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BAHIRDAR INSTITUTE OF TECHNOLOGY

FACULTY OF MECHANICAL AND INDUSTRIAL ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

Design and Fatigue Life Prediction of Composite Leaf Spring in Automobile

A Thesis Submitted to the Graduate School of Bahir Dar University in Partial Fulfillment of the Requirements for the Degree of Masters of Science

In

Mechanical Engineering

(Mechanical Design)

By:

Desalegn Azmeraw

Advisor:

Dr. Ermias G/Kidan

June, 2017

Declaration

I, the under signed, declare that this thesis work is my original work and has not been presented for a degree in any other university, and that all sources of material are duly acknowledged.

Desalegn Azmeraw		03/06/2010
Name	Signature	Date

Title of thesis: Design and Fatigue Life Prediction of Composite Leaf Spring in Automobile.

This thesis has been presented for examination with my approval a university advisor.

Dr. Ermias G/Kidan		03/06/2010
Advisor	Signature	Date

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Abstract

Leaf spring is widely used in automobiles and also one of the components of suspension system. It needs to have excellent fatigue life and less weight. As a general rule, the leaf spring must be regarded as a safety component as failure could lead to severe accidents. An attempt is made in this paper to study improve fatigue life of single composite spring with geometry optimization. The modeling of the original and optimized leaf spring is created using SOLIDWORK Software and was imported to ANSYS software for analysis. Finite element analysis (FEA) is performed to obtain maximum stress point or dangerous area, and life of original and optimized leaf spring using the ANSYS software with applying the boundary conditions. The maximum stress appears at eye end segment of the leaf spring. The FE model of the leaf spring geometry is meshed with tetrahedral elements. Mesh refinement are done on the leaf spring, so that fine mesh is obtained on the eye ends of spring, which are generally critical locations on leaf spring. The failure in the leaf spring initiated at the eye end segment of the spring, and fatigue is the dominant mechanism of failure.

Geometry optimization resulted in 57% stress reduction and life is optimized by 89% of composite leaf spring, which was achieved by material changing. And weight of E-glass/Epoxy composite leaf spring is reduced by 84.4%. Then the results of Von-Mises stress, shear stress, weight, and life of leaf spring is done using ANSYS software results. From result of geometry optimization parameter like material changing and design optimization are changes in model of leaf spring improvement in fatigue life.

Keywords: *Composite Material, FEA, Leaf spring failure, Leaf spring stress analysis*

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Nomenclatures

LWV	Light weight vehicle
W	Weight
F	Applied load
L	Span length
b	Width
h	Thickness
I	Moment of inertia
σ_{max}	Maximum stress
δ_{max}	Maximum deflection
σ_1	Maximum principal stress
σ_2	Minimum principal stress
τ_{12}	Principal shear stress
FEA	Finite element analysis
3D	Three-dimensional
FEM	Finite element method
SAE	Society of Automotive Engineers
AISI	American Iron and Steel Institute
PRO/E	Pro Engineer
m	Mass
FRP	Fibre reinforced plastics
GFRP	Glass fibre reinforced plastics
S	Strain energy
E	Modulus of elasticity

σ_t	Allowable stress
ρ	Density
N	Number of cycles to failure
r	Applied stress level
σ_u	Ultimate tensile strength
n	Number of leaves
FS	Factor of safety
σ_b	Bending stress
M	Bending moment
I	Moment of inertia
τ_{12}	Shear stress
ε	Strain
C	Compliance
DOF	Degree of freedom
v	Nodal displacement
θ	Rotational displacement
B	Strain matrix
k_e	Element stiffness
W_{Gr}	Gross weight of the vehicle
2D	Two dimensional

CHAPTER ONE

1. INTRODUCTION

1.1. Background of the Research

Today automotive manufacturers are faced with several complex challenges. In a highly competitive market, customers are demanding more for their money. Motorists wish cars that propose high performance, comfort, refinement, safety as well as increased vehicle customization. The automotive industry is also faced with governments who are consistently introducing legislation that demand improvements in fuel efficiency, reduced emissions, increased recycling and greater safety for both pedestrians and occupants. The circumstances facing the auto industry is most excellently summarized by quoting an article in the Poly motive magazine [1] "Far-reaching efforts to achieve components that are rigid, strong, safe and at the same time, as light as possible are needed in order to survive in automotive manufacturing".

