two main benefits offered by composites i.e.high strength to weight ratio and height stiffness to weight ratio. Comparison of conventional and composite material is given in Table-2.2.

$$SpecificModulus = E/\rho$$

$$SpecificStrength = \sigma/\rho$$

TABLE 2.2: Comparison of Conventional and Composite material

Material	ρ Kg/m^3	σ **MPa**	E **GPa**	σ/ρ **J/Kg**	E/ρ **N-m/Kg**
Steel	7800	2100	200	2.7	2.7
Glass Fiber	2540	3450	85	13.6	3.3
Kevlar Fiber	1440	3600	124	2.5	8.6

Composites provide better advantages as compared to conventional materials such as weight reduction, cost effective, low manufacturing complexity, reduced tooling cost, unrivalled corrosion resistance, better internal damping etc [22]. Properties and strength of composites vary significantly on variation in the orientation of the fiber placement.

The use of fiber reinforced composites in the automobile industry can be grouped into three i.e. body, chassis and engine parts. Exterior parts need high stiffness, strength against the dent and high finishing. The study indicated that with increase in weight fraction of reinforcement, the flexural strength and tensile strength increased by 123.65% and 14.5% for 20% glass reinforced composites over pure epoxy. The increased reinforcement increases the brittleness that may result in low impact strength [30]. Tensile strength is mainly dependent on the thickness and fiber orientation of laminated composites [31]. The best modulus of elasticity in longitudinal direction for reinforcement composite is unidirectional fiber types and for transverse direction, the woven reinforcement fiber types [32]. One of the studies revealed that the mechanical properties depends upon the strain rate. E-glass/Epoxy has better mechanical properties with increased strain rate and fit for structural vehicles body panels. Flexural strength of laminated composites depends on the type of resin used and the thickness [33], [28]. The tensile strength of the composites increases with strain rate. Shear strength increased by 7.06%, tensile modulus increased by 9.3%, shear modulus increased by 11.06% and tensile modulus increased by 1.82% [34]. Literature review helped to select Kevlar/Epoxy and Glass/Epoxy as the most suitable combination of composite materials for leaf spring [13], [22], [35], [36]. Some of the researches that contribute for this work are discussed below.

M.M.Shokrieh et.al [37] worked on the design and optimization of four steel leaf spring. The focus of this study was to replace the conventional leaf spring by optimized composite leaf spring having minimum weight and can withstand peripheral loading without failure. The fiberglass with epoxy resin was used for the composite leaf spring for the analysis. The analysis was performed in ANSYS V5.4 software. Using FEA analysis the numerical results obtained for stresses and deflection are compared and verified with existing analytical and experimental results. The attention was given to the optimization of the leaf spring design. The design constraint are stress and deflection. The results obtained after optimization of leaf spring showed that the stresses are much lower in composite leaf spring as compared to steel leaf spring and weight saving of approximately 80% was obtained for composite leaf spring without eyes. Also, the natural frequencies obtained for composite leaf spring are greater than road frequency and steel leaf spring to overcome failure due to resonance.

E. Mahdi a et al. [38] worked on the composite elliptic leaf springs for light weight vehicles. The study was based on the ellipticity ratio of the composite leaf spring. The elliptical configuration for woven roving composites was considered. This paper studied the impact of using ellipticity ratios extending from one to two on the performance of elliptical leaf spring made from woven roving wrapped composite. The results are examined both numerically and experimentally. The failure mechanisms for these leaf springs were also presented and cited that increasing the wall thickness leads to the increase in the maximum failure and spring rate (k). The results presented that ellipticity ratio has significant effect on the failure loads and spring rate, also the ellipticity ratio (a/b) of 2 for elliptic composite leaf spring exhibited the highest spring rate and demonstrated that these leaf spring can be used for both light and heavy weight vehicles up to significant weight saving and can meet the requirements.

Pankaj Saini et al.[39] presented the work on the design and analysis of composite leaf spring for light vehicles. The objective of this research is to replace the conventional steel leaf spring by composite leaf spring and compare their stresses and weight. In this work, three different type of materials are selected for the composite leaf spring model i.e. E-glass/epoxy, graphite epoxy and carbon epoxy. The static analysis was conducted in ANSYS and results are obtained for the stresses. The stresses for steel leaf spring are higher than composite leaf springs. However, the stresses in graphite/epoxy leaf spring are higher than steel leaf spring among other composite leaf spring. The results demonstrated that E-glass/Epoxy composite leaf spring saves weight up to 81.22% , Graphite/Epoxy saves up to 91.90% and Carbon/Epoxy saves up to 90.50% as compared to steel leaf spring. Also, it was observed that E-glass/Epoxy composite leaf spring has lower stresses and weight as compared to steel leaf spring. Hence, it is considered as replacement of steel leaf spring.

Y.N.V Santhosh Kumar et al.[40] studied the design and analysis of composite leaf spring. The main goal of this work is to analyze the Mono composite leaf spring having minimum weight to replace the conventional steel leaf spring. The material used for this purpose is E-glass/Epoxy composite. The leaf spring was analyzed in ANSYS Multiphysics. The weight obtained using E-glass/Epoxy composite leaf spring was 39.4% to that of steel leaf spring. So, a weight saving of 60.48% was achieved. From the results it was obtained that the stresses in composite leaf spring are lower than steel leaf spring and within allowable limits. It was also observed that orientation of fibers in perpendicular direction in composite laminate offered increased strength than other fiber orientations.

Danish Khan et al. [41] optimized the material selection for leaf spring. For this purpose, conventional materials and different alternate materials characteristics are studied. The spring was modelled using Creo and material properties of different conventional materials are assigned and analyzed using Ansys. The material used for this work are structural steel, aluminum alloy, copper alloy, stainless steel and titanium alloy. The results are obtained for five parameters including stresses, strains, deformation and mass for the materials listed above. The results showed that aluminum alloy found to be optimum material based on the results of five parameters listed before.

J. Melcer [42] presented the work on the vehicle road interaction in frequency domain. This work investigated one-dimensional quarter car model. The car model was derived mathematically in terms of frequency domain as a function of angular frequency to derive the power quantities. The frequency response of the system is evaluated numerically with frequency ranging from 0 to 100 rad/s. Since, the frequency functions are dynamic characteristics of the system thus can be used to assess the dynamic behavior in terms of frequency domain and also express the sensitivity of the system. Also, the critical wavelengths at periodic repetitions due to road unevenness were calculated at critical speeds.

M. Agostinacchio et al.[43] evaluated the dynamic response of different vehicles by the generation of the road profiles depicted according to the standard 1SO

8608. The Quarter car model was implemented to analyze the dynamic response at different speeds ranging from 20 to 100 km/h. The road profile was generated in software and analytical relations for quarter car model for three different vehicles were derived and analysed. The results showed that different vehicles having different masses respond unlikely. The vehicle with large mass produces dynamic overload and thus causes the vibrational stress. The road profile of 250 m long pavement with different road profiles were generated and showed varying effect on vehicle characteristics. This means that as speed and road profile changes, the dynamic load varies accordingly.

Mayuri A. Chaudhari et al.[44] presented the work on the conventional leaf spring suspension system of a heavy weight vehicle. This works aims to perform the numerical analysis on leaf spring assembly that consist of 8 leaves. The leaf spring was designed and analyzed numerically in Ansys for the load of 32.833KN. The results obtained from analytical, numerical and experimental method are compared. It was found that there is 4% and 14% variation in deflection obtained comparing the numerical method with analytical and experimental method respectively. Also, the percentage variation of 12.35% is obtained for stresses considering numerical method and analytical method while 9% variation is obtained considering numerical method and experimental method.

Hayder H. Khaleel et al.[45] presented the work on the two different types of spring which are parabolic and multi leaf spring which was demonstrated using steel and carbon compound. The springs were demonstrated in solid works and numerical analysis was performed in ANSYS to determine shear stress, total deformation, static stress and strain exposed to 5000 N and 10000 N load. The results found shows that composite multi-leaf spring have improved capacity to carry load and sustain deformation better than parabolic leaf spring. Also, the asset of carbon composite multi-leaf spring owns greater strength and dainty weight.

Erol Sancaktar et al.[46] contributed to design and manufacture the unidirectional E-glass/epoxy composite spring. The first concern of this work is to provide higher understanding of the composite spring, its method of manufacturing and characteristics. Here, the load is the primary concern of this work. The spring was modelled and analyzed in ANSYS.Also, the spring was fabricated by hand lay-up technique and a mass of 1179.4 g was obtained.

Kat J.[47] introduced approval of leaf spring model. So as to perform powerful and dependable re-enactments in the CAE procedure, exact recreation models of the vehicle and its related frameworks, subsystems and segments are required. In the vehicle elements setting recreation models of the tires, suspension, springs, damper, and so forth are required. This investigation will take a glimpse at designing an approved model of a leaf spring suspension framework utilized on business vehicles. The essential objective was to model the spring assembly to predict the forces at the point of attachments with the frame of the vehicle. The part which will get a lot of consideration in this examination is the leaf spring. Despite the fact that leaf springs are every now and again utilized practically speaking they despite everything hold incredible difficulties in making exact numerical models. It is obviously that a precise model of a leaf spring is required if exact full vehicle models are to be made.

The natural frequency of composite leaf spring is higher than steel leaf spring[37].In one of the studies it was concluded that composite leaf spring have 67.35% less stress,64.95% more stiffness and 126.98% higher natural frequency as compared to steel leaf spring [48]. Another study established that in composite material leaf spring, the deflection is reduced by 6.51% , bending strength decreased by 83.64% and material saving is 71.85% by weight in comparison to steel leaf spring [49]. In composite leaf spring maximum bending stress and deflection is lower and fatigue life of E-glass/epoxy or E-glass/vinyl ester leaf spring was 2 and 4 times higher than that of steel leaf spring [50]. Composite leaf spring have better fatigue behaviour than steel leaf spring and hybridization technique can be used to reduce weight and better performance in auto industry[17]. For composite leaf spring a reduction of shear stresses in eye-end design was noticed. The composite material is having compressive load resistance problem, but it may be avoided by using carbon fibers [51],[52]. Also, under same loading conditions, there is a great difference in the deflection and stresses of steel and composite leaf spring. Deflection of composite leaf spring is less as compared to steel leaf spring [53].

Leaf spring manufactured from composites not only reduced weight but also offers increased strength to weight ratio. And it is also observed that strain energy increases because composite materials provide necessary material properties having minimum modulus of elasticity and maximum strength in the fiber direction [54], [55]. It was demonstrated analytically and numerically that the fatigue analysis for leaf spring was carried out for mono leaf spring and presented that these leaf springs are not commonly used and more research can be presented further. [14],[35].

The fatigue analysis of the composite leaf spring using glass fiber reinforced with polymer illustrated that it has less stress, more stiffness and natural frequency than that of steel leaf spring. The fatigue life of composite leaf spring is predicted to be more as compared to the steel leaf spring [56].

The literature of past decade was analyzed and incorporated in this thesis. In previous researches, the composite leaf spring was analyzed under static loading conditions only. Some of the authors presented the numerical modal analysis of the spring while some of them worked on the experimental setup to determine the strength of composite and steel leaf spring. However, in present work, the leaf spring of Suzuki Mehran car was chosen for the analysis before which none of the research work was conducted. But this work can be extended for other vehicles having same load carrying capacity as that of Suzuki Mehran. Furthermore, all the possible analysis was summed up in this work including; Static analysis, dynamic analysis, fatigue life analysis, modal analysis and weight optimization of composite leaf spring and results were compared with conventional leaf spring.

Chapter 3

Research Methodology

Suitable methodology and solution development for any problem is the most critical part of a design and analysis-based project. In this Chapter the description of the research process is presented. This Chapter discusses the research design, selection of material properties and methods used to conduct this research. The Chapter ends with a discussion based on the analysis performed to obtain the results from the current study.

3.1 Introduction

As far as this research is concerned, the leaf spring suspension of light weight vehicle was taken for the analysis. The leaf spring suspension system had the parabolic design and material properties of steel are considered as the primary variables which could be altered to enhance stiffness to weight ratio. The weight reduction, flexibility and enhance stiffness properties of leaf spring design would be the major concerns of this research. In this research the leaf spring design of Suzuki Mehran car was taken into consideration. The conventional material used for the manufacturing of this leaf spring was 51CrV4 steel. However, the alternate material was chosen for leaf spring to conduct this research were two composite materials. These two materials were selected based on their enhanced mechanical properties as compared to conventional material. This research aims to conduct analytical analysis and numerical analysis on conventional and composite leaf spring. The analytical analysis of leaf spring involves the mathematical modelling and hand calculations while the numerical analysis involves static analysis, dynamic analysis, fatigue analysis and weight optimization using FEA software. The methodology followed is presented below.

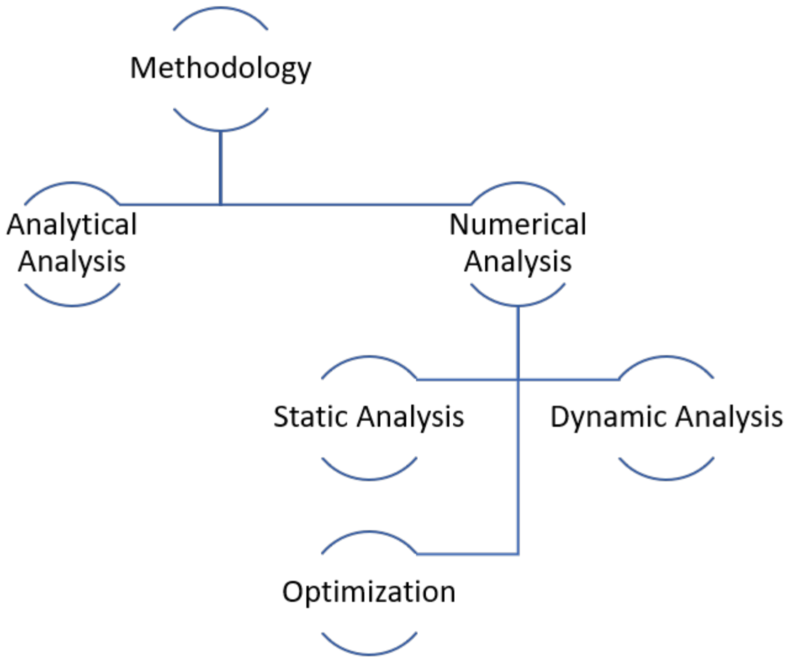


FIGURE 3.1: Sequence of the Work

3.1.1 Analytical Analysis

The analytical method is a critical approach to solve the real-life problems in an adequate manner. This method is based on mathematical model and mathematical relations that are derived from physical model. In the present study the mathematical relations are derived for leaf spring based on beam theory to predict the real-time behavior of the leaf spring as it works alike a double cantilever beam. A leaf spring is a structural member used to carry shear loads, side loads, vertical loads and bending loads.

However, the beam theory is an important concern in designing of leaf spring due to definite geometry and design principal of leaf springs. The beam theory is based on the Euler-Bernoulli assumptions and is important concern in structural mechanics because of its extensive applications in practical problems. Therefore, the beam theory was used to develop mathematical relations for leaf spring to determine bending stresses and deflection in conventional leaf spring whereas analytical relations based on laminate theory were used to determine stresses and Tsai-Wu failure theory was used to determine the failure stress in composite leaf spring. The stress concentration induced depends on the design of the spring, mechanical properties and material of the spring. Hence, it is important to compare the consequences of the materials used and to validate the strength of the model by incorporating the above-mentioned parameters.

The analytical analysis was used to find out the maximum stress in a leaf spring subjected to static and dynamic loads. The deflection of the spring determined analytically under maximum loading was in limit of the camber that was obtained from actual data of leaf spring. However, the analytical analysis based on static loads and dynamic loads verifies the physical model of the conventional leaf spring by comparing the results obtained from analytical and numerical analysis.

3.1.2 Numerical Analysis

The foremost step in numerical analysis is a parametric modeling of the physical model. The CREO software is the comprehensive CAD system developed by parametric technology. This system includes the characteristics of mechanical design, analysis and manufacturing. Hence, both composite and conventional leaf springs were created in CAD system using design parameters of actual leaf spring of vehicle being analyzed. The model was then subjected to finite element method by using Ansys software to determine the stresses and deformation induced in the model to predict the behavior under different loading conditions. The Structural analysis is the widespread application of finite element engineering practices; it includes bridges, buildings and variety of other similar structures. Other applications include ship hulls, machine housings, aircraft bodies, mechanical pistons, tools and variety of machine parts.

In structural analysis, displacements are calculated as the primary unknowns due to which other quantities like reaction forces, stress and strain, all are evaluated from the nodal displacements. There are various types of structural analyses that are available for FEA analysis in Ansys. However, some of the products that are used for this work are discussed below.

Static Analysis includes both type of static analysis under static loading conditions i.e. nonlinear and linear and are used to find out quantities such as displacements and stresses etc. Other quantities include stress stiffening, plasticity, large strain, creep, large deflection, contact surfaces and hyper elasticity. To evaluate natural frequencies and mode shapes structures modal analysis is used. Moreover, in order to compute large deformation and complex contact problems, explicit dynamic analysis is used by using explicit solver.

In finite element analysis, it is mandatory to validate the simulation model to ensure that it is not introducing numerical errors itself. The validation verifies that; the numerical assumptions are consistent i.e. it is equivalent to the true response of the system, the material behavior is properly addressed, the physics of the model is ensured appropriately, the boundary conditions are applied properly. Therefore, before performing numerical analysis on leaf spring, some of the validations were conducted to ensure that all the parameters that were used are properly addressed and applied as a standardized process. However, the FEA that was performed in present study involves the static analysis, dynamic analysis, modal analysis, fatigue analysis and optimization are discussed here.

3.1.2.1 Static Analysis

The static analysis involves the relationship among the response of the body under applied load without changing with time while ignoring the inertial and damping effects. The static analysis is used to perform finite element analysis which generally involves preprocessing, solution and post processing. This method is used in the present work to predict the behavior of the conventional and composite leaf spring under different loads applied.

Initially, the design of the conventional leaf spring of Mehran car was modelled in CAD software using actual parameters of the spring and imported in FEA software. The properties of material are assigned, element type was chosen, contact type was defined, and mesh of the model was created. The actual boundary conditions were used for the leaf spring to solve the model whereas stresses and deformations are obtained as a result of the static analysis. The same method was adopted for composite leaf springs but after creating the mesh, the plies were created for the model according to the desired stacking sequence based on the fiber orientations. The model was than solved and results are obtained for stresses and displacements for each ply. Moreover, the failure theory was also applied and failure stresses in each ply was also evaluated to determine the maximum stress that each ply can bear before failure. This is the major concern in predicting the strength of the leaf spring.

3.1.2.2 Dynamic Analysis

The response of the body under time-varying load can be defined as dynamic analysis. In this analysis the load applied on the body is varying with time while considering the inertial and damping effects. The present work encompasses the dynamic behavior of the leaf spring under dynamic loads which are generated due to road irregularities. Firstly, dynamic loads against ISO standard of road profile for class C-D are calculated and then applied on the leaf spring to evaluate the dynamic stresses and deformations induced. The boundary conditions are same as actual leaf spring, but loads are varying with time. This procedure is adopted for conventional and composite leaf spring and numerical results are compared with analytical results.

The present work was focused on the QCM (Quarter Car Model) which was used efficiently to investigate the dynamic interaction between the road profile and the vehicle. The QCM includes mass, damper and a spring. The load produced by the road profile on the axle from the wheel is not constant in space and time and can depend upon the speed, mass of the vehicle and type of the suspension. Therefore, the road profile data was used to evaluate the dynamic loads at various speeds. Hence, the stresses and deformation produced under these loads are evaluated.

3.1.2.3 Modal Analysis

The method of determining the dynamic properties of a structure under excitation by vibrational loads is termed as modal analysis. These dynamic characteristics are the inherent properties of the system and are in the form of damping ratio, natural frequency and mode shapes. The mode of the system depends upon the properties of material, geometry of the body and the applied boundary conditions. Consequently, changing the above parameters, the modes of the system will change ultimately. However, adding mass to the structure will change its natural frequency. Thus, conventional and composite leaf spring are designed with their respective material properties and same boundary conditions are applied to determine the natural frequencies of the springs and their mode shapes. The mode shape is dependent on the excitation frequency and the shape of the body at this frequency. Though, changing the excitation frequency will change the mode shape of the body.

3.1.2.4 Optimization

The demand of greater strength and reduced cost of production in automotive industry has exaggerated designers to think in three dimensions to develop rigorous techniques and methods such as “optimization” to increase the systems performance. The process of obtaining the effective and favorable conditions to achieve the optimal design under the set of prioritized constraints is known as optimization. There are different methods of optimization which can be used to maximize factors such as longevity, strength, efficiency, reliability, productivity, and utilization.

The different methods of optimization are widely used in engineering to solve various optimization problems to provide optimum solution. With the intensive improvement in technology, the optimization problems can be solved effortlessly by means of computer aided techniques. However, the optimization of the engineering structures can be done by using numerical procedure known as finite element method. The finite element method may also be used to perform the structural optimization. There are various methods in finite element optimization including first order optimization method, second order method etc. to solve various design problems depending upon the design optimization criteria.

This research employs the first order optimization technique to increase the strength and reduce the weight of the composite leaf spring. The key elements involve in optimization problem are design objective, design variables and design constraints. In this work, the objective function is to minimize the weight of the spring. The design variable are thickness and width while design constraints are stresses and deflection. However, in the present case the static analysis was carried out on the composite leaf spring to determine the stresses and deflections as a design constraint. The optimization followed by first order method was employed and best candidate point was selected from a set of design candidate points.

Chapter 4

Analytical Analysis

4.1 Introduction

In any finite element problem, the foremost step is the analytical analysis of the design to be analyzed. The analytical method involves mathematical modeling and mathematical relations to solve the real-life problems in adequate manner. To perform the analytical analysis on leaf spring, the mathematical relations are derived for steel spring based on beam theory. However, analytical relations based on laminate theory were used to determine stresses and Tsai-Wu failure theory was used to determine the failure stress in composite leaf spring. In this chapter mathematical modelling of conventional and composite leaf spring is presented. This chapter discusses the detailed mathematical relations derived from cantilever beam theory and laminate theory. The beam theory and laminate theory are used to determine total deflection and maximum bending stress in steel and composites leaf spring under same loading conditions.

4.2 Mathematical Modelling

Leaf springs are assumed as a cantilever beam if symmetry exist at the center of the spring[57]. On the other hand, it may be assumed as simply supported beam.

Consider a beam is fixed at its one end while the other end of the beam is carrying a load. The basic relation to find bending stress and deflection for the cantilever beam is given by:

$$\sigma = \frac{My}{I} \tag{4.1}$$

Where, bending stress is represented by "", the moment of interia is represented by "I", bending moment is "M" and the distance from surface to neutral axis is "y".The bending moment of spring is defined as load times length, the moment of inertia of spring having rectangular cross-section area is defined as 1/12 bt^3 and distance from surface to neutral axis is defined as "t/2". The above equation for bending stress and deflection in cantilever beam is now given by Eq 4.2 and Eq 4.3.

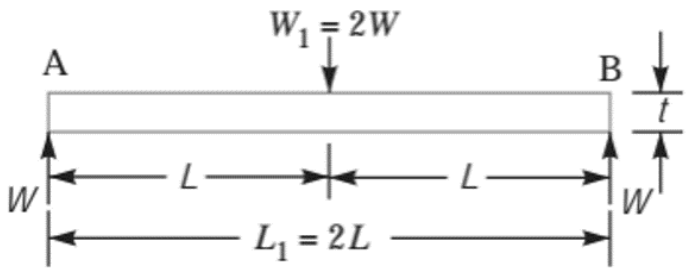


FIGURE 4.1: Simply supported beam (Flat leaf spring) [57]

$$\sigma = \frac{6WL}{bt^2} \tag{4.2}$$

$$\delta = \frac{WL^3}{3EI} \tag{4.3}$$

Where, "W" is the load, "t" is the thickness of beam, "b" is the width of beam, "L" is the length of beam, "E" is the modulus of material. If spring is simply supported beam, having length 2L and load 2W at its center, as shown in Fig. 4.1, then in above Eq 4.2 and Eq 4.3, the load W is replaced by W1=2W and length L is replaced by L1=2L. For "n" number of leaves the Eq 4.4 and Eq 4.5 are now given by

$$\sigma = \frac{6WL}{nbt^2} \tag{4.4}$$

$$\delta = \frac{WL^3}{n3EI} \tag{4.5}$$

In cantilever beam, the bending moment is max at the fixed end and in the simply supported beam the Bending moment is max at the center. To counter this bending moment, the height of leaf spring is maximum at the center, which shows that it is a simply supported beam. So the formulation of stress and deflection was based on simply supported beam theory. To overcome this problem, the cross-section area of the beam should be kept constant by varying thickness and width to provide it a uniform strength and to keep stresses low. Moreover, the bending moment causes tension in top surface fibers and compression in bottom surface fibers, whereas shear stress is maximum at the center and zero at the ends, as shown in Fig.4.2. Therefore, more than one leaves of uniform cross-section area are necessary to overcome the shear forces. In actual practice it is necessary for leaf spring to have two full length leaves and others are graduated leaves to provide stability and smooth suspension to the vehicle.

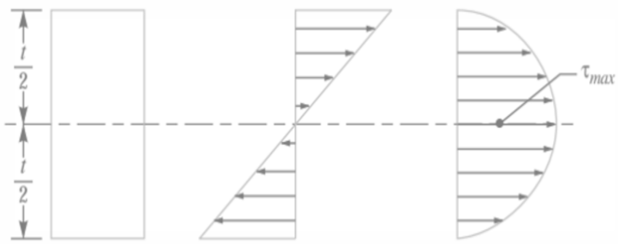


FIGURE 4.2: Bending and Shear stress in cross-section plate [58]

The two full length leaves should provide necessary strength to the spring as they have to withstand side loads, vertical loads and twisting loads. The derived relations of bending stress and deflection in case of full and graduated leaf spring are given below:

$$\sigma_F = \frac{18WL}{bt^2(2n_G + 3n_F)} \tag{4.6}$$

$$\sigma_G = \frac{12WL}{bt^2(2n_G + 3n_F)} \tag{4.7}$$

$$\sigma = \frac{12WL^3}{Ebt^3(2n_G + 3n_F)} \tag{4.8}$$

Where, n_G represents number of graduated leaves, "n_F" represents number of full-length leaves. In case of curved leaf spring having constant radius of curvature, the stresses and deflection can be found using the following equations [59]:

$$\sigma = \frac{MY_T}{NA(R - Y_T)} \tag{4.9}$$

$$\delta = \frac{2WL^3}{3Ebt^3} \tag{4.10}$$

Where, "M" is the moment, "Y_T" is the radius of curvature, "N" is the distance between axis of symmetry and neutral axis, "A" is the area and "R" is the distance from center to the neutral axis.

4.2.1 Mathematical Modelling of Steel Leaf Spring

The leaf spring of "Suzuki Mehran" car was chosen for the analysis. The basic data used for the modelling of steel leaf spring is given in Table 4.1. The maximum load carrying capacity of leaf spring is determined by the data provided by "Suzuki motors". The curb weight of the vehicle is 620 Kg. The front axle gross weight and rear axle gross weight is 480 Kg and 500 Kg respectively. So, the gross weight of the vehicle is 980 kg which means that the maximum load carrying capacity of two semi-elliptic leaf spring used in Suzuki Mehran is 9800 N. So, load acting on one leaf spring assembly is 4900 N.

The Calculation of the load and effective length of leaf spring is calculated by the consideration of cantilever beam. Suppose leaf spring acts as a two cantilever beams because loads are acted at two ends of the spring and fixed at the center. So,

FIGURE 4.3: Actual leaf spring installed in Mehran

the load acting on half length of the spring is found to be 2450 N. The ineffective length of the spring is the portion of the spring which is unbent. The unbent portion is due to support and clamps i.e. "U bolts". The calculated distance between two "U bolts" used is 96 mm. So, the effective length of the spring is calculated to be 450.6 mm. The stress generated in steel leaf spring is calculated by considering the factor of safety of 1.2 for the leaf spring. The allowable tensile strength "Σt" and allowable yield strength "Σ_y" for the leaf spring are calculated to be 1125 N/mm^2 and 1000 N/mm^2 respectively. Moreover, bending stress in leaf spring was found to be 965.2 N/mm^2 using the following relation.

$$\sigma = \frac{6WL}{nbt^2} \tag{4.11}$$

Since the value of bending stress obtained is less as compared to allowable stress of the material so design is safe. The leaf spring deflection was calculated to be 114.69 mm by using the following relation as:

$$\sigma = \frac{4WL^3}{nEbt^3} \tag{4.12}$$

Since, the deflection produced in leaf spring is less than its camber i.e. 127 mm. Therefore, the spring is in safe limit under maximum static loading.

TABLE 4.1: Basic data of Suzuki Mehran leaf spring

Parameters	**Dimensions**
Material	50CrV4
Total length of the spring	965.2 mm
Length of front half	520.7 mm
Length of rear half	444.5mm
No of full-length leaves (nf)	02
No of graduated leaves (ng)	01
Radius of curvature (R)	980.44mm
Thickness of leaf (t)	7mm
Width of the leaf	50mm
Young's Modulus (E)	2.1x105 N/mm2
Tensile Strength (σt)	1350-1650 MPa
Yield Strength (σy)	>1200 MPa
Poisson ratio	0.3
Density	7800 Kg/m^3
Thermal Conductivity ambient temperature	40-45 W/m^0K
Specific Heat capacity 50/10 ^{o} C	460-480 J/Kg^oK
Central Band width (ineffective length)	80 mm
Rebound Clip width	20 mm
Eye Internal diameter (Front)	40 mm
Eye Internal diameter (Rear)	29 mm
Weight	65 Kg
Spring rate	3.2673 N/m

4.3 Mathematical Modelling of Composite Leaf Spring

The mathematical relations used to determine the bending stress and deflection in composite leaf spring are different from steel leaf spring. In conventional leaf spring, the whole spring is made up of uniform steel material whereas in composite leaf spring, the material is piled up in the form of laminates. The properties of the material are defined by its laminate. The laminate consists of plies bonded to each other. Each ply can be defined by its material, fiber orientation and location from reference axis. The material and fiber orientation in a laminate may be different from other piles resulting in increased material properties.

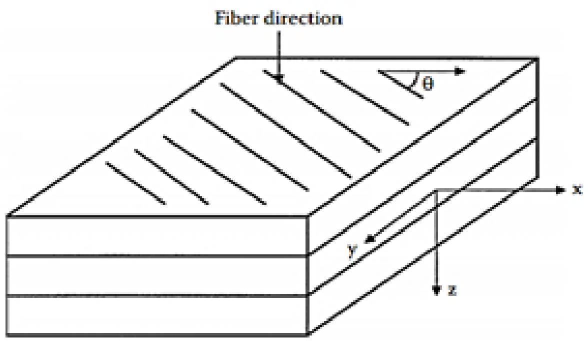


FIGURE 4.4: Schematic of Laminate[60]

In automotive industries the leaf spring are manufactured from different types of composite materials including graphite epoxy, Kevlar epoxy, carbon epoxy and E-glass epoxy etc. The strength of these materials depends upon the properties of the laminates. Carbon fibers and E-glass fibers have low tensile strength as compared to kevlar fibers. However, the specific strain energy density of materials helps in selecting the optimal material for leaf springs [37]. The figure shows the comparison of these materials based on specific strain energy density which was calculated from fatigue strength for dynamic loading using the relation given by [61]:

$$s = \frac{1}{2}\frac{\sigma^2}{\rho E} \tag{4.13}$$

Where 'S' is the specific strain energy, '' is the allowable stress, '' is the density of the material and 'E' is the modulus of elasticity of the material. It was evident from the Fig.12 that HT-carbon/epoxy has high specific strain energy then rest of the materials due to which it possesses high stiffness, strength and low weight. These properties make it a good choice for designing of leaf spring. But it has low impact strength and corrosion resistance and also the cost of this material is high as compared to other composites due to which this material can be neglected.

The density of Kevlar fiber is low resulting in significant weight reduction compared to other materials. It was also observed from figure that Kevlar/epoxy possesses good specific strain energy capacity, low cost, low weight and high strength made it suitable for structures which are weight efficient and require high strength. Kevlar composite with compact fiber orientation and lay-up made the material more robust and vigorous. On the other hand, the specific strain energy density and strength of S2-glass fibers are higher than E-glass fiber but cost is beyond the cost optimal design.

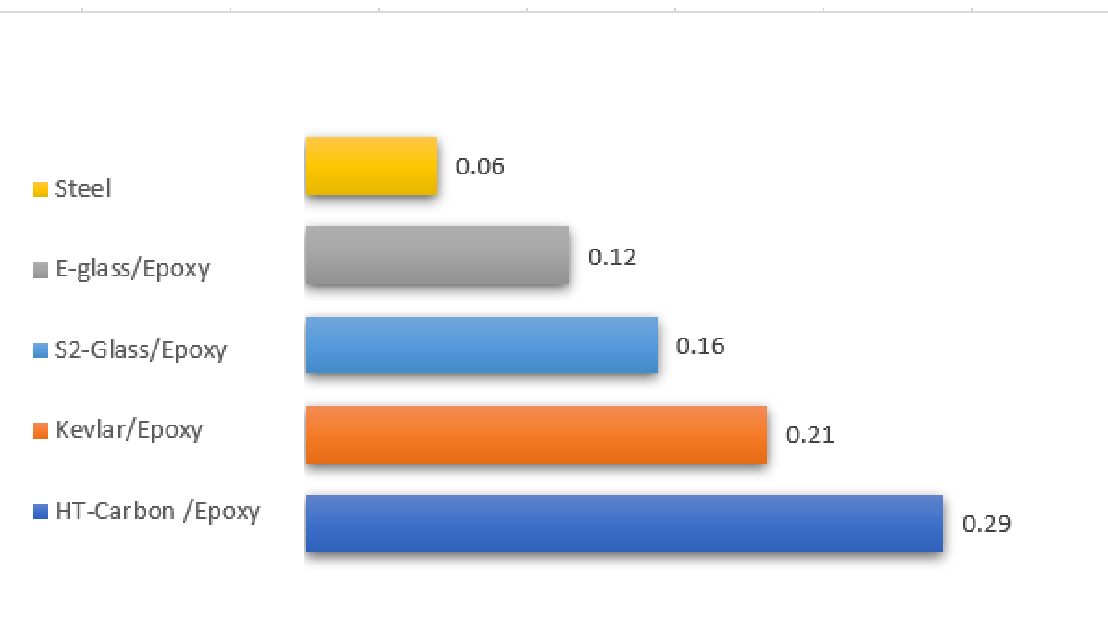


Figure 4.5: Strain energies for spring materials

The fatigue strength of materials is also the major design criteria while selecting the desired material for leaf sprig. The fatigue strength of Kevlar/epoxy, E-glass/epoxy and HT-Carbon/epoxy are found to be 352 MPa, 241 MPa and

672 MPa respectively. The impact strength of both Kevlar and E-glass fiber are higher than carbon fibers due to which the fracture propagation energy required for bot Kevlar and E-glass fibers are higher. Therefore, for this work, the two materials selected based on specific strain energy capacity for spring material are Kevlar/epoxy and E-glass/epoxy.

However, in present work semi-elliptical steel leaf spring of Suzuki Mehran car was examined under static loading and replaced by Kevlar fiber reinforcement oriented at $[0/45/-45/90/90/-45/45/0]2$ and E-glass fiber reinforcement oriented at $[0/0/45/0/0/-45/0/0/90/0]s$ angles with epoxy matrix using hand lay-up stacking. These stacking sequences were considered because it provides the better impact resistance due to 0° ply on the exterior side and 45° ply at the constant interface[62]. The properties of both composite materials used for analytical analysis of composite leaf spring are given in Table 4.2

Table 4.2: Properties of E-glass and Kevlar epoxy Composite [70],[71]

Properties	**E-glass/Epoxy**	**Kevlar/Epoxy**
Tensile modulus along X-direction(Ex).MPa	34000	80000
Tensile modulus along Y-direction(Ey).MPa	6530	55000
Tensile modulus along Z-direction (Ez). MPa	6530	80000
Longitudinal Tensile strength of the material. MPa	900	1400
Compressive strength of the material. MPa	450	335
Shear modulus along XY-direction (Gxy), MPa	2433	2200
Shear modulus along YZ-direction (Gyz), MPa	1698	1800
Shear modulus along ZX-direction (Gzx), MPa	2433	2200
Poisson ratio along XY-direction (Nuxy)	0.366	0.34
Poisson ratio along YZ-direction (Nuxy)	0.217	0.34
Poisson ratio along ZX-direction (Nuxy)	0.217	0.4
Mass density of the material (ρ). Kg/m^3	2600	1400
Flexural modulus of the material. MPa	40000	196.43
Volume fraction of fibers, Vf	60%	60 %

The properties of composite materials were used for macromechanical analysis to determine the stresses and strains in composite leaf springs. The macromechanical analysis was performed on a laminate to determine the midplane strains and curvatures. The midplane strains were used to find the developed stresses under maximum loading conditions. The Fig. 4.6 shows the location of plies from the midplane of the laminate.

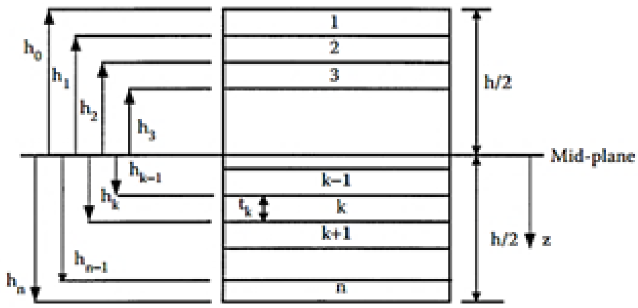


FIGURE 4.6: Coordinates of plies from Midplane of a laminate[60]

The thickness of the laminate 'h' can be found by the summation of all the 'n' plies in a laminate. The thickness of each ply is donated by 'tk'. The laminate thickness, location of the laminate midplane, the z-coordinate of each ply from bottom and top surface can be found using the following equations respectively.

$$h = \sum_{k=1}^{n} t_k \tag{4.14}$$

$$midplane = \frac{h}{2} \tag{4.15}$$

Ply k: (k=2,3,. . . . .n-2,n-1):

$$h_{k-1} = \frac{-h}{2} + \sum_{1}^{k-1} t(topsurface) \tag{4.16}$$

$$h_{k-1} = \frac{-h}{2} + \sum_{1}^{k-1} t(Bottomsurface) \tag{4.17}$$

Ply n:

$$h_{n-1} = \frac{h}{2} - t_n(Topsurface) \tag{4.18}$$

$$h_n = \frac{h}{2}(Bottomsurface) \tag{4.19}$$

To analyze the composite laminate the mathematical relations given in equation were derived to find the bending moment, normal forces and shear forces in the laminate. These relations were used further to determine the stresses in the laminated composite under applied loading. To find the midplane strains and curvatures in composite material the following relations can be used.

$$\begin{bmatrix} N_x \\ N_y \\ N_{xy} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{13} \\ A_{21} & A_{22} & A_{23} \\ A_{31} & A_{32} & A_{33} \end{bmatrix} \begin{bmatrix} \epsilon_x^\circ \\ \epsilon_y^\circ \\ \gamma_{xy}^\circ \end{bmatrix} + \begin{bmatrix} B_{11} & B_{12} & B_{13} \\ B_{21} & B_{22} & B_{23} \\ B_{31} & B_{32} & B_{33} \end{bmatrix} \begin{bmatrix} K_x \\ K_y \\ K_{xy} \end{bmatrix} \tag{4.20}$$

$$\begin{bmatrix} M_x \\ M_y \\ M_{xy} \end{bmatrix} = \begin{bmatrix} B_{11} & B_{12} & B_{13} \\ B_{21} & B_{22} & B_{23} \\ B_{31} & B_{32} & B_{33} \end{bmatrix} \begin{bmatrix} \epsilon_x^\circ \\ \epsilon_y^\circ \\ \gamma_{xy}^\circ \end{bmatrix} + \begin{bmatrix} D_{11} & D_{12} & D_{13} \\ D_{21} & D_{22} & D_{23} \\ D_{31} & D_{32} & D_{33} \end{bmatrix} \begin{bmatrix} K_x \\ K_y \\ K_{xy} \end{bmatrix} \tag{4.21}$$

The extensional matrix [A], coupling matrix [B] and bending stiffness matrix [D] can be related to transformed reduced stiffness matrix $[\bar{Q}]$ through the thickness of the laminate and can be given as:

$$A_{ij} = \sum_{k=1}^{n} [\bar{Q_{ij}}]_k (h_k - h_{k-1}) \tag{4.22}$$

i = 1,2,6; j=1,2,6

$$B_{ij} = \frac{h}{2} \sum_{k=1}^{n} [\bar{Q_{ij}}]_k (h_k^2 - h_{k-1}^2) \tag{4.23}$$

i = 1,2,6; j=1,2,6

$$D_{ij} = \frac{h}{3} \sum_{k=1}^{n} [\bar{Q_{ij}}]_k (h_k^3 - h_{k-1}^3) \tag{4.24}$$

i = 1,2,6; j=1,2,6

The transformed reduced stiffness matrix $[\bar{Q}]$ can be determined from the following relations.

$$\bar{[Q]} = [T]^{-1}[Q][R][T][R]^{-1} \tag{4.25}$$

Where, The transformed matrix [T], reduced stiffness matrix [Q] and Reuter matrix [R] can be given as.

$$[Q] = [S]^{-1} = \begin{bmatrix} \frac{1}{E_1} & \frac{-v_{12}}{E_1} & 0 \\ \frac{-V_{12}}{E_1} & \frac{1}{E_2} & 0 \\ 0 & 0 & \frac{1}{G_1 2} \end{bmatrix} \tag{4.26}$$

Where, E_1, E_2, G_{12}, V_{12} are the properties of the material.

$$[T] = \begin{bmatrix} c^2 & s^2 & 2sc \\ s^2 & c^2 & -2sc \\ -sc & sc & -s^2 \end{bmatrix} \tag{4.27}$$

c = $\cos(\theta)$ s= sin (θ)

$$[R] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 2 \end{bmatrix} \tag{4.28}$$

The equations used to determine the global strain and global stresses on the top and bottom of each ply of composite leaf spring can be found by the following equations respectively.

$$\begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{bmatrix} = \begin{bmatrix} \epsilon_x^\circ \\ \epsilon_y^\circ \\ \gamma_y^\circ \end{bmatrix} + z \begin{bmatrix} K_x \\ K_y \\ K_{xy} \end{bmatrix} \tag{4.29}$$

$$\begin{bmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{bmatrix} = \begin{bmatrix} \bar{Q_{11}} & \bar{Q_{12}} & \bar{Q_{13}} \\ \bar{Q_{21}} & \bar{Q_{22}} & \bar{Q_{23}} \\ \bar{Q_{31}} & \bar{Q_{32}} & \bar{Q_{33}} \end{bmatrix} \begin{bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{bmatrix} \tag{4.30}$$

The above-mentioned equations were solved using Matlab® with the available design data for composite leaf springs. The stresses at the top and bottom of each ply was calculated. The failure criteria based on Tsai-Wu theory was applied. This theory is based on strain energy theory given by Beltrami and applied to the lamina in plane stress. The lamina should satisfy the following criteria otherwise it would be considered as fail.

$$H_1\sigma_1 + H_2\sigma_2 + H_6\tau_{12} + H_{11}\sigma_1^2 + H_{22}\sigma_2^2 + H_{66}\tau_{12}^2 + 2H_{12}\sigma_1\sigma_2 < 1 \tag{4.31}$$

Where $H_1, H_2, H_6, H_{11}, H_{22} and H_{66}$ can be found using strength parameters of the material and H12 can be found from empirical relation given by Tsai-Hill failure theory. The value obtained from Tsai-Wu failure theory was found to be 0.4 in case of E-glass/epoxy and 0.31 in case of Kevlar/epoxy material which satisfy the above equation.

Chapter 5

Numerical Analysis

5.1 Introduction

In modern engineering manufacturing, computer-aided engineering techniques play a vital role in design process of various types of different products. Thus, the time dedicated for the development of the new model has been reduced significantly due to increase complexity of the developed system as noticeable in electronic and automotive industry. Moreover, the cost required to develop and asses a new model was also reduced. Though, it should be considered that model to be designed should have enough complexity to provide excellent accuracy and valuable information about the systems performance and simultaneously modest to reduce the computing time. However, to achieve these contradicting tasks, different techniques need to be investigated to develop the model.

This chapter investigates the numerical parameters that were used to conduct the numerical analysis. In section 5.2, the CAD model of the leaf spring was designed to perform the numerical analysis. In section 5.3, some validations were conducted before performing the numerical analysis. Different models were analyzed and validated to determine the required parameters for the analysis of leaf spring. As It was assumed that leaf spring is a simply supported beam, therefore three-point bending analysis was performed on beam to determine the appropriate mesh size for the model. Moreover, flat leaf spring, single curved leaf spring, contact analysis for multi-leaf spring and modal analysis was also performed to determine the mesh sensitivity and which type of contact for multi-leaf spring was preferable.

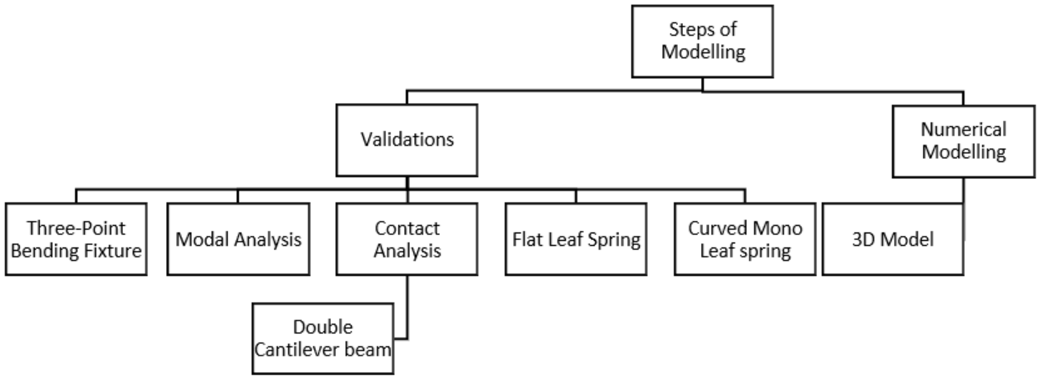


FIGURE 5.1: Modelling Chronology

5.2 CAD Modelling

The initial step in any analysis is the CAD modelling of the physical entity. In present work, the solid modelling of leaf spring was carried out in CREO 3.0 software with available design data given in table 5.1. The leaf spring was first designed according to conventional leaf spring dimensions and later a single leaf of same dimensions was designed as composite leaf spring because in actual practice it is manufacture as mono-leaf spring. In case of optimization for composite leaf spring, the spring was modelled as two cantilever beams separately with same radius of curvature as original leaf spring. However, the spring was not symmetric in shape, so the length of first half was more than the rear half length of the spring. The original length of spring was 965.2 mm and length of first half was measured to be 520.7 mm and rear half-length was measured to be 444.5 mm from center of front eye to the center of rear eye unit. The front and rear half-length of the spring was then divided into 11 parts having same cross-sectional area using space-claim software. The steel leaf spring has density of 7800 Kg/m^3 whereas Kevlar/epoxy and E-glass/epoxy has the density of 1400 Kg/m^3 and 2600 Kg/m^3 respectively.