In order to preserve natural resources and cost effects, weight minimization has been the major focus of automotive industries. Now a day's weight minimization can be achieved generally by the replacement of better material, design optimization and enhanced manufacturing process. Springs are important suspension essentials on any vehicle, essentially to reduce the vertical vibrations, impacts and bumps due to road abnormalities and made a comfortable ride. The leaf spring suspension holds about 10-20% of vehicle unsprung mass. Thus, it becomes an essential component for weight minimization [4]. The mass minimization can be accomplished by selecting better materials and optimized design of leaf spring [7-9].

Figure 1.1: Leaf Spring

There are various types of springs available for suspension system. A leaf spring can be considered as the simple type of spring, normally used for suspension system in vehicles.

It's generally like a slender arc-shaped having some length of a steel spring of rectangular cross-section. The axle is placed at the center of the arc, at the end eyes are used for attaching to the vehicle body. From the time 1970s leaf springs were very general on automotive. The key characteristic that gives the smoothness of a vehicle is its suspension. Now days extensively used suspension systems in automotive are the Leaf springs. It is also called as a semi-elliptical spring or cart spring, which is similar to an arc-shaped length of a steel spring with a rectangular cross-section. We can fasten a leaf spring directly at both ends (eyes) of the frame or directly to the one end usually the front end, whereas the other end is attached with the shackle, a short swinging arm. For the smooth riding in very heavy vehicles, a leaf spring prepared out of multiple leaves in multiple layers stacking at the top of each other often started with gradually shorter leaves is used to provide ease in riding in lighter weight vehicles as shown in figure below.

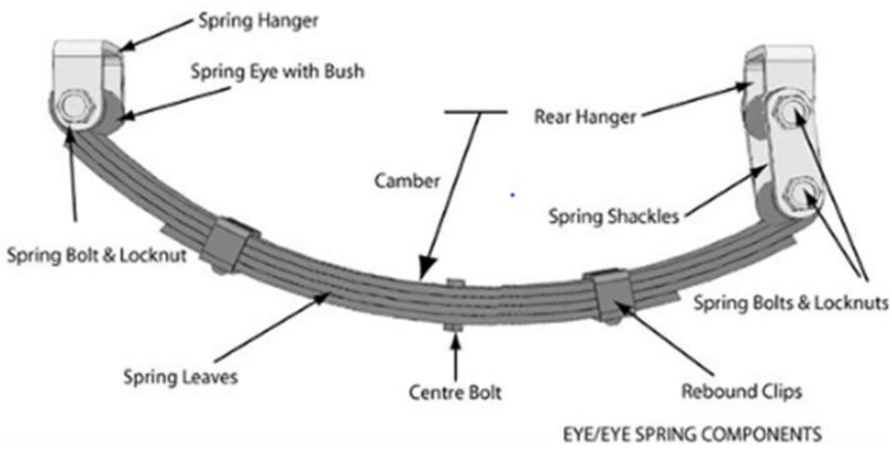


Figure 1.2: Components of leaf spring

The automotive manufacturer tends to enhance soothe of user and achieve appropriate stability of comfort riding virtues and economy. The researchers are very fascinated in the replacement of steel leaf spring by some composite leaf spring because of high strength to weight ratio. On the other hand, there is a restriction for the amount of applied loads in springs. The amplification in applied load creates complexity at geometrical arrangement of vehicle height and erodes other parts of vehicle.

So, springs design in concerned of strength and toughness is enormously significant. Minimization of spring mass is also key parameter in enhancement of car dynamic. By substitution of steel leaf spring with composite leaf spring will minimize spring mass in addition to resistance increase under the effect of applied loads. Increasing opposition and innovations in automotive field tends to alter the existing products or replacing old products by new and sophisticated material products. A suspension system of automotive is one of the areas where these innovations are carried out regularly. Leaf springs are generally used in suspension systems to absorb shock, loads or fuel efficiency and improved riding qualities in automotive like light vehicles, heavy duty trucks and in rail systems [2].

1.2. Composite Leaf Spring

A composite material is the combination of two or more materials that produce a synergistic effect so that the combination produces aggregate properties that are different from any of those of its constituents attain independently. This is intentionally being done today to get different design, manufacturing as well as service advantages of products. Up on those products leaf spring is the focus of this project for which researches are running to get the best composite material, which meets the current requirement of strength and weight reduction in one, to replace the existing steel leaf spring.

Advantages of Composite Leaf Spring

- ✓ Reduced weight.
- ✓ Due to laminate structure and reduced thickness of the mono composite leaf spring, the overall weight would be less.
- ✓ Due to weight reduction, fuel consumption would be reduced.
- ✓ They have high damping capacity; hence produce less vibration and noise.
- ✓ They have good corrosion resistance.
- ✓ They have high specific modulus and strength.
- ✓ Longer fatigue life.