5.2.1 Modelling of Steel Leaf Spring

The material used for the manufacturing of steel leaf spring is 51CrV4. The chemical composition of such material is given in table5.1. The conventional leaf spring consist three leaves, two of them are full length leaves and one is graduated leaf slightly smaller in length. The fist leaf was curved at both ends to form the shape of the eyes for attachment to the frame of the body. All three leaves of the spring were modelled separately then combined together with the help of constraints. The 3D model of steel leaf spring is given in Fig.5.2

TABLE 5.1: Chemical composition of material used for conventional leaf spring

Material	**Chemical Composition**
51CrV4	**C % — Si% — Mn% — P% — S% — Cr% — V% — Fe%**
-	0.47-0.55 — >0.4 — 0.7-1.1 — >0.025 — >0.025 — 0.9-1.2 — 0.10-0.25 — Rest

FIGURE 5.2: 3D model of Steel leaf spring

5.2.2 Modelling of Composite Leaf Spring

The design parameters used for composite leaf spring are same as steel leaf spring. The properties of composite material depend upon the fiber volume fraction and the type of resin used. In the present work the properties of E-glass/Epoxy and Kevlar/Epoxy obtained from experimental work based on three-point bending and tensile testing was used in the modelling of composite leaf spring. The properties of both composite materials given in Table 4.2 were used to model the composite leaf spring.

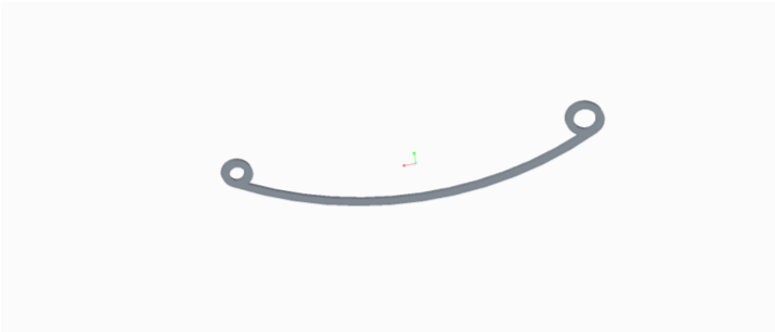


Figure 5.3: 3D model of Composite leaf spring

5.3 Numerical Modelling

ANSYS is used to design, model and to carry out analysis of leaf spring. As discussed in the preceding section two types of leaf spring are considered for this study i.e. Original curved leaf spring model and Flat leaf spring model. For both cases the finite element model is used, first they are plotted in XY plane then extruded in the Z-direction. Areas are selected to make the contact between two surfaces of two leaves. Validations were carried out for different contact types by comparing it with Analytical results, it was found that 'No separation' contact type was found most suitable.

5.3.1 Validations

It is important to validate the tool before calculating or analyzing the actual physical model and to predict the error between analytical or numerical values. Therefore, validation was carried out with help of simple geometries which helps in determining the exact solution of the design and are discussed in this section.

It was assumed that leaf spring are double cantilever beam symmetric at the center. A series of loading's are applied and as a result, stresses and deflection are determined analytically and numerically to calculate the range of error between these results. The percentage error between numerical and analytical result provides the information about the deviation of the results from the real value. Therefore, a simple beam of steel material having same width and thickness as steel leaf spring with maximum length of 1000 mm was analyzed as a test case. A series of loads were applied on the end of the beam and maximum bending stress found from numerical and analytical analysis was 1384 MPa and 1346 MPa respectively at given load of 550 N. It was observed that the relative change between these values was found to be 2.6 %. The percentage error calculated between the numerical and analytical results is insignificant and acceptable. The analytical relations to determine bending stress and deflection for cantilever beam are given below:

$$\sigma = \frac{6WL}{nbt^2} \tag{5.1}$$

$$\delta = \frac{4WL^3}{nEbt^3} \tag{5.2}$$

Three-Point Bending Fixture

The three-point bending design for leaf spring was also taken into consideration for the analysis because in actual practice the leaf spring is loaded at three point when it is mounted in the vehicle. The front eye is attached to the frame, rear eye is also attached to the frame with the help of additional link "shackle" to provide rotation and translation and center of the spring is attached to the axle of the vehicle. So, there are three points of loading. In some cases, the leaf spring is mounted above the axle instead below the axle to provide enough space for attachment. A series of loadings are applied, as a result stresses and deflection are determined analytically and numerically to determine the range of error between these results. The design parameters for simple cantilever beam is given in Table 5.6. The analytical relation for stresses and deflection for three-point bending fixture are given below:

$$Q_f = \frac{3FL}{2bh^2} \tag{5.3}$$

$$\delta = \frac{PL^3}{48EI} \tag{5.4}$$

TABLE 5.2: Parameters for Three-point bending fixture model

Specifications	
Length(L)	160 mm
Width(b)	15mm
Thickness(t)	2 mm
Support diameter	3 mm
Intender diameter	3 mm
Young's Modulus(E)	2.1 x 105 MPa

TABLE 5.3: Comparison of Analytical and Numerical values for Three-Point bending fixture

Load	**Analytical**		**Numerical**		**% Error**	
N	ρ MPa	δ Mm	σ MPa	δ mm	ρ %	δ%
100	100	0.0634	102.33	0.0569	2.33	10.25
200	200	0.1269	204.66	0.1139	2.33	10.24
300	300	0.1905	304.66	0.1709	2.33	10.28
400	400	0.2539	409.32	0.2279	2.33	10.24
500	500	0.3174	511.65	0.2849	2.33	10.23

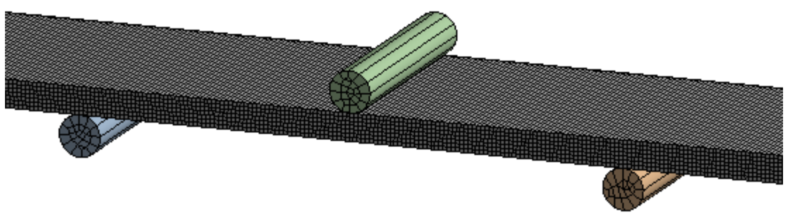


FIGURE 5.4: Three-Point Bending fixture Model

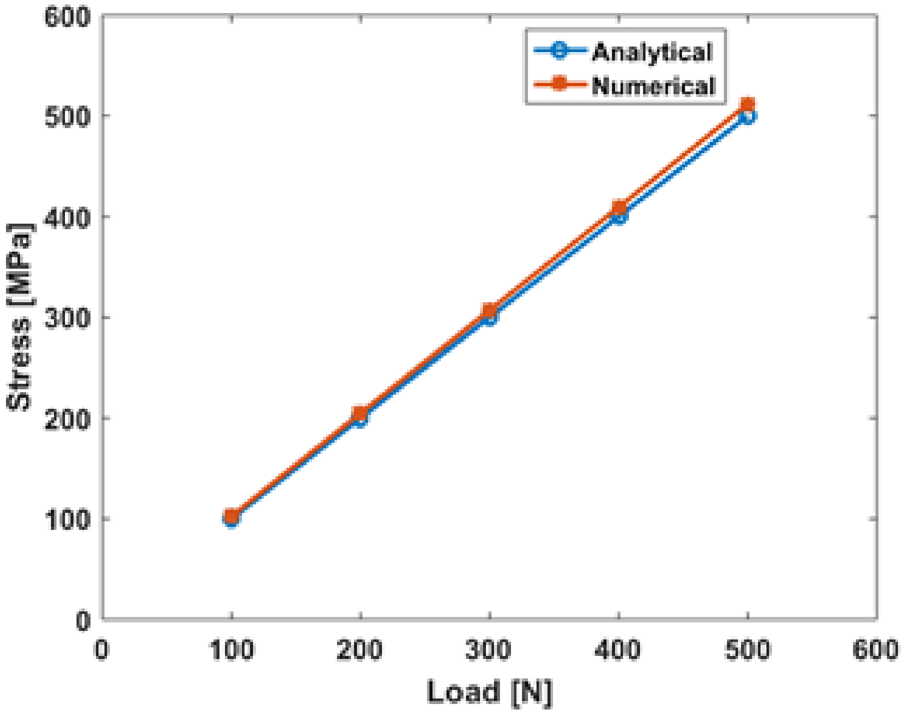


FIGURE 5.5: Variation of Stress with load applied

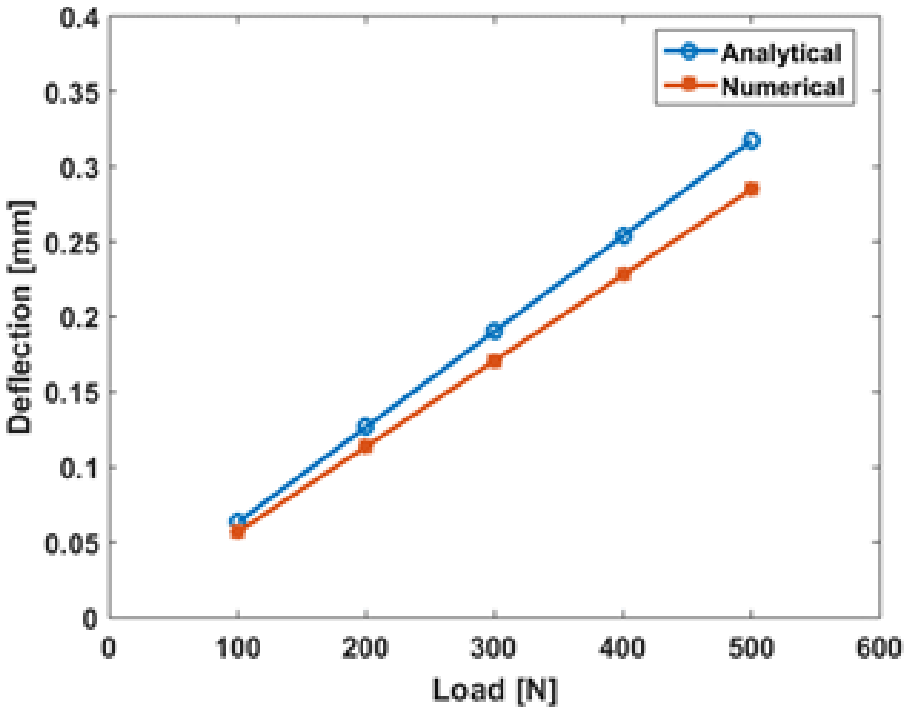


FIGURE 5.6: Variation of deflection with load applied

The graph shows the comparison of numerical and analytical results obtained for three-point bending fixture model. The maximum load of 500 N was applied and bending stress was calculated from numerical and analytical analysis. However, maximum bending stress from analytical and numerical analysis was found to be 500 MPa and 511 MPa. It was observed that as increase in load will increase the stresses in the beam. The same trend was observed in case of deflection for three-point bending model. The maximum bending stress was obtained at the supports and maximum deflection was obtained at the middle of the beam. found that the percentage error between the analytical and numerical results is insignificant and acceptable.

Contact Analysis

The contact analysis was performed to predict out the desired contact type between the surfaces of leaf spring for numerical analysis. The numerical values vary from one contact type to another therefore it is necessary to determine the contact type for the design being modelled to obtain the precise results. Thus, analytical relations were used to determine the stress produced in double cantilever beam and results are compared with numerical solution obtained for two different type of contact between double cantilever beam. A series of loadings are applied, as a result stresses and deflection are determined analytically and numerically to determine the range of error between these results. The design parameters for simple cantilever beam is given in Table 5.8. The analytical relations for bending stress and deflection in more than one beam are defined as:

$$\sigma = \frac{6WL}{nbt^2} \tag{5.5}$$

$$\rho = \frac{6WL^3}{Enbt^3} \tag{5.6}$$

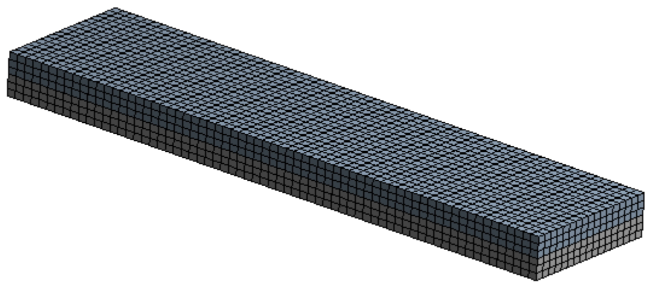


FIGURE 5.7: Model of Two Beams with contact surface

TABLE 5.4: Parameters for two beams with contact surface

Specifications	
Length(L)	200 mm
Width(b)	50 mm
Thickness(t)	7 mm
Young's Modulus(E)	2.1 x 105 MPa
Contact Analysis	• Bonded • No Seperation

TABLE 5.5: Comparison of Analytical and Numerical values for Three-Point bending fixture

Load	**Analytical**		**Bonded Contact**		**Error Bonded**		**No Separation**		**Error No Speration**	
N	σ MPa	δ mm	σ MPa	δ mm	σ%	δ%	σ MPa	δ mm	σ %	δ%
150	36.73	0.666	18.07	0.164	50.8	75.3	38.05	0.65	3.5	1.5
250	61.22	1.11	30.13	0.27	50.7	75.6	63.43	1.08	3.6	2.7
350	85.71	1.55	42.18	0.38	50.7	75.4	88.80	1.52	3.5	1.9
450	110.20	1.99	54.23	0.49	50.7	75.3	114.17	1.95	3.5	2.0
550	134.69	2.44	66.29	0.60	50.7	75.4	139.55	2.39	3.5	2.0

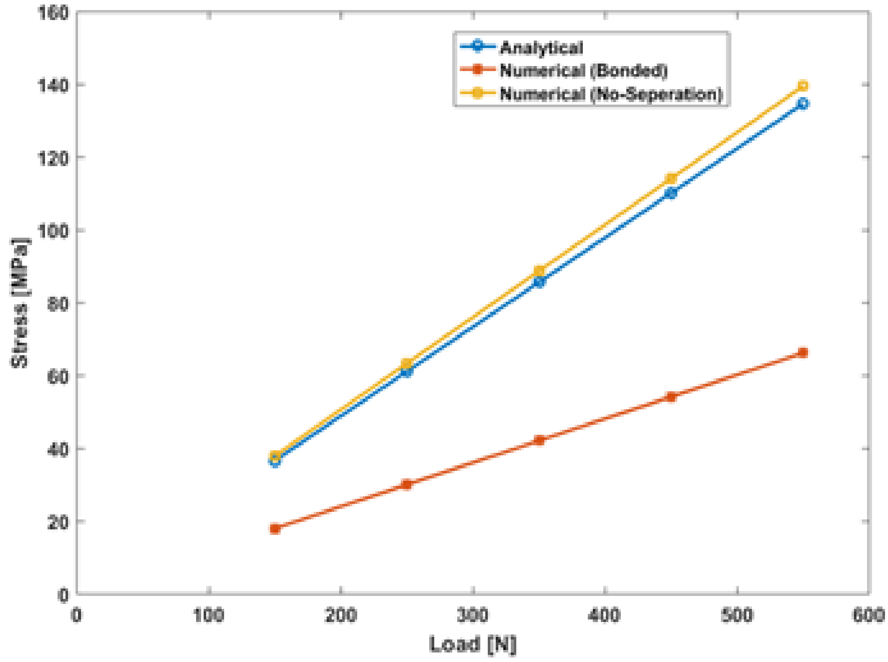


FIGURE 5.8: Variation of Stresses with load for different contact types

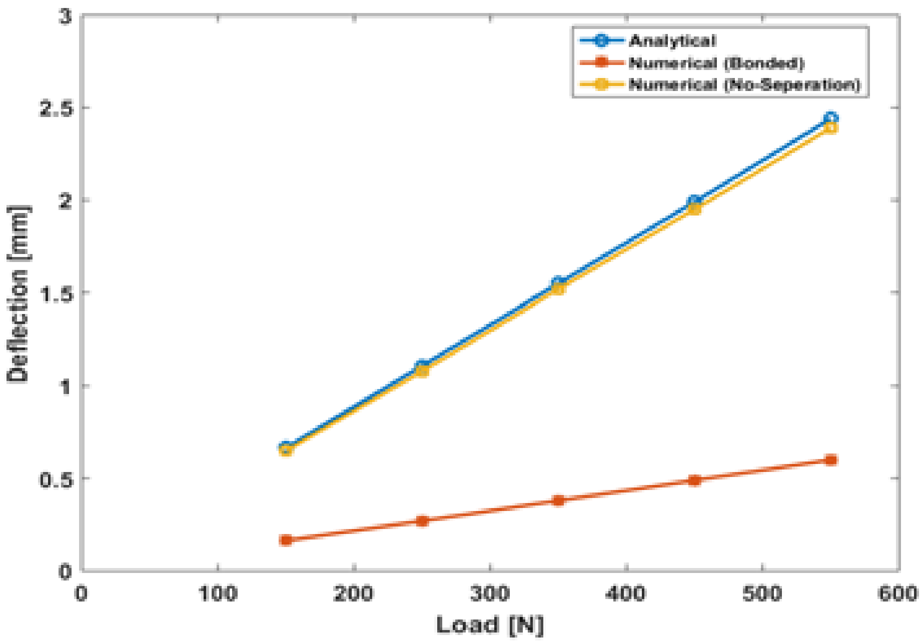


FIGURE 5.9: Variation of deflection with load for different contact types

The graph shows the comparison of numerical and analytical results obtained for contact type between two beams. It was observed that graph trends almost horizontally for bonded contact type whereas trend of non-separation contact type is almost parallel to the trend of analytical result. The percentage error between the analytical results and numerical results obtained with bonded contact type is relatively higher than no separation contact type. The contact type between two surfaces depends upon the geometry of the design.

Flat Leaf Spring

The flat leaf spring was also considered for the validation process. The leaf spring consisting of three straight beams without an arc was considered for the analytical and numerical analysis. A no-separation contact type was considered between the beams. A series of loadings are applied, as a result stresses and deflection are determined analytically and numerically to determine the range of error between these results. The design parameters for simple cantilever beam is given in Table 5.10. The analytical relations for bending stress and deflection in flat leaf spring and two or more beams place one above the other are defined as:

$$\sigma = \frac{6WL}{nbt^2} \tag{5.7}$$

$$\delta = \frac{6WL^3}{nEbt^3} \tag{5.8}$$

Where, 'n' is the number of beams, 't' is the thickness of the leaf, 'b' is the width of the leaf, 'E' is the modulus of the material.

TABLE 5.6: Parameters for flat leaf spring

Specifications	
Length(2L)	965.2 mm
Width(b)	50 mm
Thickness(t)	7 mm
Modulus of Elasticity (E)	$2.1x10^5$ MPa

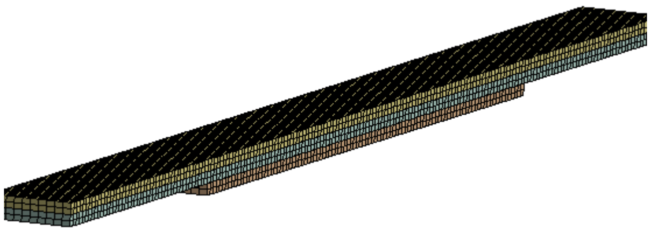


FIGURE 5.10: Flat leaf spring model

TABLE 5.7: Comparison of analytical and numerical values for flat leaf spring

Load	**Moment**	**Analytical**		**Numerical**		**% Error**	
N	Nm	ρ	δ Mm	ρ MPa	δ mm	ρ %	δ%
150	72.39	177.28	28.08	180.76	19.474	1.96	30.6
250	120.65	295.46	46.81	301.27	32.45	1.96	30.6
350	168.91	413.65	65.53	421.77	45.43	1.96	30.6
450	217.17	531.84	84.26	542.28	58.42	1.96	30.6
550	265.43	650.03	102.98	662.79	71.403	1.96	30.6

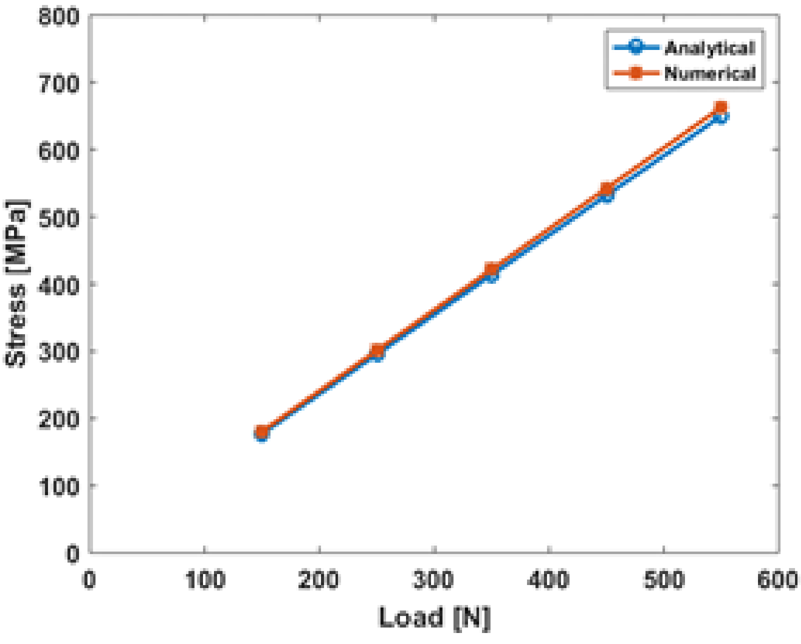


FIGURE 5.11: Variation of Stress with load applied for flat leaf spring

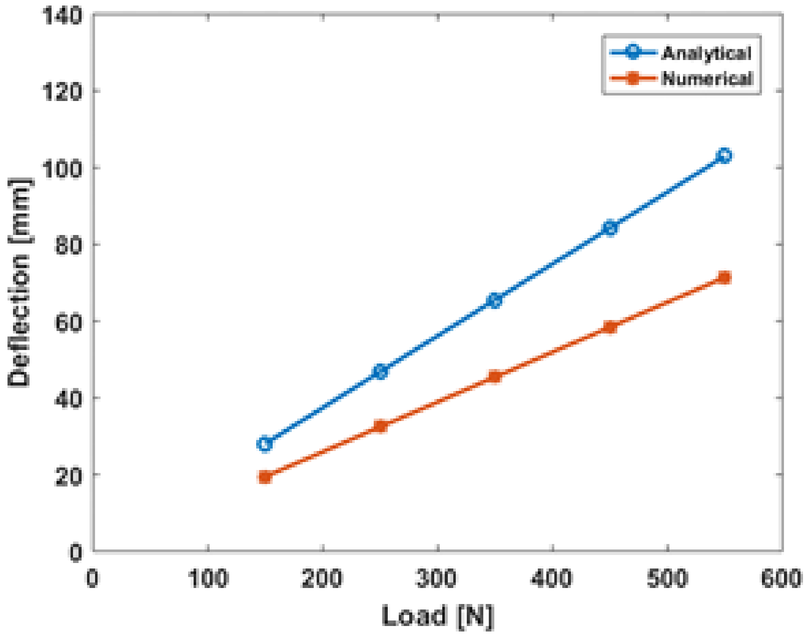


FIGURE 5.12: Variation of deflection with load applied for flat leaf spring

The graph shows the comparison of numerical and analytical results obtained for flat leaf spring.It was observed that increase in load will increase the stresses with small percentage error, but deflection shows relatively high percentage error due to flat geometry of leaf spring and due to one of the leaves is relatively smaller than other two leaves of the spring.

Curved Mono-Leaf Spring

The curved mono leaf spring having same arc radius as that of the actual leaf spring was analyzed analytically and numerically to determine the stresses and deflection produced in it. The mono leaf spring has the same design parameters as that of the actual leaf of the spring encompasses. A series of loadings are applied, as a result stresses and deflection are determined analytically and numerically to determine the range of error between these results. The design parameters for curved mono leaf spring is given in Table 5.12.

The analytical stress and deflection in curved leaf spring can be calculated from the relations are given in Eq 5.9 and 5.10 [66]:

Table 5.8: Parameters for Curved Mono-Leaf Spring

Specifications	
Length(2L)	965.2 mm
Width(b)	50 mm
Thickness(t)	7 mm
Modulus of Elasticity (E)	2.1 $\times 10^5$ MPa
Camber	127 mm
Area	350 mm^2
Radius of curvature	980.44 mm
Distance from center to neutral axis (R)	983.93585 mm
Distance from centroidal axis to neutral axis (N)	0.00415 mm
Distance from neutral axis to outer surface (Y_t)	3.50415 mm
Distance from neutral axis to inner surface (Y_c)	3.49585 mm

$$\sigma = \frac{M(r_o - R)}{ANr_o} \tag{5.9}$$

$$\delta = \frac{2WL^3}{3Ebt^3} \tag{5.10}$$

These relations were used to determine the stresses and deflections in a single curved steel leaf spring. Prior to this, flat leaf spring having two full length leaves and one graduated leaf was modelled and analyzed. However, modelling of flat steel leaf spring is important in order to compare it with flat composite leaf spring. The flat steel leaf spring was modelled without eye units because in this work composite leaf spring without eyes was modelled for the static and dynamic analysis. Specifications for curved leaf spring is the same as that of flat leaf spring and given in Table to determine the numerical and analytical results.

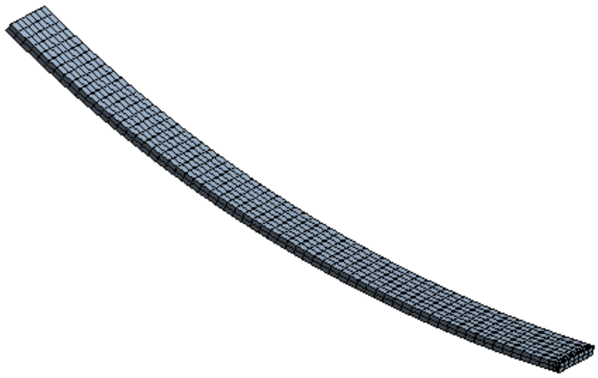


Figure 5.13: Curved Mono-Leaf spring Model

Table 5.9: Comparison of analytical and numerical values for Curved Mono-leaf spring

Load	**Moment**	**Analytical**		**Numerical**		**% Error**	
N	Nm	ρ	δ Mm	ρ MPa	δ mm	ρ %	δ%
150	72.39	168.45	18.72	176.97	21.47	5.05	12.8
250	120.65	280.75	31.20	294.95	35.78	5.05	12.8
350	168.91	393.05	43.69	412.93	50.00	5.05	12.7
450	217.17	505.35	56.17	530.91	64.41	5.05	12.7
550	265.43	617.68	68.66	648.89	78.72	5.05	12.7

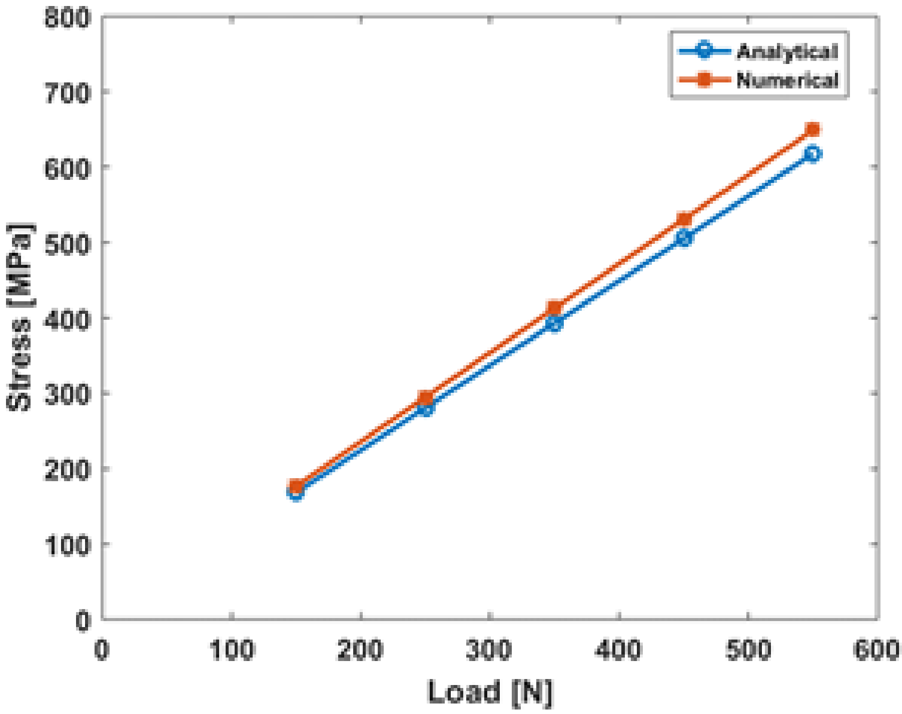


FIGURE 5.14: Variation of Stress with load applied on Curved Mono-leaf spring

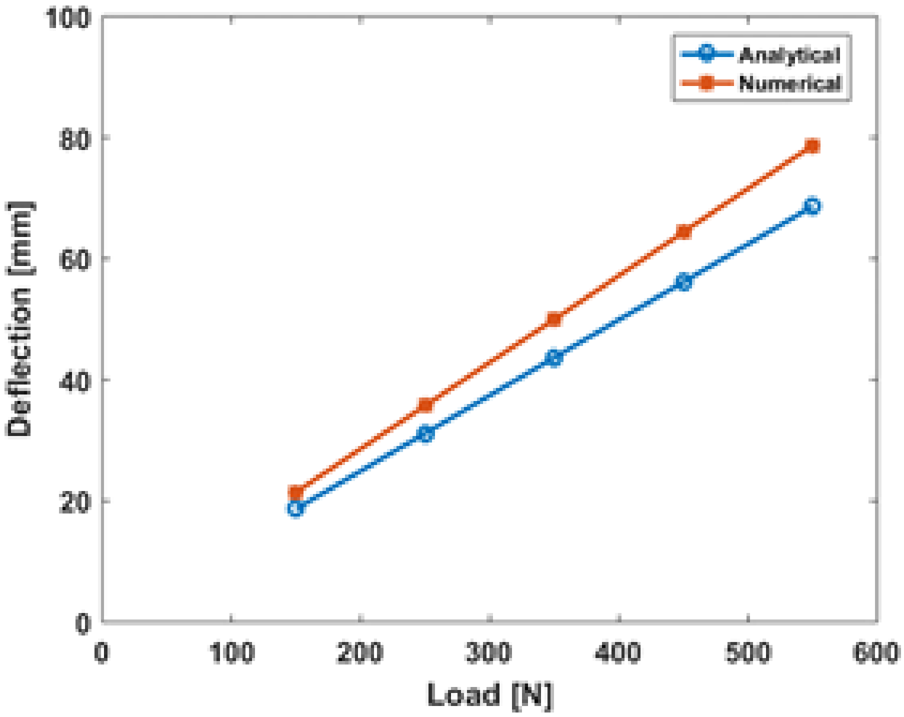


FIGURE 5.15: Variation of deflection with load applied on Curved Mono-leaf spring

The graph shows the comparison of numerical and analytical results obtained for curved leaf spring.Both results showed a good approximation within acceptable range. These results are obtained at mesh size of 7 mm and mesh sensitivity analysis was performed for curved beam. The Fig.5.16 shows the convergence diagram for curved beam and it was observed that at mesh size of 7 mm the solution converges with small increment in values for bending stresses.

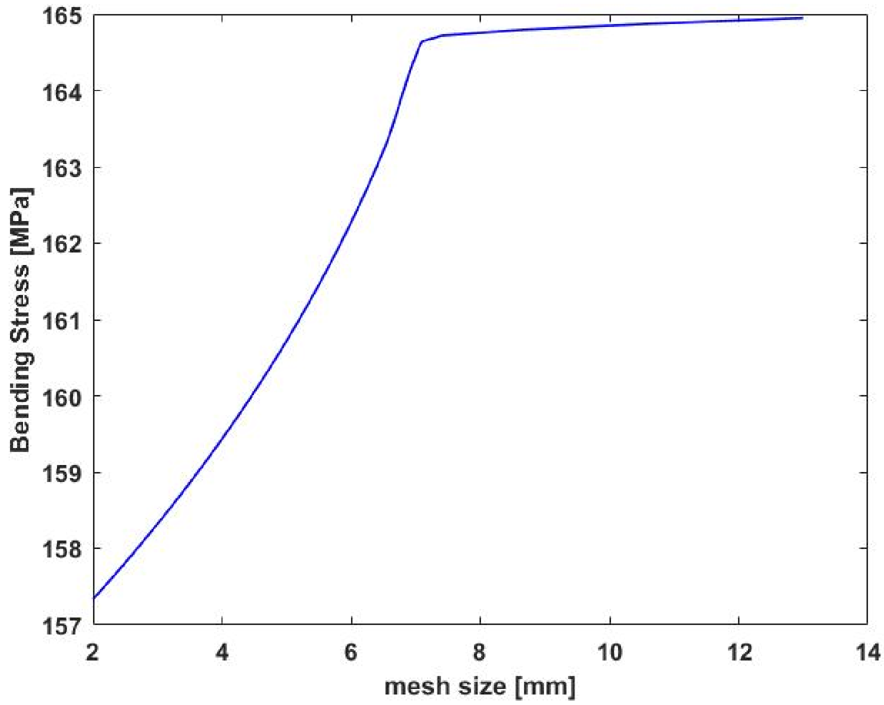


FIGURE 5.16: Mesh Convergence for Curved beam

Modal Analysis of Beam

The modal analysis was performed for different types of beams i.e. simply supported beam, cantilever beam, and fixed-fixed type beam to validate the analytical and numerical results and to determine which type of beam possess the maximum natural frequency. The parameters of beam used for modal analysis is given in table 5.10. The modal analysis provides the information about the frequency of the system. The first three frequencies of different types of beams are given in Table 5.11.

TABLE 5.10: Parameters for Cantilever beam

Specifications	
Length(2L)	965.2 mm
Width(b)	50 mm
Thickness(t)	7 mm
Modulus of Elasticity (E)	2.1×10^5 MPa
Camber	127 mm
Area	350 mm^2
Area moment of Inertia (I)	1429.16 mm4
Density (ρ)	$7.8 \times 10^-6 Kg/m^3$

TABLE 5.11: Comparison of analytical and numerical values obtained for different type of beams using modal analysis

51CrV4	**Fixed-Fixed Condition**		**Cantilevered**		**Simply Supported**	
	Analytical	Numerical	Analytical	Numerical	Analytical	Numerical
Mode-1 Frequency (Hz)	40.08	40.36	6.30	6.32	17.68	17.68
Mode-2 Frequency (Hz)	110.47	111.22	39.47	39.61	70.71	70.72
Mode-3 Frequency (Hz)	216.98	217.69	110.52	110.89	159.12	159.16

The results given in above Table 5.11 shows the comparison of the numerical and analytical results based on natural frequencies for different types of beams. The results show the good approximation between the analytical and numerical results. It was observed that cantilever beam has lowest natural frequency than other types of beams. Beside natural frequency, there are some other factor which influences the modes of the structure e.g. mass and boundary conditions etc. Increasing mass will increase the frequency of the system and varying boundary conditions shows varying natural frequencies. However, leaf springs are simply supported beams so the boundary conditions of simply supported type will be used to determine the mode shapes of the system.

5.4 Finite Element Modelling

Finite element analysis is a computer aided numerical method which is widely used in all engineering applications to calculate strength, displacement, deflection and behaviour of real-life objects. It can also be helpful in predicting the mechanical behaviour of engineering structures under static and dynamic loadings, which is the main focus of this study. In finite element method, the complex geometry is divided into minute parts or elements. Each individual part has its own behaviour which is represented by a set of equations, representing the behaviour of the whole structure. Thus, all these equations are solved by workstation simultaneously and generates result which in return predicts the behaviour of the individual elements; hence it can generate deflection and stresses of the whole structure. FEA includes variety of analysis for simple to complex structures and it is very efficient method in solving problem involving large complexities. Thus, finite element analysis was used to analyze the leaf spring under static and dynamic loading conditions.

5.4.1 Assumptions for Analysis

1. Automobile is assumed to be stationary for static loading condition and no other forces act on the spring except the weight of the vehicle.
2. There are 2 Semi-elliptic leaf springs attached to the rear axle of the vehicle and weight of the vehicle was distributed on both leaf springs equally.
3. The geometrical properties are same for both conventional and composite leaf spring.

5.4.2 Meshing

The discretization of complex geometries into smaller elements is known as meshing. It involves the generation of nodes at each element. The speed of simulation, accuracy and convergence were influenced by the size and the type of the meshing used. However, to achieve the accurate solution and faster results, the meshing tools required to be better and more reliable. In this work the element size was taken as 7 mm and quadrilaterals were used as a mesh type for steel leaf spring. In case of composite leaf springs, the mesh type used was quadrilaterals with element size of 1.499 mm. Triangle mesh were also employed to refine the mesh in both steel and composite leaf springs where necessary.

Chapter 6

Static Analysis Of Leaf Spring

6.1 Introduction

In engineering science, stress analysis is one of the utmost significant steps, as most of the structural and engineering component failure is due to stress. In order to investigate the true magnitude of a stress at which structure or components are failed, a suitable stress analysis should be required along with suitable loading and boundary conditions. However, finite element analysis is usually used to perform stress analysis. The stress analysis of composite based structures is more complex than other structures made of conventional materials. The composite and steel leaf springs are analyzed under maximum loading of the vehicle.

This chapter investigates the behavior of steel and composite leaf spring under static loading conditions. It outlines the stress analysis techniques which are used to calculate numerical and analytical stress produced in both conventional and steel leaf spring. In section 6.2, the boundary conditions of leaf spring are defined and in section 6.3, the load acting on leaf spring was calculated. In section 6.4, the stress analysis is presented. The bending stress and deflection were obtained as a result of the analysis. The mesh sensitivity analysis based on steel leaf spring was also done in this section. The composite leaf springs was analyzed under static loading conditions was also discussed in section 6.4.

6.2 Boundary Conditions

The numerical solution of mechanical design can be obtained by applying initial conditions and constraints based on the physical model known as boundary conditions. In vehicles, the leaf springs are mounted on its axle and eyes of the spring are attached to the frame. The front eye is attached to the frame with the help of pin joint and allow the eye to rotate while the rear eye of the spring is attached to the frame with the help of link known as shackle which allows the translation of the spring in one direction. The other end of the shackle is connected to the spring with pin joint due to which it also rotates. The figure 6.1 shows the actual boundary conditions of the spring in which the front eye labeled “A” rotates around Z-axis and rear eye labeled “B” also rotates around Z-axis but it also translates in X-direction due to shackle. The force is applied at point “C” which is equal to the weight of the vehicle in case of static analysis but it varies with time while conducting dynamic analysis.

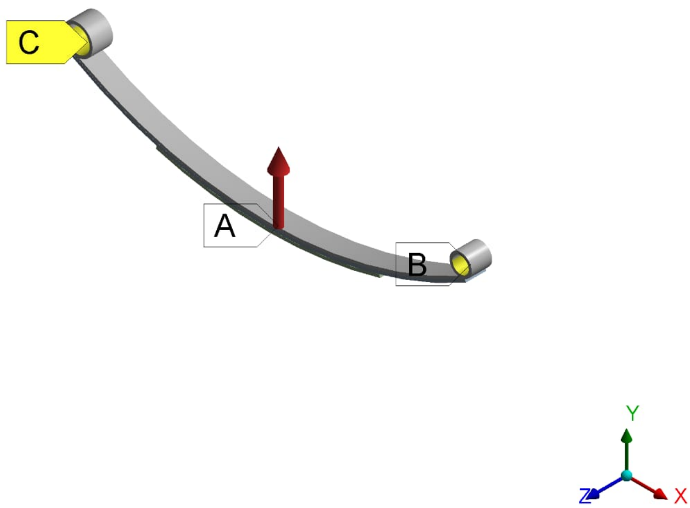


FIGURE 6.1: Boundary conditions of leaf spring

6.3 Load Calculation of Leaf Spring In

In more realistic situation the front half of the leaf spring incorporates the forces produced during braking and acceleration of vehicle, are assumed to be more critical than forces produced at the rear half. So, the front half is analysed first under maximum loading condition. The forces acting on leaf spring is divided into front and rear half of leaf spring as shown in Fig 6.2. These forces are found using engineering static techniques. So, under maximum load of 4900 N, the vertical force acting on front half is 2256.57 N. The side loads are assumed to be 75 % of the vertical load[37]. These loads are produced when vehicle is turning and thus change in angular momentum takes place. Therefore, this load is found to be 1692.375 N. The longitudinal load occurs due to change in linear momentum during braking and acceleration of vehicle. The maximum possible twisting angle between the axle and each eye was considered as 9°. The twisting angle is produced due to wind up torque, which is the relative motion of the rear wheels when one of the wheels moves up and other moves down. Applying these forces on front half of Kevlar/epoxy leaf spring, the maximum normal stress was found to be 72.57 MPa. Similarly, applying these forces on front half of E-glass/epoxy leaf spring, the maximum normal stress obtained is 53.66.

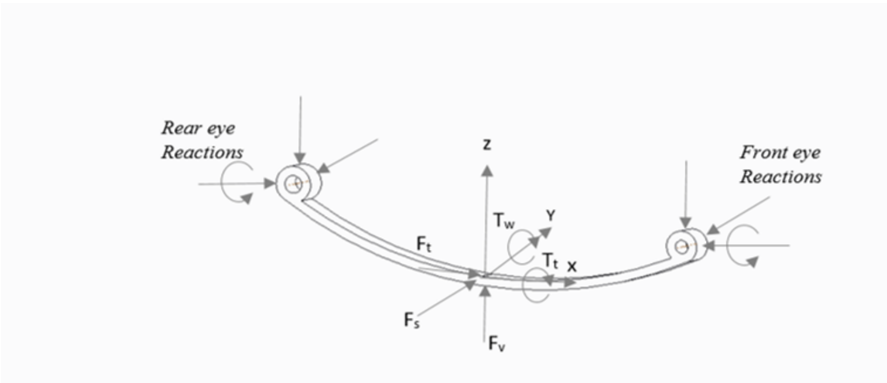


FIGURE 6.2: Forces acting on leaf spring. Vertical load (Fv), Side Load (Fs), longitudinal load (Ft), twisting torque (Tt), windup torque (Tw).[37]

6.4 Stress Analysis

The stress analysis of mechanical design is of significant importance and involves two type of analysis i.e. static stress analysis and dynamic stress analysis. The stress analysis provides the information about the capacity of the design to sustain the applied load under defined boundary conditions. This section discusses the static stress analysis of both steel and composite leaf spring under maximum loading and actual boundary conditions.

6.4.1 Steel Leaf Spring

A static stress analysis on composite and steel leaf spring was performed under maximum loading conditions. The leaf spring was modelled in CAD software and imported in ANSYS to perform the static analysis. Initially, the mesh model was created using body sizing technique with mesh size of 7 mm. The boundary conditions (BC) are applied as discussed in section 6.3 to calculate the stresses and deflection. The amount of stresses and deflection produced depends upon the BC's applied on the model and generate the contours. The contour plots of the stresses and deflection provides the information about the variation of these values from maximum to minimum.

In present work, longitudinal strength of material is more than its compression strength. Therefore, the compressive surface of leaf spring was taken into more consideration due to high probability of failure at this side. In case of Steel leaf spring this stress was found to be 971.6 MPa more than 965.2 MPa obtained from analytical solution. The deflection in case of steel leaf spring was 109.99 mm. The yield strength of steel is 1125 MPa than the stress obtained through numerical solution which means that stresses produced under maximum loading conditions are within the specified range. The Table. 6.1 shows the comparison of analytical and numerical results for steel leaf spring.

TABLE 6.1: stress in Steel leaf spring

Analytical		Numerical	
σ	δ	σ	δ
(N/mm^2)	mm	(N/mm^2)	mm
965.2	114.69	971.59	107.99

6.4.1.1 Mesh Sensitivity Analysis

The convergence analysis was performed for finite element model to determine the mesh size sensitivity. The stresses and displacements were evaluated as a function of mesh size and results were recorded and given in Table. It was observed that the solution for stresses and displacements converge at 7 mm and no more variations were observed. The solution shows that the results were converged and no additional mesh refinement was required.

TABLE 6.2: stress in Steel leaf spring

No of Elements	Nodes	Bending Stress
846	7262	956.98
939	8046	959.54
1035	8878	960.14
1556	12608	963.72
1752	1428	970.47
1520	11406	972.82
1620	12648	974.47
2772	20939	974.43
3512	26328	971.60
7414	46365	971.62

However, the meshing of steel leaf spring as it involves the shell elements. The mesh model of steel leaf spring is shown in Fig. The numerical analysis includes

FIGURE 6.3: Meshed FE model of Steel Leaf Spring

static analysis, dynamic analysis and optimization which are discussed in next chapters.

6.4.2 Composite Leaf Spring

Composite materials have a good characteristic of storing strain energy in the direction of fibers. Since the strain energy stored in leaf spring varies linearly with allowable stress and inversely with modulus of elasticity in the longitudinal direction. The strength of composite material depends upon the fiber orientation and its stacking sequence. Therefore, quasi-isotropic lay-up is selected for Kevlar/epoxy and E-glass/epoxy material. Different stacking sequences for E-glass epoxy and Kevlar epoxy composite material were considered for the analysis For instance, the (0/0/45/0/0/-45/0/0/90/0)s and (0/45/-45/90/90/-45/45/0)s stacking sequences were considered for E-glass epoxy and Kevlar epoxy respectively. The selection of such stacking sequence was based on the dynamic loads acting on the spring due to road irregularities. However, Kevlar epoxy composite material does not provide good resistance in case of compressive stresses but provides substantial weight savings than E-glass epoxy composite.

The finite element analysis (FEA) for composite materials is slightly different from other materials such as steel. In case of composites, the material is created for the model prior to the analysis based on the selected stacking sequence while in case of other materials, the model is created with the desired material. Since, composite possess orthotropic properties therefore several additional properties are required to solve the problem involving composite materials.

Thus, composite leaf springs are designed in ANSYS software using ACP-Pre tool. The leaf spring model generated in CAD software was imported in ANSYS and mesh was created. Furthermore, the composite leaf spring was imported to ANSYS ACP-PRE to create a material for composite leaf springs. The stacking sequence as defined earlier was used as the direction of fibers in a lamina and thickness of each lamina was 1.3125 mm for both E-glass/epoxy and Kevlar/epoxy material. So, the total thickness of the laminate was 21mm. The total number of laminates were 16. The 0° fiber orientations were chosen along the longitudinal directions. Each lamina contains the properties of the whole material.