Disadvantages of Composite Materials

- ✓ The leaf must span from one side of the car to the other. This can limit applications where the drive train or another part is in the way.
- ✓ Steel coils are commodity items where as a single composite leaf spring costs more than two of them.

- ✓ Composite mono leaf allows for considerable variety in shape, thickness, and materials. They are inherently more expensive to design, particularly in performance applications.

Applications of Composite Materials

- ✓ Fiber epoxy composites have been used in aircraft engine to enhance the performance of the system. The pilot's cabin door of aircrafts has also been made with fiber glass resin composites and these are now used in other transport systems
- ✓ In Railway carriages, it is desirable to reduce the weight of rail car bodies as well as heavy transport vehicles, which in turn reduce power and braking requirements. It also reduces maintenance costs.
- ✓ Assembling of various parts is usually done by adhesive bonding, using resins that are catalyzed to cure at room temperature in a short time.
- ✓ An important consideration in the use of composites is light weight.

1.3. Leaf Spring Failure

Leaf springs are a limited life component. Regardless of how well a spring is maintained or how favorable the operating conditions are; all springs will eventually fail from fatigue caused by the repeated flexing of the spring. Once the spring life limit is reached a fatigue failure will or has occurred.

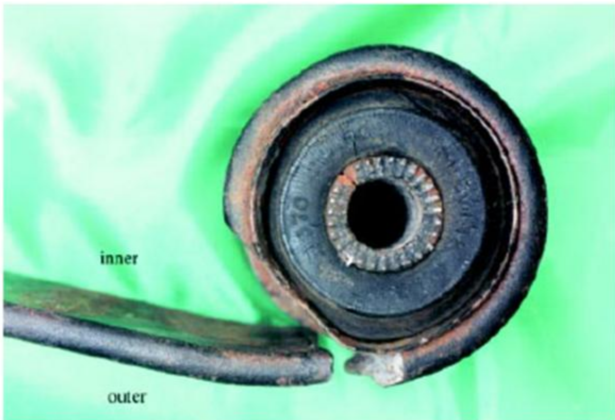


Figure 1.3: Failure of Leaf Spring

Factors influencing fatigue life of leaf spring:

Overloading

- ✓ The higher the loads or deflections seen by a spring, the lower its fatigue life.

Shock Absorbers

- ✓ A properly functioning shock absorber will tend to reduce the spring deflection as the vehicle hits a bump. Lower spring deflections mean lower operating stresses on the spring which in turn gives longer fatigue life. This is especially true for full taper springs which do not have the high interleaf friction to help dampen spring deflections. Worn or missing shock absorbers must be replaced to maximize spring life.

Brake Adjustments

- ✓ Improperly adjusted brakes can also reduce spring life. Under braking, springs are expected to absorb some of the braking forces. If the brakes on an axle are unevenly adjusted one spring will have to absorb more than its share of braking force which can reduce its fatigue life.

Protective Coatings

- ✓ Corrosion is one of the major factors in reducing spring life. Proper paints and care during handling and installation can help to slow the spread of spring corrosion. On full taper springs the only acceptable coating is the individual painting of each leaf with zinc-rich paint. This paint may be recognized by its characteristic gray color.

Surface Condition

- ✓ The condition of the spring surface also has an effect on fatigue life. Generally, a fatigue crack will start at some sort of surface defect on the spring leaf. Therefore, care needs to be used when manufacturing and installing springs to reduce these defects to a minimum.

Shot Peening

- ✓ Extensive testing indicates that shot peening can increase the life of springs by a factor of three or more. It is not enough, however, to simply shot peen the first one or two leaves in an assembly-all leaves must be shot peened.

Decarburization and Steel Quality

- ✓ Improper manufacturing methods can also reduce fatigue life. For example, poorly controlled heat-treat furnaces can excessively decarburize the leaf surface.

- ✓ Decarburization is the loss of carbon from the steel surface which will result in a soft leaf surface once heat-treating is complete. This soft layer will not be able to handle the spring stresses and will lead to early failure. Poor steel quality can also influence spring life. If the steel has excessive impurities in it, the fatigue life will be reduced.

Maintenance

- ✓ Finally, improper maintenance will affect spring life.
- ✓ Spring eyes and other suspension components should be regularly greased to prevent binding.
- ✓ U-bolts should never be reused.
- ✓ Axle seats, top plates and other components should be periodically inspected and replaced as required.