The same boundary conditions were applied as in case of steel leaf spring. The materials required to design the composite leaf spring models were created with the given fiber orientations and stacking sequence as discussed before. Different stacking sequence for Kevlar/Epoxy and E-glass/Epoxy material were used to modelled the composites leaf springs. The stacking sequence is very important in modelling. As the properties of the material varies with change in the stacking sequence and fiber directions in each ply. For instance, the schematic shown in Fig 35 shows the fiber orientations and stacking sequence of composite material model, where the lines drawn corresponding to the angles represents the fiber orientation and lay-up represents the stacking sequence of the ply.

The model was then simulated in ANSYS static structural tool to evaluate stresses and deflection. The solution of static analysis was transferred to ACP-post tool to determine the interlaminar stresses and factor of safety of composite leaf springs.

Interlaminar stress in Kevlar/epoxy leaf spring and E-glass/epoxy leaf spring was found to be 262.12 MPa and 148.9 MPa respectively while the margin of safety was found to be 1.25 for Kevlar/Epoxy leaf spring and 1 for E-glass/Epoxy leaf spring. However, margin of safety is one unit less than factor of safety. It means that factor of safety for Kevlar spring and E-glass spring was 2.125 and 2 respectively.

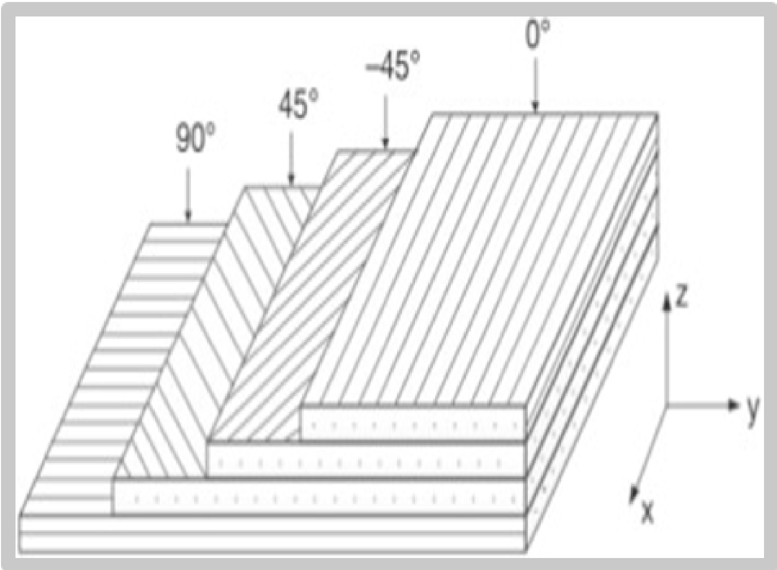


FIGURE 6.4: Fiber orientations in laminates

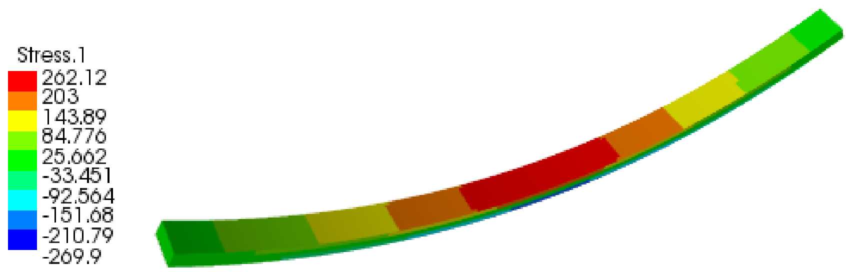


FIGURE 6.5: Interlaminar Stresses in Kevlar/Epoxy Leaf spring

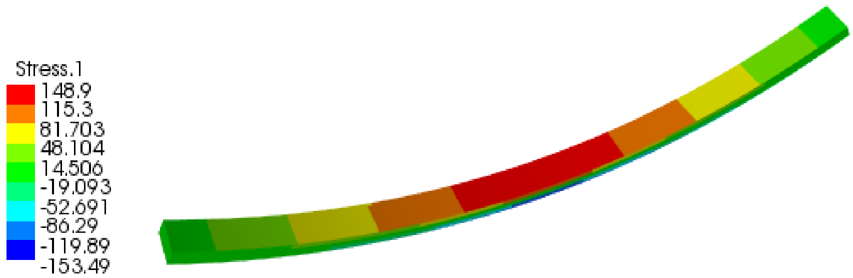


FIGURE 6.6: Interlaminar Stresses in E-glass/Epoxy Leaf spring

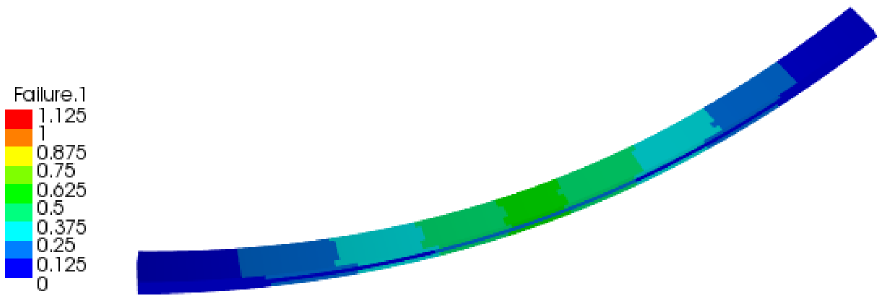


FIGURE 6.7: Margin of Safety for Kevlar/Epoxy Leaf Spring(FOS=2.125)

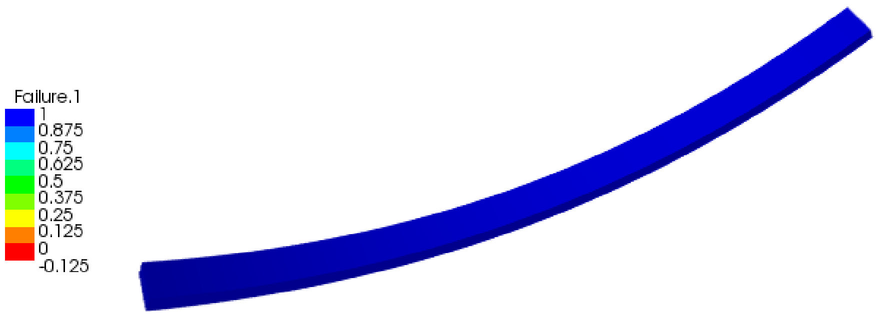


FIGURE 6.8: Margin of Safety for E-glass/Epoxy Leaf Spring (FOS=2)

6.5 Static Analysis Results

The leaf spring suspension system used in light weight vehicle was analyzed using analytical and numerical method. The bending stress and deflection of conventional and composite leaf springs are calculated as a result of analytical analysis. The analytical result for conventional leaf spring can be easily found using beam theory whereas the analytical result of composite leaf spring can be found by using laminate theory. The results are compared with numerical solution and shows good approximation. It was observed that the stresses produced in ordinary leaf spring was 965.2 MPa and 269.5 MPa in case of Kevlar/epoxy leaf spring and 145.32 in case of E-glass/epoxy leaf spring without eye units, the stress contours in conventional and composite leaf springs are shown in Fig.6.9 through 6.11.

Since, composite have anisotropic properties due to which all components of stresses are considered to as important factor in analysing the behaviour of leaf spring. The maximum stresses at the compressive side of Kevlar/Epoxy and E-glass/Epoxy were 269 MPa and 145 MPa respectively near axle seat and decreases in longitudinal axis towards the eyes of the spring. In case of Steel leaf spring this stress was found to be 971.6 MPa more than 965.2 MPa obtained from analytical solution. However, the compression strength of composite is more than the stresses obtained i.e. 517.1 MPa in case of Kevlar/epoxy composite, 450 MPa in case of E-glass/epoxy composite. The shear stress was evaluated to be 31 MPa and 19 MPa in Kevlar/Epoxy and E-glass/Epoxy leaf spring respectively are also in the range of shear strength of the material.

In more realistic situation the front half of the leaf spring incorporates the forces produced during braking and acceleration of vehicle, are assumed to be more critical than forces produced at the rear half. The forces acting on leaf spring is divided into front and rear half of leaf spring as shown in figure9.1. These forces are found using engineering static techniques. So, under maximum load of 4900 N, the vertical force acting on front half is 2256.57 N. The side loads are assumed to be 75 % of the vertical load [32]. These loads are produced when vehicle is turning and thus change in angular momentum takes place. Therefore, this load is found to be 1692.375 N. The longitudinal load occurs due to change in linear momentum during braking and acceleration of vehicle. If we assume, a vehicle moving at the velocity of 120 km/h reaches to 0 velocity in 5 s while brakes are applied, the forces in longitudinal direction would be 3260 N. The maximum possible twisting angle between the axle and each eye was considered as 9°. The twisting angle is produced due to wind up torque, which is the relative motion of the rear wheels when one of the wheels moves up and other moves down. Applying these forces on front half, the maximum normal stress on front half was found to be 136.19 MPa.

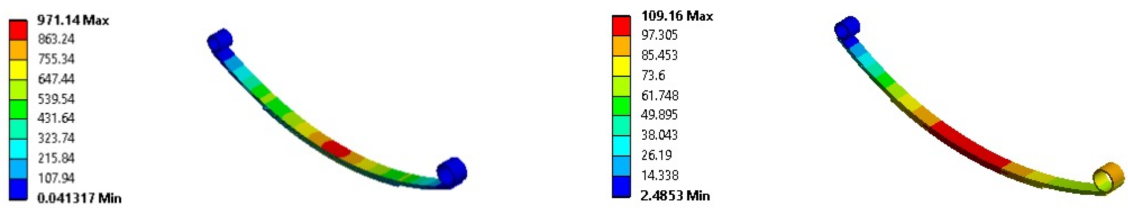


Figure 6.9: Stress and Deflection in Steel leaf spring

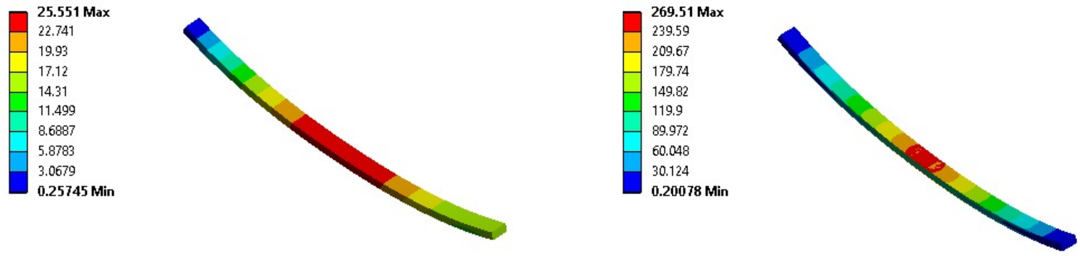


FIGURE 6.10: Stresses and Deflection in Kevlar/Epoxy Leaf spring

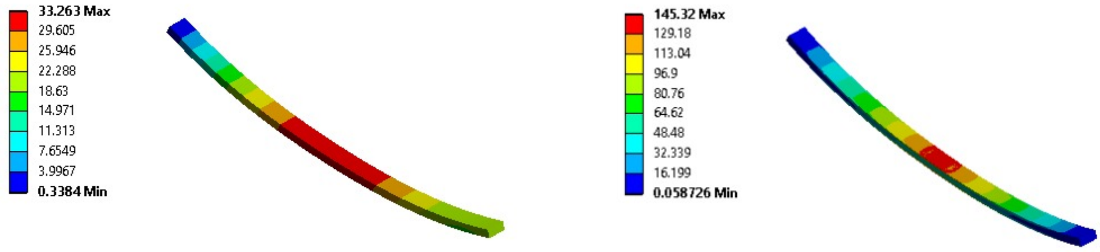


FIGURE 6.11: Stresses and Deflection in E-glass/Epoxy Leaf spring

The results of static analysis on composite and steel leaf spring design are validated and indicated that stresses are much lower in composite leaf spring as compared to conventional leaf spring. The composite leaf spring was optimized to decrease the weight of spring and can withstand external forces during operation. The stress failure criterion of Tsai-Wu was considered as limiting stresses.

However, as a result of static analysis the contours generated in both steel and composites leaf springs showed that the maximum bending stress was produced at the point of attached to the axle of the vehicle. As, leaf spring behaves like a simply supported beam therefore all the stress concentration is at the middle of the spring while shear is produced at the eyes of the spring. But in case of composite leaf springs interlaminar stresses are also present between each ply of the laminate. These interlaminar stresses are responsible for the failure of the model. Therefore, it is important to use appropriate stacking sequence for a design.

Chapter 7

Dynamic Analysis Of Leaf Spring

7.1 Introduction

The vibration characteristics of a structure is its intrinsic property and can be determined by using dynamic analysis. The vibrations induced in mechanical design involves high structural stresses due to resonance (large displacements) which results in its failure. Therefore, the study of dynamic behavior of mechanical design is considered to be of significant importance while designing mechanical components/structures. The fatigue failure also occurs due to vibrations induced in mechanical design which can be reduced by improving design characteristics i.e. by avoiding resonant frequencies. In vehicles, the vibrations are induced by road irregularities which causes discomfort to the passengers. The vibrations cause change in displacement of the system and amount of displacement depends upon the damping resistance of the system. The suspension system used in vehicles are designed efficiently to overcome the vibrations induced by road profile and give comfort to the riders. However, suspension systems have been improving with period of time to minimize the vibrations and to enhance the comfort level. In recent years, the improvements have been made to improve the suspension system of the vehicle and to reduce the noise of vibration with increased damping coefficient. Nevertheless, the interest of automobile industries has been shifted from steel leaf springs to composite leaf spring while taking the dynamic load of the vehicle into consideration. This chapter deals with the dynamic analysis of composite leaf spring. The effect of vibration induced by road profile and dynamic load produced on the leaf spring are also discussed here. Moreover, this chapter also includes the modal analysis of conventional and composite leaf spring.

7.2 Dynamic Load Calculation

The purpose of the dynamic analysis is to study dynamic loads induced in vehicles due to irregular road surfaces. The irregular surface of the road profile generates oscillations in the vehicle which is inconstant in space and time and depends upon the road class as defined by ISO standard[12] and length of the irregular road pavement. The dynamic force is produced in vehicle depends upon the type of the suspension system used, mass of the vehicle and speed with which vehicle translates through road pavement. The force induced in vehicle due to dynamic interaction between road pavement and vehicle can be defined by considering QCM (Quarter Car Model)[43]. The QCM for any vehicle can be modeled by considering the mass 'm' of the vehicle, stiffness constant 'k' and the damping coefficient 'c'. The schematic and free body diagram of QCM is shown in Fig.7.1 and its equations of motion are:

$$m_s.\ddot{z} = -C_s.(\dot{z} - \dot{y}) - k_s.(z - y) \tag{7.1}$$

$$m_u.\ddot{y} = -C_s.(\dot{y} - \dot{z}) - k_s.(y - z) - k_t.(y - z) \tag{7.2}$$

Where 'm_s' is Sprung Mass, 'm_u'is Un-sprung Mass, 'C_s' Damper, 'k_s' Spring Stiffness, 'k_t' is Tyre Stiffness.

The vibrations induced in the vehicle has sinusoidal variations i.e. N due to system oscillations which results in exchange of transient vertical force with the road. The symbol 'N' represents the dynamic overload amplitude and 'Ω' is the system pulse.

The dimensionless equation of sinusoidal variation is given in Ref [43] was used to predict the dynamic force induced in the model and can be written as:

$$\frac{|N|}{K_t H} = \Omega \sqrt{\frac{[m_s m_u \Omega^2 - k_s(m_s + m_u)] + c_s \Omega^2 (m_s + m_u)^2}{d^2(\Omega^2) + c_s^2 \Omega^2 e^2(\Omega^2)}} \tag{7.3}$$

Where

$$d(\Omega^2) = m_s.m_u\Omega^4 - [(k_t + k_s).m_s + k_s m_u]\Omega^2 + k_s.k_t e(\Omega^2) = k_t - (m_s + m_u).\Omega^2$$

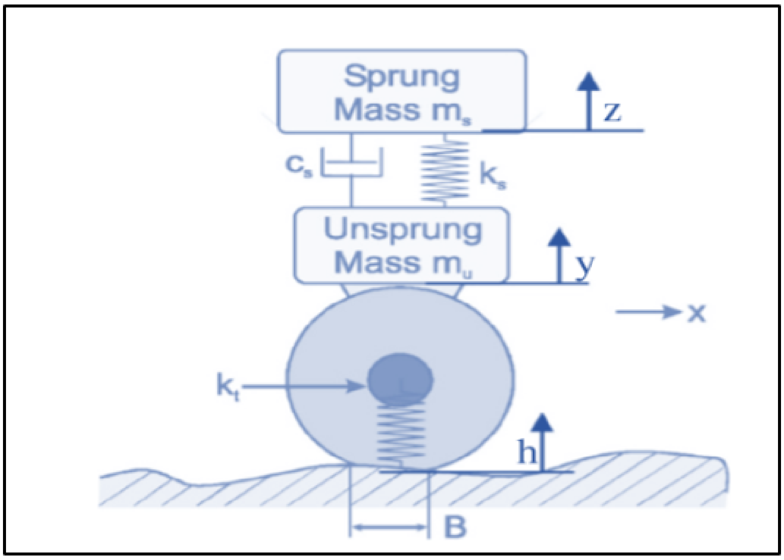


FIGURE 7.1: Quarter Car Model Schematic

7.2.1 Generation of Road Profile

According to ISO standard, the road profiles were classified into seven categories with corresponding road roughness value 'K' limits 9 and given in table 7.1 . The high roughness value 'k' represents high degree of damage of road surface and allow transit of vehicles at very low speeds. However, in this work the road profile for class C-D corresponds to roughness value k=6 was constructed with help of Matlab code for a section of total 250m long road pavement. The random generated road profile for ISO class C-D is shown in fig 7.2. The road surface profile of class C-D defines the average road condition with h_{max}=±50.

Table 7.1: ISO Road Roughness values

Uper Limit	Lower Llimit	K
A	B	3
B	C	4
C	D	5
D	E	6
E	F	7
F	G	8
G	H	9

After the road surface profile was generated for class C-D based on ISO standard, the next step is to predict the behavior of the vehicle under these road conditions. The vehicle moving at different speeds on the road come across surface irregularities due to road roughness and generate vertical oscillations in the vehicle through the interaction between the wheel of the vehicle and the road profile. The oscillations produced are dynamic in nature and causes the suspension system to absorb these dynamic loads. The dynamic force produces stresses in the vehicle resulting in discomfort to the passengers and also reduces the fatigue life of the suspension system and the body of the vehicle.

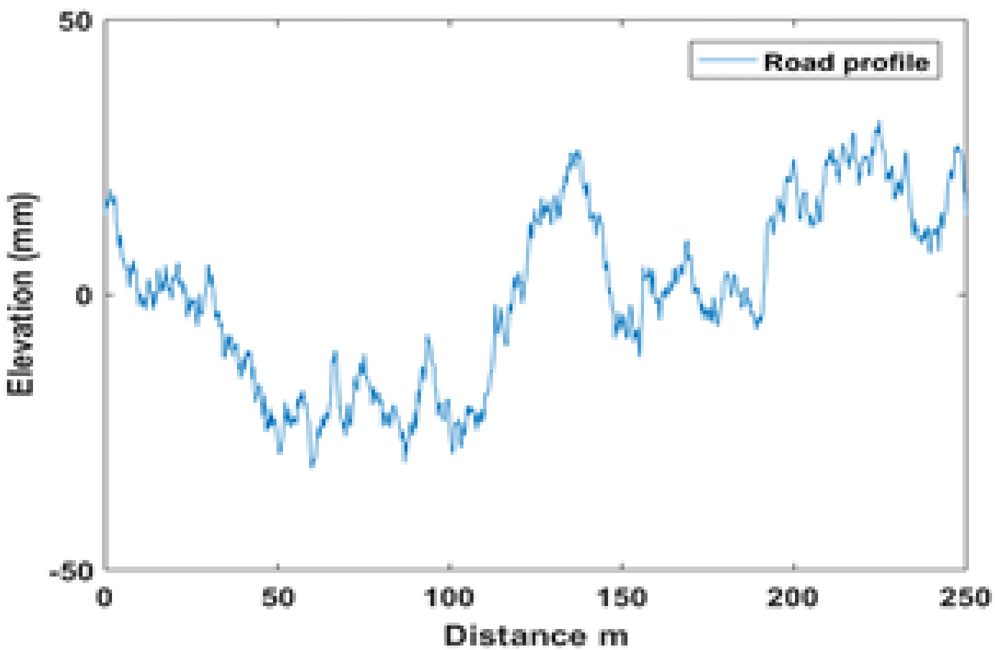


Figure 7.2: Road profile for ISO class C-D

7.3 Characteristics Of Vehicle

The vehicle characteristics are of significant importance in determining the stresses produced in vehicle by the dynamic forces and are given in table 7.2. The equations of motion determined analytically from the free body diagram of QCM and road profile graph for class C-D was used to determine the dynamic load produced in the vehicle for different speeds (20, 40, 60, 80 and 100 km/h). The Matlab® software was used to develop the code to determine the dynamic load for the vehicle travelling at different speeds on the generated road class profile.

TABLE 7.2: ISO Road Roughness values

Parameters	**Car**
Sprung mass ms [kg]	480
Unsprung mass mu [kg]	40
Spring stiffness ks [N/m]	21000
Spring damping cs [N.s/m]	1500
Tire stiffness kt [N/m]	146119

The dynamic load calculated analytically was used to determine the stresses produced by it in the vehicle suspension system. Thus, these dynamic loads were used to determine the stresses produced by it in leaf spring under various speeds of the vehicle. The QCM model was developed in MATLAB representing the actual model of the suspension system of the vehicle as shown in Fig 7.3.

The QCM model similar to the actual suspension system of the vehicle was used to analyze the behavior of the suspension springs of the system over an irregular surface. For this purpose, the road profile generated as discussed in previous section was used to examine the amount of stress induced in the spring under poor dynamic road conditions. The generation of stresses in leaf spring due to dynamic loads causes the spring to absorb energy transmitted from road to the axle of the vehicle. The amount of dynamic load depends upon the road class and the stresses produced in the suspension spring depends upon the speed of the vehicle. Both these factors are major concern for designing of leaf spring.

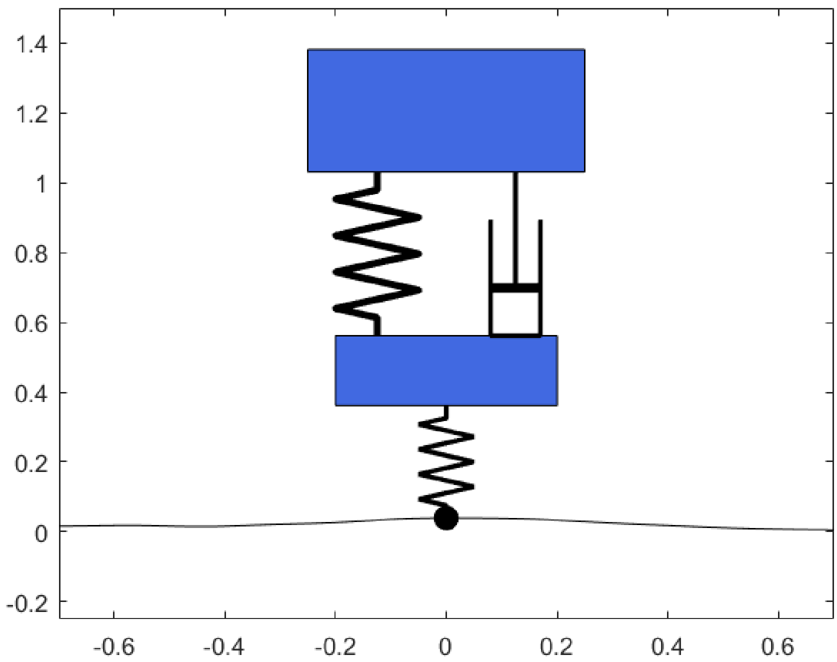


Figure 7.3: Matlab Quarter Car Model (QCM)

The dynamic load produced due to road irregularities causes the wear of the springs. Therefore, while designing the suspension springs, these parameters are also encountered from designer point of view. Although, there are several other factors including: material, type of loading, life, factor of safety, strength, stiffness, weight, camber height, width, length and thickness etc. to design the leaf spring for dynamic road conditions.

In present work, the leaf spring made of composite materials i.e. Kevlar/epoxy and E-glass/epoxy material were analyzed under dynamic loading conditions. The dynamic loads produced by the road due to motion of the wheel of the vehicle on irregular road surface was obtained as a function of time. In this work, the surface profile was generated and road profile velocity was calculated to perform the dynamic analysis. The road profile velocity at various speeds was considered to determine the maximum dynamic load that can be achieved from the road profile for road class C-D. The dynamic interaction of the wheel and the road was simulated in MATLAB.The simulation was conducted for the leaf spring designed from composite materials for Suzuki Mehran Car.

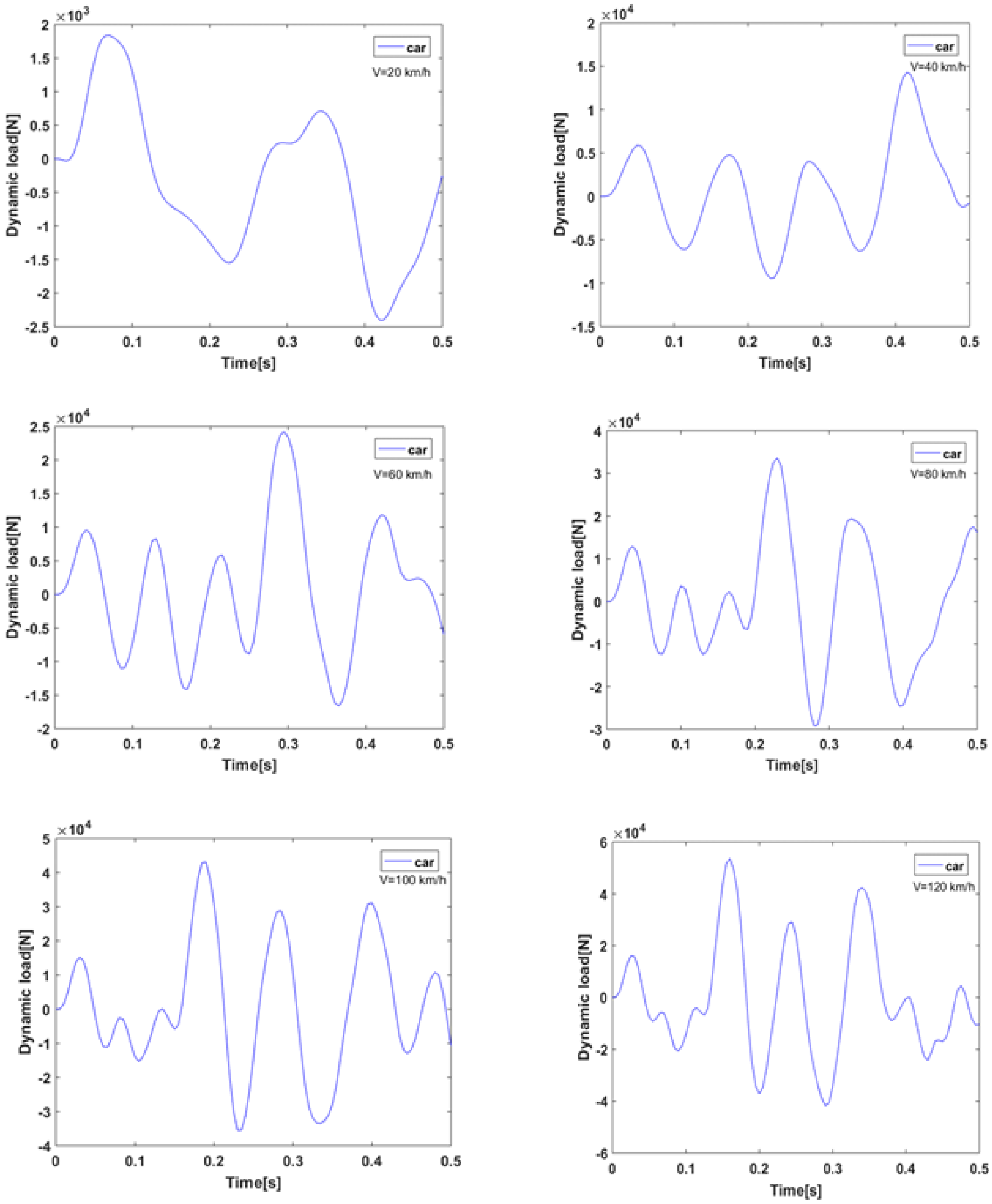


FIGURE 7.4: Dynamic load of a car at different speeds for ISO Road class C-D

The generation of dynamic load in the vehicle does not only depend upon the speed with the vehicle moves but also depends upon the road surface profile. Moreover, the dynamic interaction of vehicle with irregular road also generates resonance at various level of speeds. Thus, the suspension system of the vehicle is of major concern in dynamic analysis. Though, the dynamic load transmitted from road surface to the vehicle was determined analytically by taking the second derivative of vehicle road profile data along the road pavement and second equation of motion was applied by considering the sprung mass of the vehicle in to account. The dynamic loads obtained are used as input parameters in ANSYS to calculate the amount of stresses induced in composite leaf springs. The Fig.40 shows that an increase in the speed of the vehicle on an irregular road will increase the dynamic force transmitted to the vehicle. The stresses induced due to these loads are discussed in chapter 9.

7.4 Dynamic Analysis Results

The vertical loads on the rear axel of the vehicle are divided into two components i.e. static and dynamic components. The static component induces mainly due to the weight distribution of the vehicle on the axel whereas dynamic component occurs due to road irregularities that causes vertical oscillations in vehicle. The amount of vertical push due to road irregularities depends on the characteristic of the vehicle, road surface profile and speed of the vehicle. However, at relatively low speeds the vibrations induced due to static component is negligible. The road surface profiles are classified into eight classes according to ISO 8608 from class A to class H, depending upon the power spectral density associated with each class. The classification of road profiles along with their power spectral density are discussed in [ref vibrations induced by surface]. The ISO 8608 was used by assuming that road surface profile is periodic function of combination of large number of bumps with different amplitudes. However, according to ISO classification road profiles can also be generated artificially by the simple harmonic function as:

$$h(x) = \sum_{i=0}^{N} \sqrt{\triangle n}.2^k.10^{-3}.(\frac{n_0}{i.\triangle n}).\cos(2\pi.i.\triangle n.x + \varphi_i) \tag{7.4}$$

where: n_0 is spatial frequency, $\triangle n = 1/L$; $N = \frac{n_{max}}{\triangle n}$, $n_{max} = 1/B$; x is abscissa variable ranging from 0 to L;φ is phase angle with value ranging from 0 to 2π; k is constant. The value of 'k' depends on the road profile classification given by ISO standard.

The car model was analysed for the calculation of dynamic loads by solving QCM equations analytically and road profile was generated with the help of equation using MATLAB. The maximum dynamic load of 55864.84N and maximum acceleration of 114 m/s^2 are obtained at speed of 120 km/h. The stresses and deformation induced due to the application of dynamic loads at various speeds are simulated using ANSYS for composite leaf springs are shown in Fig 7.5 through Fig 7.8. The results indicated that as the speed of the vehicle increase relative to the road pavement, the stresses induced in the spring tends to increase. The maximum of 90000 cycles are applied for 0.5 seconds under dynamic loading conditions to evaluate the stresses at various speeds and it observed during simulation that Kevlar/Epoxy leaf spring can withstand the maximum number of cycles even at speed of 120 Km/h but in case of E-glass/Epoxy it was concluded that at relatively low speeds the spring can withstand maximum number of cycles for time period of 0.5 seconds. However, if the time period of simulation increases the stress produced in both composite leaf spring would decrease. It was observed from the results that Kevlar/Epoxy leaf spring can withstand maximum number of cycles for time period of 0.5 seconds whereas in case of E-glass/Epoxy leaf spring, the number of cycles applied varies as the speed of the vehicle tends to increase because E-glass/Epoxy have different strength parameters as compared to Kevlar/Epoxy leaf spring.

The strength of the composite leaf springs depends upon the fiber orientation and stacking sequence of the laminates. Thus, if the fiber orientation and stacking sequence changes the load bearing capacity of the spring changes accordingly. The stress produced in both Kevlar/Epoxy and E-glass/Epoxy leaf spring due to dynamic load at various speeds of the vehicle was found to be less than stress produced in steel leaf spring at same speed. It was also noted that number of cycles applied for steel leaf spring are less as compared to composite leaf springs.

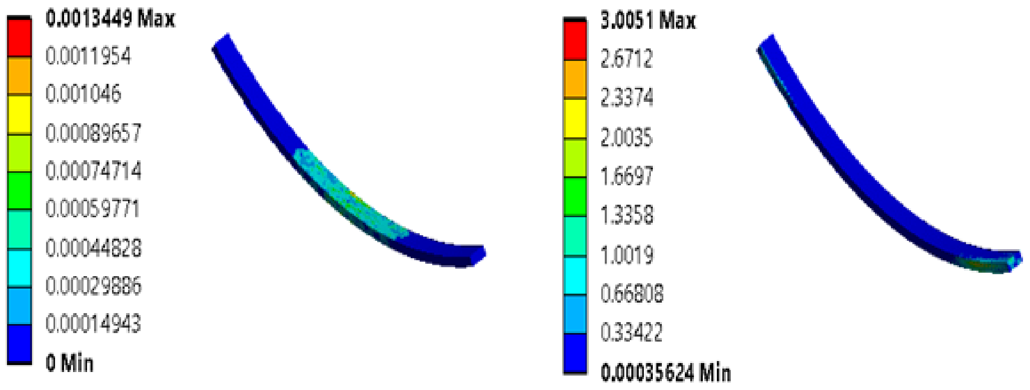


FIGURE 7.5: Stresses in E-glass epoxy spring under dynamic load at speed of 20 and 40 km/h

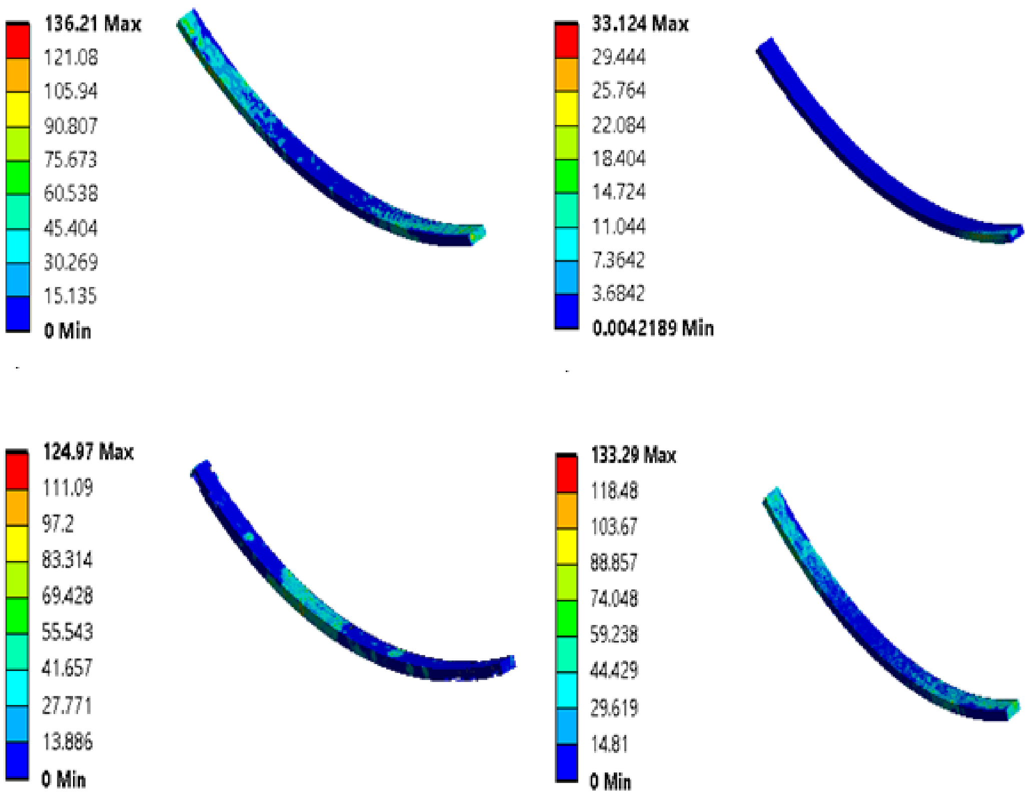


FIGURE 7.6: Stresses in E-glass epoxy spring under dynamic load at speed of 60, 80, 100 and 120 km/h

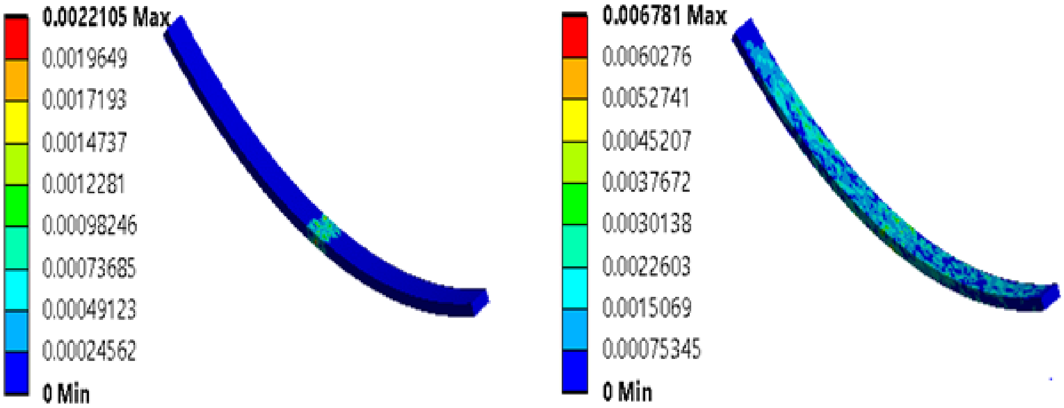


FIGURE 7.7: Stresses in Kevlar epoxy spring under dynamic load at speed of 20 and 40 km/h

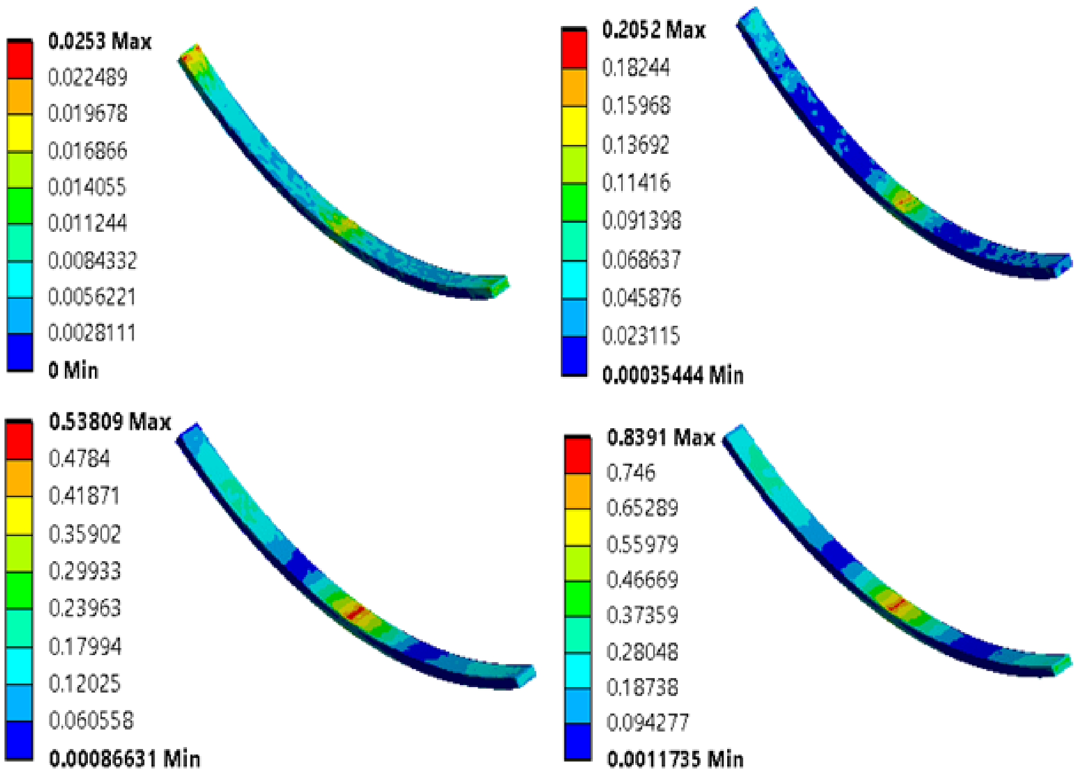


FIGURE 7.8: Stresses in Kevlar epoxy spring under dynamic load at speed of 60, 80, 100 and 120 km/h

7.5 Modal Analysis of Leaf Spring

The modal analysis does not include external loadings and torques but depends upon the equation of motions. The modes of structure are its intrinsic properties and depends upon the properties of the material and the sort of boundary conditions applied. The nature of the vibration in the structure varies, if the material properties (mass) and boundary conditions of the system changes. However, adding mass to the structure changes its modal frequency. The modal analysis was performed on both conventional and composite leaf spring for fixed-fixed type boundary condition and depicted in Table 7.3. The table shows the modal frequency for fixed-fixed type boundary conditions as leaf spring are fixed at both ends.

The actual boundary conditions were applied on leaf springs and their mode shapes were also determined. The mode shapes are the deformed pattern of the structure or model at different natural frequency. The deformation induced in the system against each mode shape can also be determined from modal analysis. However, the modal frequency varies from one mode shape to another. The frequencies at first six mode shape was determined for conventional and composite leaf springs for the actual boundary conditions and results are discussed here. The frequencies of composite leaf springs are always higher than steel leaf spring due to low weight of composites. It was obvious that, adding mass to the system will decrease its natural frequency. Therefore, both composite leaf springs are lesser in weight as compared to steel leaf spring so their natural frequency are higher than steel leaf spring.

Table 7.3: Modal analysis of steel and composite leaf spring for fixed-fixed type

		51CrV4	**Kevlar/Epoxy**	**E-glass/Epoxy**
Mode-1	Frequency (Hz)	80.75	136.36	89.45
Mode-2	Frequency (Hz)	197.45	211.52	133.56
Mode-3	Frequency (Hz)	235.66	403.13	259.75

7.6 Modal Analysis Results

The natural frequency is the inherent property of the material which varies with increase and decrease in the mass of the material. Thus, adding mass to the body will decrease its natural frequency. This means that body having low mass will vibrate more intensely as compared to body having greater mass.

Thus, leaf springs should have high natural frequency than maximum frequency caused by road indiscretions that was found to be 12 Hz to avoid resonance [68]. Therefore, in the present work the weight of composite leaf spring is less as compared to steel leaf spring, so it was assumed that the natural frequency of composite leaf spring is higher than steel leaf spring. In composite leaf springs the natural frequencies are higher than steel leaf spring. However, the natural frequency of Kevlar/epoxy leaf spring is greater than E-glass/epoxy and steel leaf spring which means that the Kevlar/epoxy leaf spring has greater life as compared to E-glass/epoxy leaf spring. Higher the natural frequency of the leaf spring means that spring can sustain low frequencies produced due to dynamic road surface. Therefore, composite leaf spring is considered as good replacement of steel leaf spring. The mode shape and total deformation produced in both composite leaf spring as shown in Fig.7.9 and Fig.7.10 was also determined to predict the deformed shape produced for each leaf spring at these natural frequencies. It was observed that deformation produced in Kevlar/epoxy leaf spring is slightly higher than E-glass/epoxy leaf spring.

Table 7.4: Shows the comparison between natural frequencies of steel and composite leaf spring

f(Hz)	**1**	**2**	**3**	**4**	**5**
Steel (51CrV4) leaf spring	39.1	119.1	156.3	260.6	430.2
Kevlar/Epoxy leaf spring	76.4	180.4	262.1	468.8	501.1
E-glass/Epoxy leaf spring	61.7	160.8	217.8	395.4	446.9

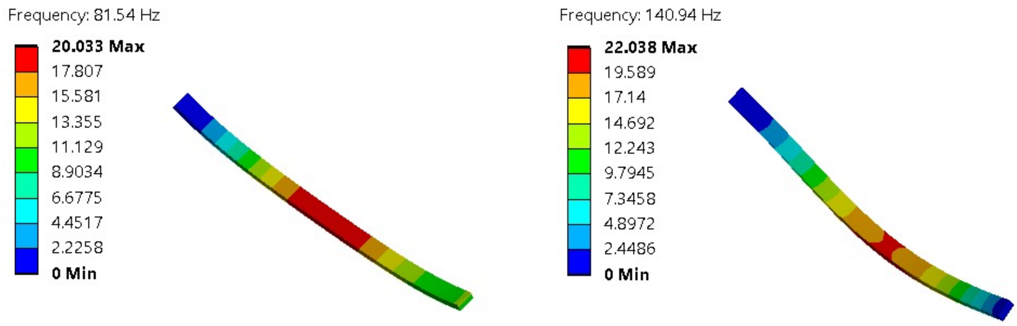


FIGURE 7.9: 1st and 2nd Mode shape of Kevlar epoxy Leaf Spring

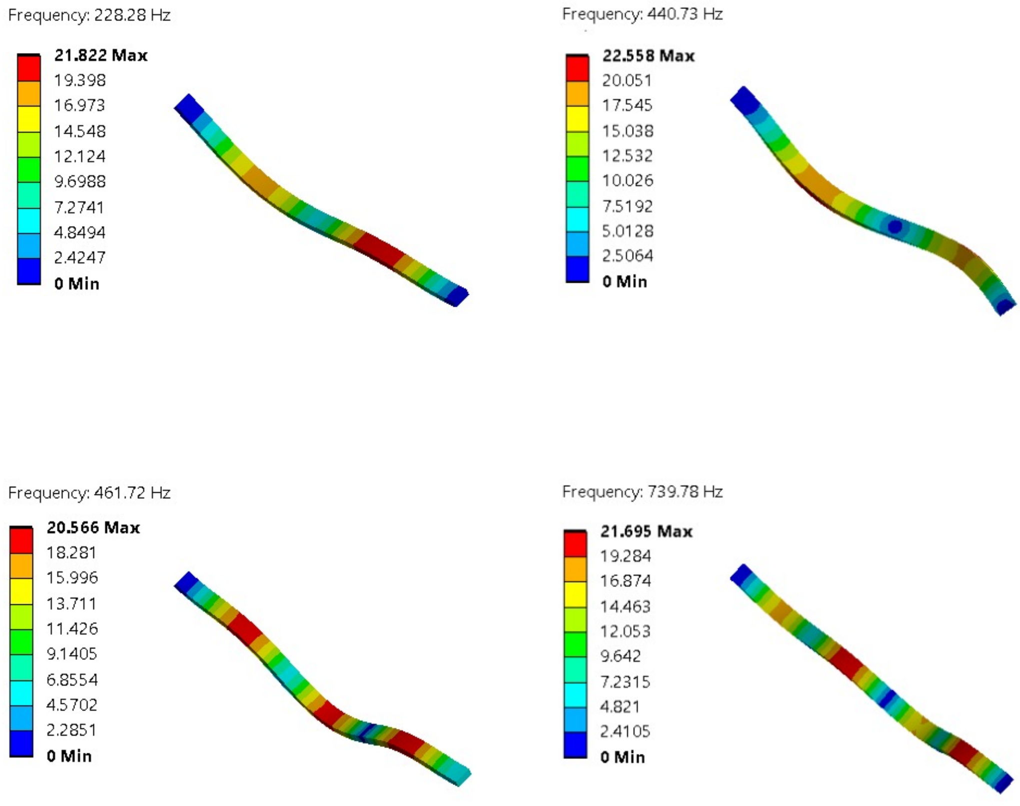


FIGURE 7.10: 3rd, 4th, 5th and 6th mode shape of Kevlar epoxy leaf spring

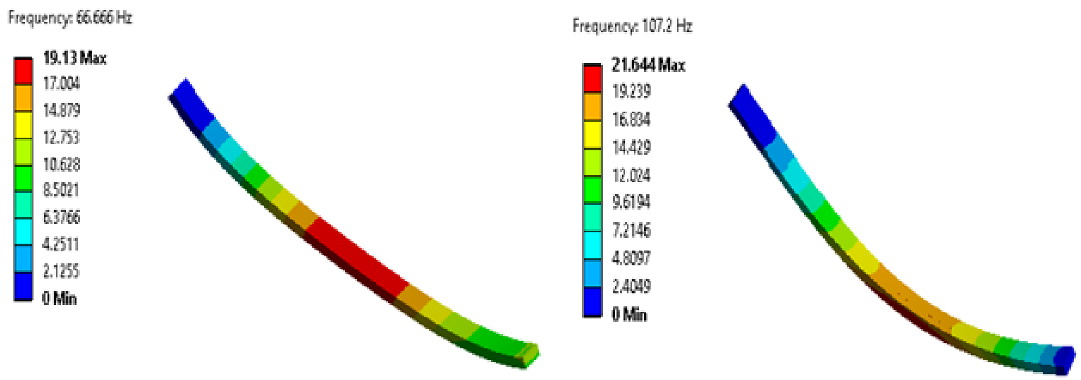


FIGURE 7.11: 1st and 2nd Mode shape of E-glass epoxy Leaf spring

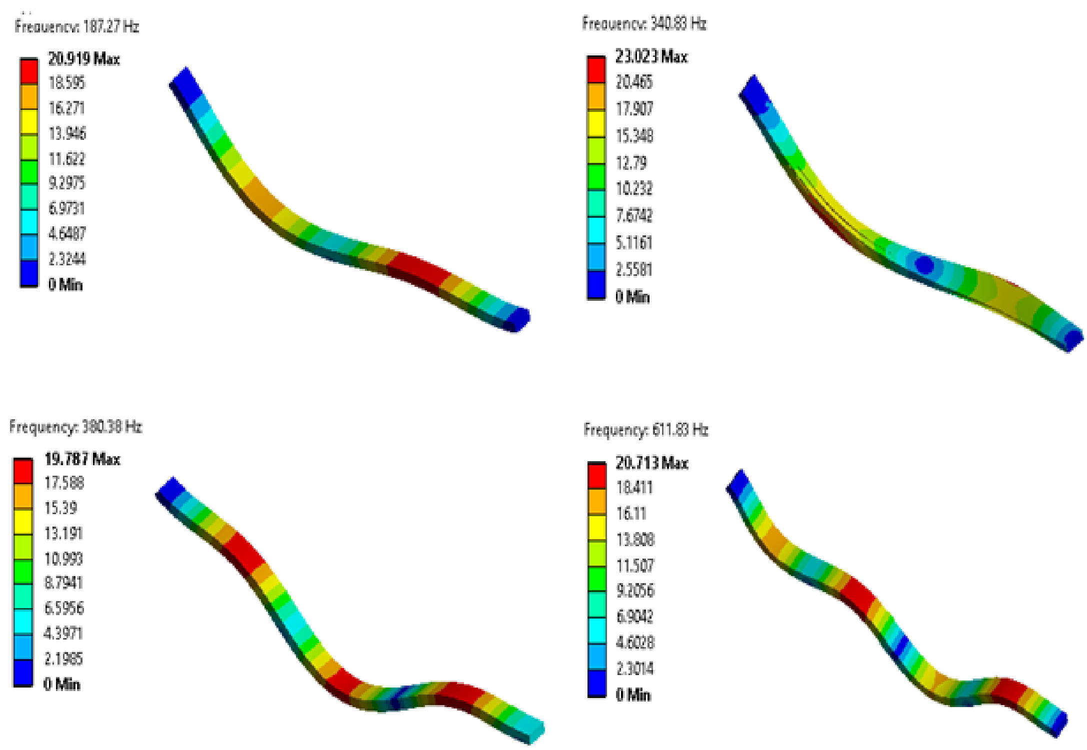


FIGURE 7.12: 3rd, 4th, 5th and 6th Mode shape of E-glass epoxy Leaf spring

7.7 Fatigue Life of Leaf Springs

The fatigue life of a structure is a major concern for the designer to predict the total life of the structure after which it tends to fail. Fatigue life is a critical design

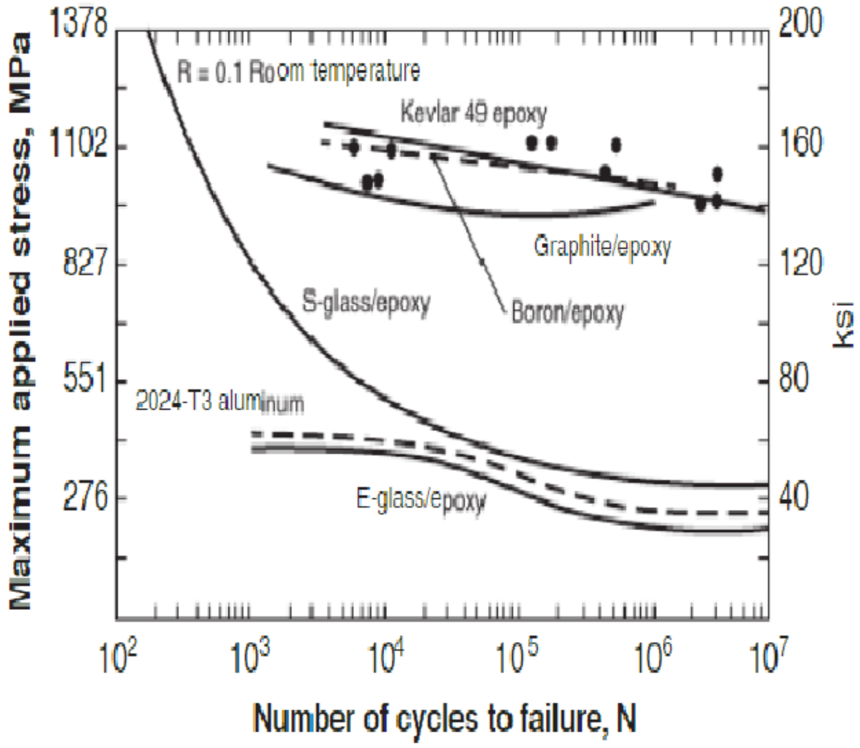
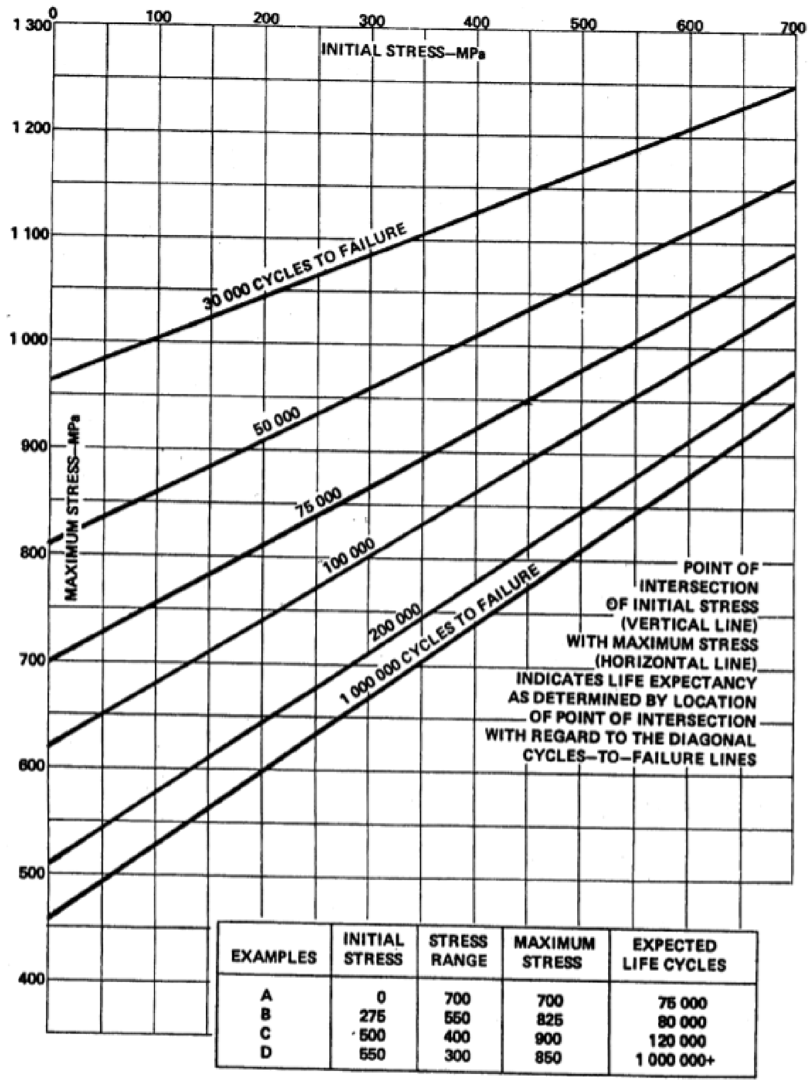


FIGURE 7.13: Fatigue life Composite Materials

parameter which helps in modeling the structure with an increased life. Therefore, it is necessary to determine the number of cycles of failure of the structure.

In this work the fatigue life of both leaf springs were determined analytically. An accelerated test procedure for calculating fatigue life of leaf springs is suggested by SAE for quick results. As per the procedure outlined in ref [66], the fatigue life of both steel and composite leaf spring was determined.

The initial stress of 490 MPa was obtained at initial deflection of 76 mm. The final maximum stress was found to be 971 MPa at deflection of 106 mm. The fatigue life of steel leaf spring under initial and maximum stresses were obtained as less than 10,00,000 cycles from S-N Curve given in Fig.7.13.



EXAMPLES	INITIAL STRESS	STRESS RANGE	MAXIMUM STRESS	EXPECTED LIFE CYCLES
A	0	700	700	75 000
B	275	550	825	80 000
C	500	400	900	120 000
D	550	300	850	1 000 000+

Figure 7.14: Fatigue life of Steel leaf Spring

However, the fatigue life of composite leaf springs was determined from the analytical relations developed by Hawng and Han and is given by the following equation:

$$N = B(1-r)^{\frac{1}{c}} \tag{7.5}$$

In the above equation, 'N' is the number of cycles of failure, 'r' is the applied stress level= σ_{max}/σ_{UT} , where 'σ'_{max} is the maximum stress, '$\sigma UT'$ is the ultimate tensile strength of the material. 'B' and 'C' are modeling parameters of the materials.

The fatigue life of composite leaf springs was determined at various stress levels given in Table.

TABLE 7.5: Fatigue life at various stress levels

Maximum Stress	**Applied Stress Level "r"**	**Applied Stress Level "r"**	**Number of cycles to failure**	
MPa	E-glass/epoxy	Kevlar/epoxy	E-glass/epoxy	Kevlar/epoxy
100	0.11	0.07	7519343	9982367
200	0.22	0.14	2932766	6750113
300	0.33	0.21	991086	4415239
400	0.44	0.29	275583	2588871
500	0.56	0.36	49292	1540702
600	0.67	0.43	6326	863362
700	0.78	0.5	350	448403
800	0.89	0.57	12	210941
900	1	0.64	-	86762
1000	-	0.71	-	29431
1100	-	0.79	-	5860
1200	-	0.86	-	771
1300	-	0.93	-	24
1400	-	1	-	-

The results tabulated shows that Kevlar/epoxy leaf spring has greater fatigue life than E-glass/epoxy leaf spring. These values also help in predicting the S-N curve for composite leaf spring and is given in Fig. The number of cycles at maximum applied stress level was also determined from S-N curve and are found to be greater than 40,00,000 cycles in case of E-glass/epoxy leaf spring and greater than 10,00,000 cycles for Kevlar/epoxy leaf spring under static loading conditions. In case of dynamic loading, the maximum number of cycles for E-glass/Epoxy reduced to more than 30,00,000 cycles and 8,00,000 cycles in case of Kevlar/Epoxy leaf spring.

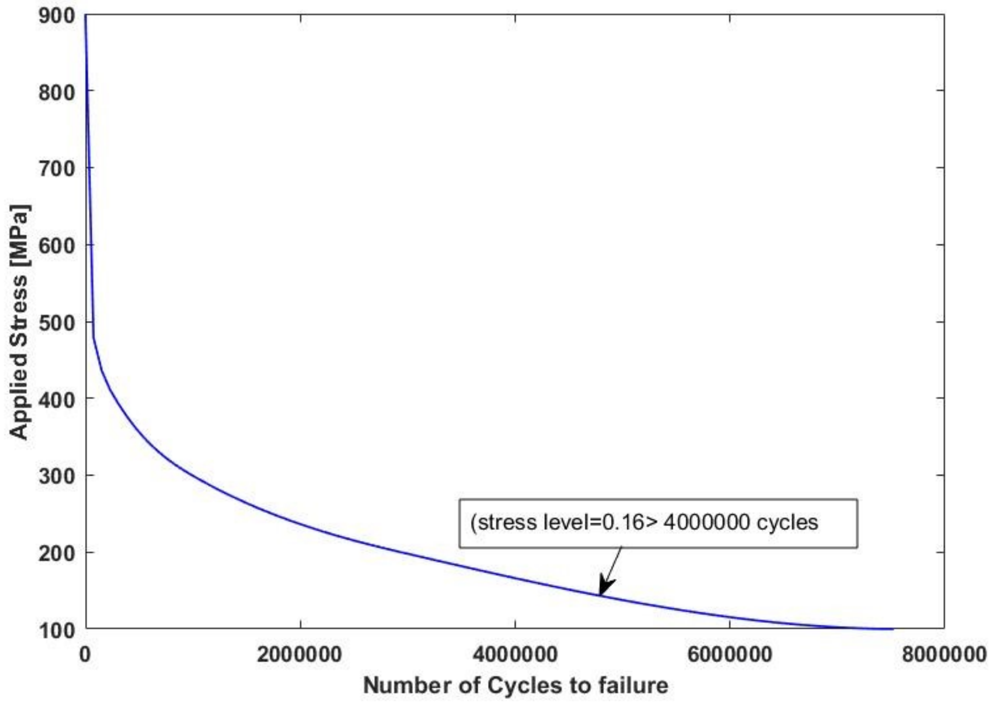


FIGURE 7.15: S-N Curve for E-glass/epoxy Leaf Spring

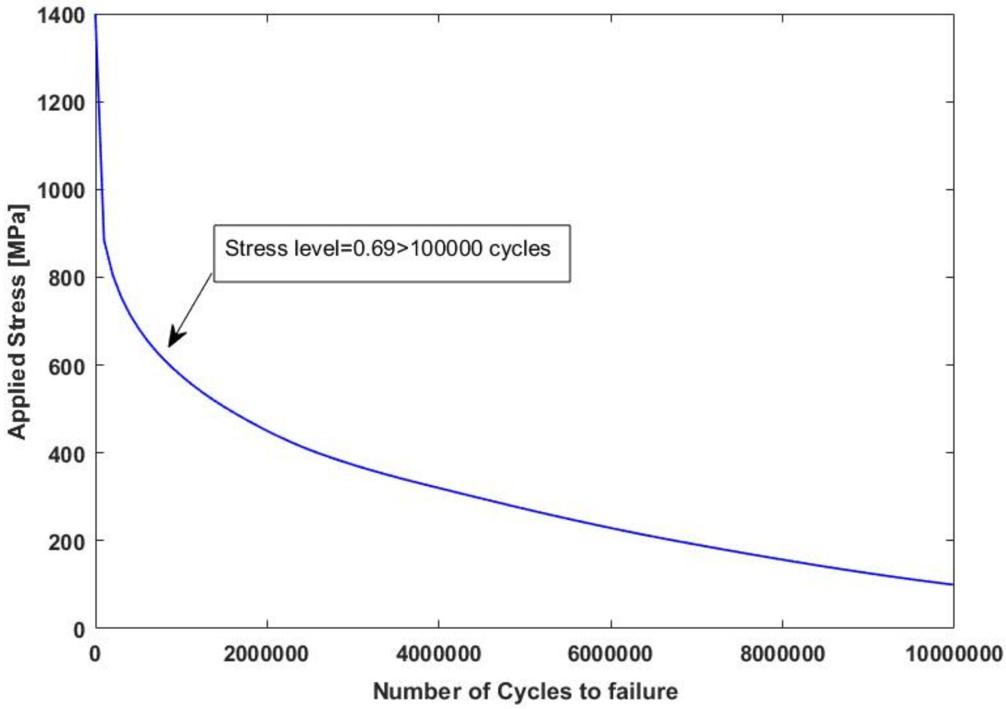


FIGURE 7.16: S-N Curve for Kevlar/epoxy Leaf Spring

Chapter 8

Weight Optimization Of Leaf Spring

8.1 Introduction

The ever-increasing demand of low cost of production with high demand of strength of engineering structures and components has prompted engineers to think in four dimensions to investigate for different methods of decision making such as engineering optimization. This method was developed to help engineers to improve the efficiency of the existing design systems and to improve the performance by introducing innovative methods. In engineering optimization, a candidate point from different design alternatives is selected based on the objective of optimization.

In recent years, optimal design is major concern for laminated composite materials as they are extensively used in all engineering fields. Some of the researches based on this area are discussed in refs [67],[68],[69],[70],[71]. The optimization of composite leaf spring is considered as an essential part in terms of optimal design, as composite leaf spring is single leaf design of same length and curvature as steel leaf spring. Therefore, to achieve optimum geometry, designer must verify different complicated solutions before concluding the leaf spring design. In actual engineering practice there are various types of methods which are used to solve different optimization problems. However, different design problems have different optimization methods to solve it efficiently. Hence, it's the discretion of an engineer to use appropriate, efficient and accurate method for his design problem. This chapter explains the procedure of optimization technique that was used to reduce the weight of composite leaf spring under static and dynamic loading condition.

8.2 Optimization

In present scenario the optimization of leaf spring is performed using ANSYS 19.0 software. The leaf spring was divided into front half and rear half-length for optimization, as it is considered as two cantilever beams according to SAE standard [72]. Each half of the leaf spring was divided into 11 segments as shown in figure 8.1. The finite element model of leaf spring was designed using SCDM 19.0 as a shell model. However, using shell elements, the computing time increases.

The design of experiments (DOE) was created based on the geometry parameters which are also called design variables of the optimization problem and response surface modelling was performed based on DOE. Further, the mass of the spring is introduced as objective function of the problem which needs to be reduced and it's the main goal of the optimization. The design variables, constraints and objective function of the optimization problem are discussed below.

i. Design variables

The quantities that are independently varied without manipulating other quantities are known as design variables. To achieve the optimum design, design variables must be varied accordingly. The length of spring and curvature are assumed to be same as steel leaf spring. Therefore, width and thickness along the spring are obviously considered as design variables. It is important to ensure the cross-section area to be equal along the entire length of the spring, so the fiber volume fraction would be constant along the spring length. The width and thickness at each point are considered as design variables and is illustrated by the following equation.

$$X_1 = [w_1, w_2, ..., w_i, .., w_n] \tag{8.1}$$

$$X_2 = [t_1, t_2, ..., t_i, .., t_n] \tag{8.2}$$

However, considering constant cross-section area and width along the spring reduces the number of design variables and conserve time. If 'A' is the cross-section area of the spring and 'wi' is the width of 'ith' cross-section, then the vector equation of design variables now takes the form

$$X_3 = [w_1, w_2, ..., w_i, .., w_n, A] \tag{8.3}$$

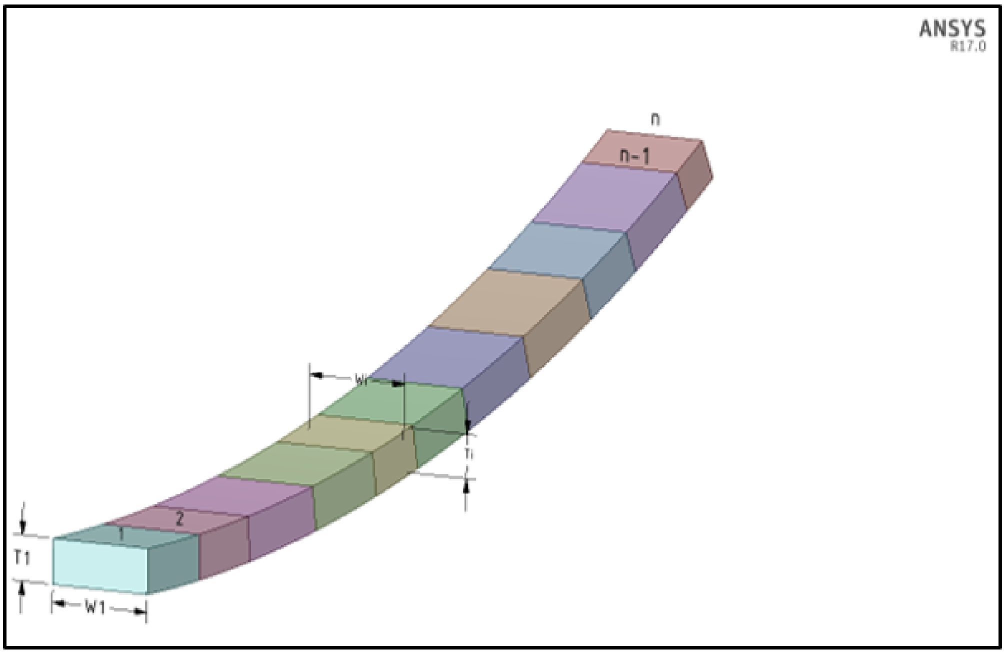


FIGURE 8.1: Leaf spring design variables

ii. Constraints

The variables which are function of design variables and cannot be varied independently are known as constraints. It limits the design within some specified values.

They may be equality or inequality constraints. In leaf spring design, stresses are considered as inequality constraints, which means that the maximum value of parameter is within the specified limits. In present scenario the leaf spring design is laminated composite in plane stress. So, Tsai-Wu failure theory was considered as most reliable and versatile failure criterion to evaluate inequality stress constraint and is depicted by following equation

$$F_{22}S^2\sigma_2^2 + F_1S\sigma_1 + F_6S^2\tau_{12} + F_2S\sigma_2 + F_{66}S^2\tau_{12}^2 + F_{11}S^2\sigma_1^2 + 2F_{12}S^2\sigma_1\sigma_2 \quad (8.4)$$

Where F's components can be found using the strength parameters, σ's are stress components and S is the factor of safety. If this equation is violated, then lamina will be considered as failed. The equality constraint on deflection states that the spring has maximum deflection under given load and is depicted by the following equation:

δ =Constant

The above equation refers to very closed feasible region due to which numerical errors can occur and convergence problems takes place during optimization. So, to converge this problem to global optimum point, two constraints δ_1, δ_2 are introduced and can be given as

$$\delta_1 \geq \delta - \epsilon \quad (8.5)$$

$$\delta_2 \leq \delta + \epsilon \quad (8.6)$$

Where δ is the deflection and $\delta$is infinitely small value. Hence, these equations refer 2ϵ as the effective feasible region and helps in smooth converging. It is important to add another constraint for the width of leaf spring from ends to the axle seat, which refers that the difference between two consecutive widths would be a positive value to overcome undesirable converging.

$$w_{i+1} - w_1 \geq 0 \tag{8.7}$$

where $i = 1, 2, 3, ..., n-1$

iii. Objective Function

The dependent variables that must be minimized or maximized to achieve the optimum point is known as objective function. In case of leaf spring, weight is the major concern in reducing the overall mass of the vehicle. Therefore, Composite materials are used in leaf spring to reduce its weight and provide high strength to it. In this work, weight of leaf spring is chosen as objective function and is illustrated below.

Minimize

$W = [w_1, w_2, ..., w_n, A]$

Subject to:

$$F_{22}S^2\sigma_2^2 + F_1S\sigma_1 + F_6S^2\tau_{12} + F_2S\sigma_2 + F_{66}S^2\tau_{12}^2 + F_{11}S^2\sigma_1^2 + 2F_{12}S^2\sigma_1\sigma_2 < 1 \tag{8.8}$$

$$\delta_1 \geq \delta - \epsilon \tag{8.9}$$

$$\delta_2 \leq \delta + \epsilon \tag{8.10}$$

$$w_{i+1} - w_1 \geq 0 \tag{8.11}$$

Initially the design of experiments (DOE) were used to create the sample of design space for both type of composite leaf spring using custom with sampling type for DOE to build a statistical model to predict out minimum weight and maximum stresses. The response surface was created using genetic algorithm for prediction purpose and also helpful in performing the sensitivity analysis to predict which variables are important to be designed to achieve the objective. The optimization was conducted using screening method for one hundred samples. The screening method includes the randomly generated sample of the design space and select the best ones. This method helps in evaluating more accurate results for the present work. However, there are other methods of optimization depending upon the complexity of the problem.

The result of optimization gives three candidate points for different values of masses and stresses for the optimized geometry and one of the best candidate points was selected on the basis of minimum weight and stresses. However, the minimum weight obtained for Kevlar/Epoxy and E-glass/Epoxy composite leaf spring without eyes has weight approximately equal to 338.7g and 366.8g respectively.

8.3 Joints Design

Most often, the eyes of composite leaf spring are designed separately, to carry out maximum load of the vehicle. The joint design is important in manufacturing of eyes because they are critical elements of leaf spring that can withstand the whole weight of vehicle. The type of joints used for the attachment of leaf spring to the body are shown in Fig8.2. Therefore, joints are properly designed to sustain static and dynamic loading. Entirely, joint types have their own advantages and disadvantages.

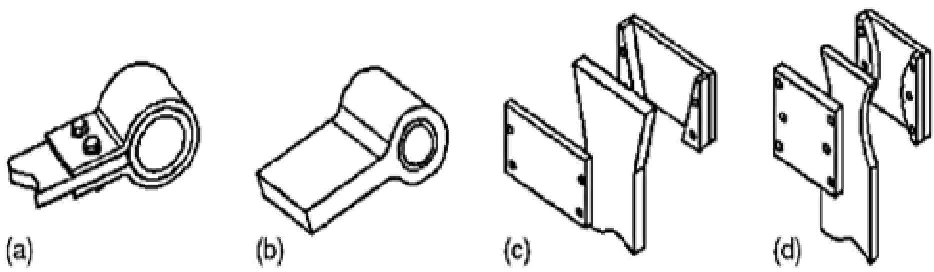


FIGURE 8.2: Joint types for composite leaf spring [37]

From the Fig.8.2 The steel eye joint type (a) can be attached to the spring either by drilling bolts or pins. The drilling action causes higher stress concentration near the holes which may led to the fracture of composite material. However, they can be connected easily and are less expensive. In case of (b) joint type, eyes of spring can be manufactured from the same composite material due to which the stress concentration does not exist. The material should be strengthened at the surface joining the eyes and the spring to avoid failure. The unidirectional fibers may cause splitting at the joining surface so necessary reinforcement is desirable. Though these joints are difficult to manufacture and involves high cost. The steel eyes with same profile can be mounted on the spring easily using these joints along with rubber pads. The drilling does not cause any stress concentration in these joints. The disadvantage of these joints involves the squeezing of the composite material due to available space. Therefore, a reliable angle of conical part should be desired to overcome this problem.

Low manufacturing cost and reduced weight of the leaf spring with desired strength is the major concern of the present work. So, keeping in view the aforementioned points steel eye joints was proved to be beneficial. Although, these joints are attached by drilling in composite spring and thus stress concentration in these areas was considered and analysed if it can fulfil the requirements of desired strength it would be used.

In present work the eyes of the leaf spring were analysed with additional lay up to strengthen the spring at the ends due to presence of side loads and longitudinal loads at the eyes of the spring. The failure in leaf spring may occur due to shear and tension among other failure modes. However, in this work unidirectional composite was considered wherein failure occurs due to splitting of unidirectional fibers. So, it is necessary to provide strength at the eyes of the spring using more compact lay-up sequence. The dimensions at the ends are assumed to be 50 mm wide, 50 mm long and 21 mm thick with 6 mm hole diameter for the attachment of steel eyes. The layup of [$\pm$ 45/0/90/$\pm$45/0/90/$\pm$45]2 is considered at the ends of the spring to provide necessary strength for both composite leaf spring.

In actual practice, the ends of the leaf spring become nearly flat under loading. Thus, the only forces that are acting on the eyes of the spring are side forces and longitudinal forces. The results indicate that the maximum stresses obtained in longitudinal direction, transverse direction and in xy plane are found to be less than one for both composite leaf springs as depicted by Tsai-Wu failure criteria. Therefore, critical loading acted on the eyes can be supported by the composite leaf spring. The maximum stresses under critical loading are found to be σ_{xx}= 13.47 MPa, σ_{yy}=15.33 MPa and σ_{xy}= 13.06 MPa for E-glass epoxy end joint and σ_{xx}= 16.36 MPa, σ_{yy}=17.8 MPa and σ_{xy}= 16.74 MPa for Kevlar epoxy leaf spring joint.

8.4 Results of Optimization

The results of optimization indicate that thickness increases linearly while the width of the spring decreases with the same trend from the ends towards the axle seat along the length of the spring, as shown in Fig 9.12 through fig 9.16. The same trend was observed for E-glass/epoxy leaf spring but the graph of thickness and width are steeper compare to Kevlar/epoxy leaf spring.

The longitudinal stresses are calculated along the length of the composite springs. It was observed that the stresses at the axle seat are higher and then decrease towards the end of the spring. The maximum longitudinal stresses at the axle seat was found to be 178.31 MPa and then decreases towards the ends of the Kevlar/Epoxy leaf spring whereas the longitudinal stress at the front half end and rear half end was found to be 5.98 MPa and 14.55 MPa because the length of front and rear half is dissimilar from each other. The maximum longitudinal stress in case of E-glass/Epoxy leaf spring was found to be 126.06 MPa at the axle seat whereas minimum longitudinal stress at the front half end and rear half end was found to be 8.90 MPa and 9.03 MPa.

The deflection along the length of the front and rear half of both composite leaf springs are also determined and found that maximum deflection occurs at the eyes of the spring. Thus, deflection at the front and rear eye of the Kevlar/Epoxy leaf spring are found to be 103.2 mm and 94.59 mm respectively. Moreover, deflection at the front and rear eye of the E-glass/Epoxy leaf spring are found to be 124.05 mm and 111.31 mm respectively.

In optimization, both composite leaf springs are divided into front and rear half as already discussed. Therefore, the longitudinal stresses produced at the middle of the full-length spring is approximately the sum of the stresses produced at the axle seat of both the front and rear half of both composite leaf springs. The maximum shear is produced at the eyes of the spring and decreases near the axle seat whereas bending stresses was found to be maximum near the axle seat.

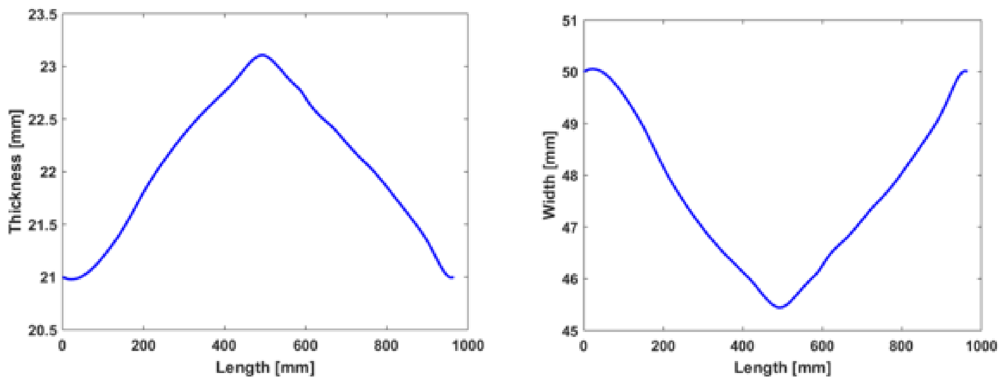


FIGURE 8.3: Variation of thickness and width along the length of optimized Kevlar/Epoxy spring

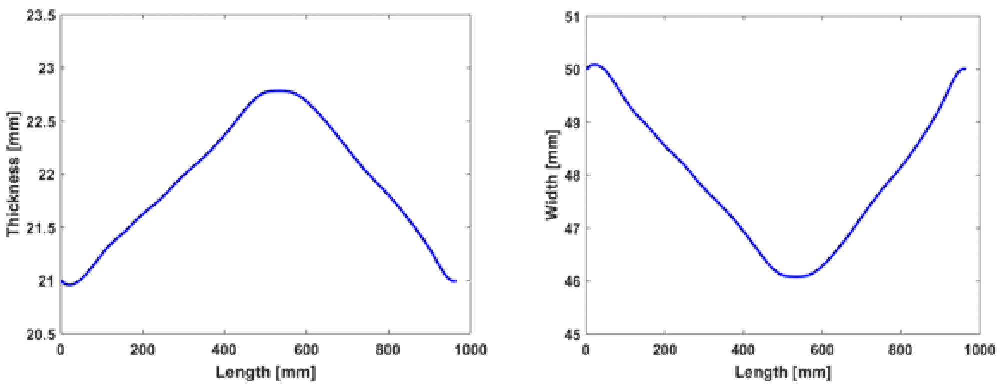


FIGURE 8.4: Variation of thickness and width along the length of optimized E-glass/Epoxy spring

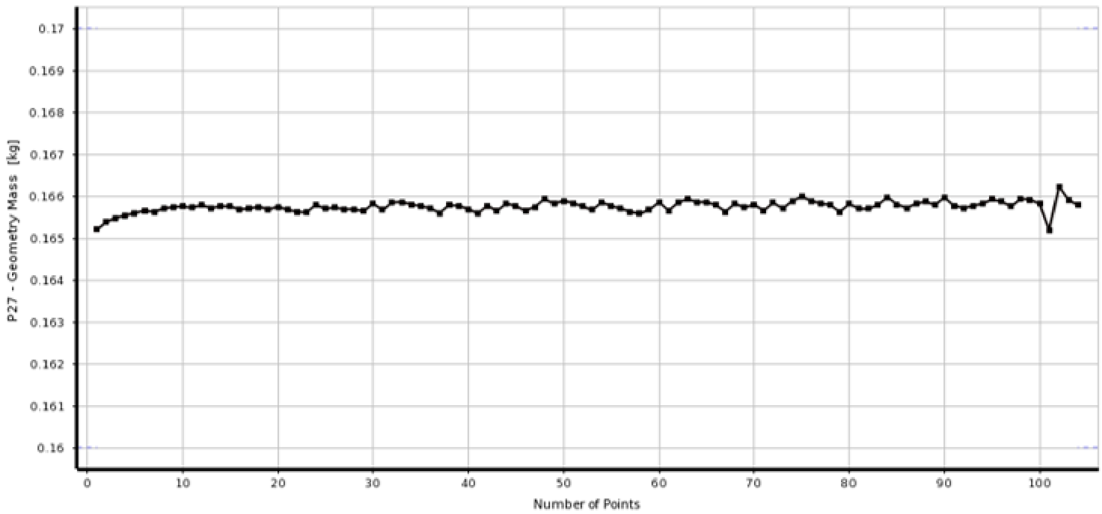


FIGURE 8.5: Variation of Mass with iteration number

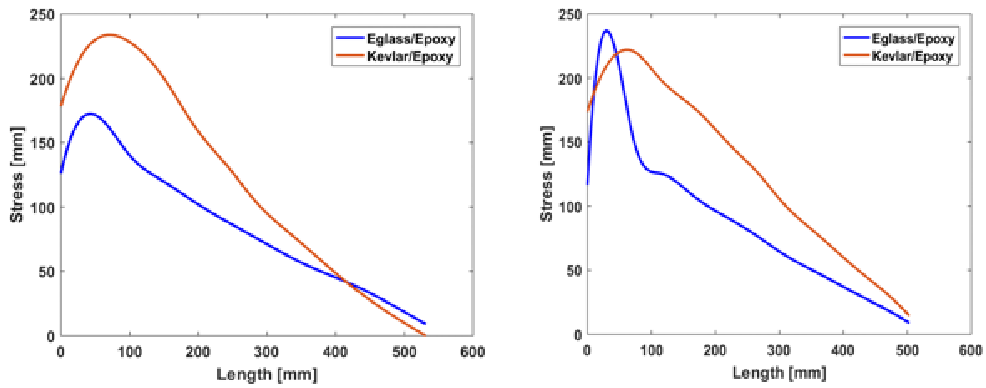


FIGURE 8.6: Variation of longitudinal stresses at the front and rear half along the optimized E-glass/Epoxy spring

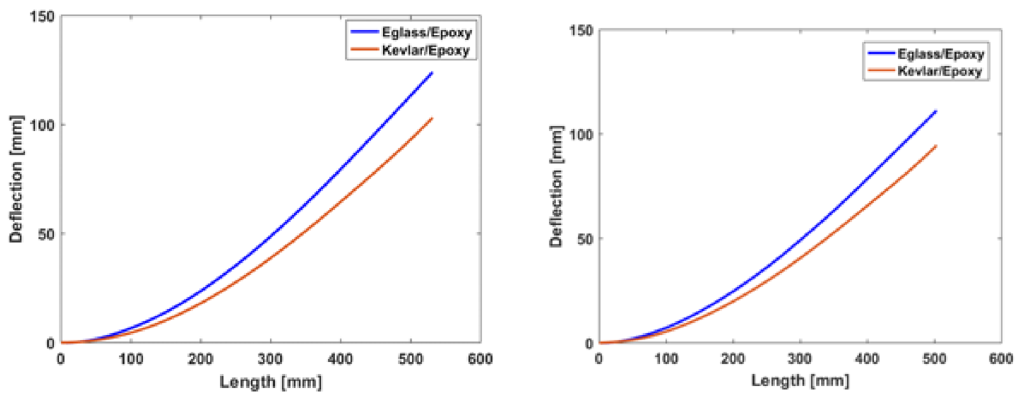


FIGURE 8.7: Deflection of Front-Rear half of Kevlar/Epoxy and E-glass/Epoxy Leaf spring

After optimization the analytical analysis was performed on both front and rear half of both Kevlar/epoxy and E-glass/epoxy leaf spring. Analytical analysis is the validation of numerical results to ensure the validity of a model being analyzed. Analytical relations and MATLAB were used to compute stresses using the criteria of Tsai-Wu failure in each ply of leaf spring and was compared with numerical results. The stresses on the bottom of each ply was compared with theoretical results to ensure the validity of model. However, the maximum numerical value of stresses produced on top and bottom of each ply is found to 262 MPa and 263 MPa in Kevlar/epoxy leaf spring respectively while 148 MPa and 149 MPa was found at top and bottom of each ply in E-glass/epoxy composite leaf spring.

The value of stresses obtained from analytical result was found to be in good approximation with numerical result. The difference between numerical and analytical values for both Kevlar/epoxy and E-glass/epoxy leaf spring was found to be 0.76 % and 0.73% respectively. The percentage change between these values are negligible and acceptable.

Chapter 9

Results And Discussion

9.1 Comparison of Conventional and Composite Leaf Spring

A comparative study was conducted for conventional and composite leaf springs on the basis of strength and weight of the spring. The stresses obtained from numerical analysis of conventional and composite leaf springs were compared and tabulated in Table 9.2. It was observed that in conventional leaf spring the stresses are found to be higher as compared to composite leaf spring under same loading capacity. The stresses obtained from the numerical analysis of leaf springs are within the specified limit of the material used. The comparison of conventional and composite leaf springs is shown in Fig. 9.17 with the help of bar graph to properly demonstrate the difference between stresses and weight among both leaf springs.

It was already known that the density of composites is much lesser and specific strain energy of composite materials are higher than metallic materials. Therefore, it was concluded that the composite leaf springs have better tendency to absorb dynamic loads and greater number of cycles of failure than conventional leaf spring.

TABLE 9.1: Comparison of Steel and Composite leaf spring

Parameter	**Steel leaf spring**	**Kevlar/epoxy leaf spring**	**E-glass/epoxy leaf spring**
Load (N)	4900	4900	4900
Bending stress (MPa)	965.2	268.51	145.32

The results indicated that the stresses produced in case of Kevlar/Epoxy leaf spring is 72.18 % and 84.9 % in case of E-glass/Epoxy. Also, the weight reduction in case of Kevlar/Epoxy was found to be 89.1% and 88.2% in case of E-glass/Epoxy leaf spring.

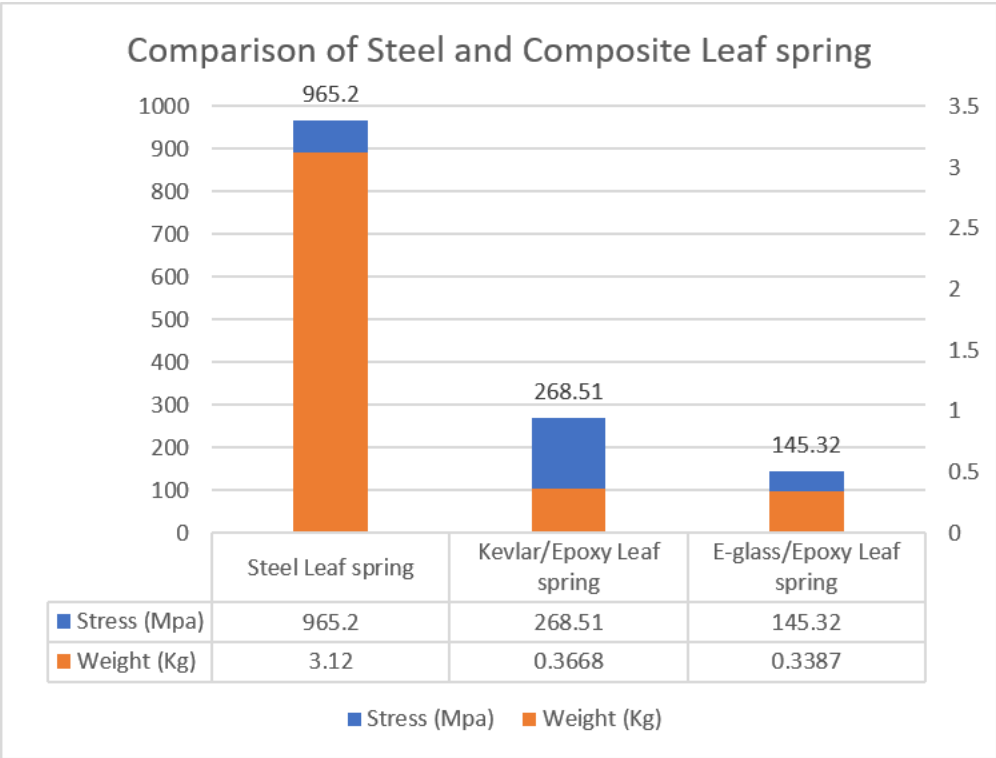


FIGURE 9.1: Comparison of Steel and Composite Leaf Spring

The figure 9.17 shows the comparison of the stresses and weight of the composite and steel leaf spring under same loading and boundary conditions. It can be observed that the stresses produced in steel leaf spring are higher than E-glass/epoxy and Kevlar/epoxy leaf spring, whereas the stresses produced in E-glass/epoxy are minimum than rest of the two leaf springs. But it cannot be predicted from the graph of static analysis that the spring with minimum stresses can have a greater life. Therefore, dynamic analysis is needed to determine the dynamic behavior of the spring under maximum loading conditions. It was also observed that weight of the optimized Kevlar/epoxy leaf spring have minimum weight than other two leaf spring and fulfil the criteria of weight optimization. It may be noted that weight and stresses produced in steel leaf spring are higher than composite leaf spring. Moreover, it was observed from the result of dynamic analysis that Kevlar/epoxy have higher tendency to sustain the dynamic loadings than steel and E-glass/epoxy leaf spring.

9.2 Conclusion

The laminated quasi-isotropic composite Kevlar/epoxy and E-glass/epoxy was used for leaf spring static and dynamic stress analysis. The results indicated that the fiber orientation and stacking sequence plays a vital role to enhance the strength of material. It was studied that 45° fiber orientation is required to obtain the desired strength and was considered in present work, whereas failure in 0° and 90° fiber orientation occurred due to fiber fracture and matrix failure respectively. The stresses in composite leaf springs were found to be much lower than steel leaf spring with substantial weight savings. It was observed that shear stresses produced are also within the specified limits of the material. The deformation in composite leaf spring is less than specified camber range and increase in load shows a linear variation in stresses. Since, optimization of composite leaf spring increases the load bearing tendency of spring and reduction in weight. Therefore, optimization of spring using first order method is considered due to ease of convergence.

In present work, Kevlar/Epoxy composite leaf spring replaced the conventional leaf spring due to its high strength to weight ratio and stresses reduction in composite leaf spring was observed. The inconsiderable amount of weight reduction of 89.1% and 88.2% was achieved by use of Kevlar/Epoxy and E-glass/Epoxy composite material respectively. The total deformation was also less as compared to steel leaf spring. Therefore, the use of composite leaf spring proves to be economical. Moreover, design optimization was achieved to model more compact system that have reduced stresses and weight to provide better fuel economy.

Kevlar/Epoxy is found to be better than conventional leaf spring. So, it is considered as the replacement of steel leaf spring.

9.3 Future Recommendations

1. Composite leaf springs as analyzed in this research can be fabricated with same properties, dimensions and compare experimental work with numerical and analytical results already obtained from this research.

2. Unidirectional composite with ply angle of $0°$ can be analyzed under same loading capacity as obtained for Suzuki Mehran Car.

3. Buckling analysis and impact load test of leaf spring can be performed with different composite materials.

4. The effect of ply angle and several combinations for composites can be studied and analyzed.

5. The fabrication of composite leaf spring using neural network can be performed.

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