Spring failures may be categorized into three types:

Early Life Failures

- ✓ These types of failures occur generally due to a spring defect, installation problem or overload. This may be due to the material used, the manufacturing processes or improper installation techniques. This type of failure may also be caused by a short-term overload condition.

Midlife Failures

- ✓ Once the spring has passed the time in service which would expose early life failures, a very low failure rate should be observed, assuming the spring is subjected to normal service.

Late Life Failures

- ✓ At this point, the frequency of spring failures will tend to increase rapidly as the useful life of the spring has been reached. By this time the spring steel has been fatigued and corroded to a point where its useful life is over.

Failures occurring in early and midlife of the spring are usually most economically handled by repairing the broken leaf rather than replacing the spring. Failures in older springs occur at a point when all leaves have reached their fatigue life the spring should now be replaced. The difficulty, of course, is determining what type of failure the spring has experienced.

Basically, the condition of the spring, as well as its service history, will indicate if the spring should be repaired or replaced.

1.4. Problem Statement

Leaf spring is one of the critically loaded components in the vehicle parts. Failure of mechanical assembly component is a common phenomenon due to fracture that occurs almost everywhere in mechanical structures. The main cause of failure of leaf spring is due to large bending behavior [30]. Fatigue is the primary cause of failure of leaf spring due to the cyclic loading, continuous running of the vehicle there is a declination (maximum deformation) in the level of soothed offered by the spring, low strength in leaf springs be likely to break and deteriorate at the eye end segment which is extremely near to the shackle and at the middle, and also the usual steel leaf spring having more weight, which additionally influences the fuel efficiency.

The eye end segment regions in leaf springs have been identified as the highest stressed or critical location of a leaf spring and are often the sight of fatigue crack initiation. Therefore, due to this, this paper was proposed to analyze the geometry design for the modified composite spring, stresses acting on the leaf spring, and also replacing better material, design optimization and enhanced manufacturing process will improve life and weight of the spring.

1.5. Objectives of The Work

1.5.1. General Objective

The main objective is design analysis of the light weight vehicle (LWV) composite leaf spring for weight optimization and improving fatigue life.

1.5.2. Specific Objectives

The following are important points regarding to this objective of the study.

- ✓ Study leaf spring and its design.
- ✓ Study selection of proper composite material,
- ✓ Study the effect of material on surface fatigue.
- ✓ Geometric modeling of leaf spring using SOLIDWORK software.
- ✓ To carry out analysis of existing and modified leaf spring for the same loading condition using analytical method and ANSYS WORKBENCH software.
- ✓ Finding the maximum stress location and its magnitude in leaf spring.

- ✓ Comparison of static, fatigue analysis and optimization of leaf spring using ANSYS WORKBENCH software.
- ✓ Recommendation of new fatigue life prediction for light weight vehicle leaf spring.

1.6. Scope and Limitation of the project

This research covers the design and fatigue life prediction of composite leaf spring for a light weight four-wheeler vehicle. But, the design is limited to the static loading and single (mono) composite leaf spring only.

1.7. Methodology

1.7.1. Literature survey

There is a vast amount of literature related to Finite Element Analysis and optimization of leaf spring. Many research publications, journals, reference manuals, newspaper articles, handbooks; books are available of national and international editions dealing with basic concepts of FEA. The literature review presented here considers failure analysis, optimization and finite element modeling of a leaf spring.

1.7.2. Analytical Method

In the design of leaf springs, it is assumed that the leaf spring is a beam with two supports. The leaf spring is designed by considering selecting material for the modified component with its mechanical and material properties. When the leaf spring is overloaded and when the vehicle is continuously running in the irregular surface at which the bending moment is maximum. In this study the design analysis for leaf spring is performed and also the material used for the suspension is optimized. And also, the bending moment, stress, deflection, weight, and fatigue life are the main target for calculations of the leaf spring.

1.7.3. Modeling and FEM analysis `

The finite element method is numerical analysis technique for obtaining approximate solutions to a wide variety of engineering problems. Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. These steps will lead to the stresses and displacements in the component. In this study, this component is analyzed using SOLIDWORK software for 3D modeling and ANSYS 15.0 WORKBENCH software for existing steel leaf spring and the modified single composite leaf spring.

1.7.4. Procedures in FEA

a. Generation of the Geometry of Leaf Springs

Finite element analysis of leaf spring is done by geometry generation using SOLIDWORK software and analysis is using through the ANSYS WORKBENCH 15.0 software. Using proper type of loading and boundary condition are very important in finite element analysis. The solid model generated for the leaf spring is shown in figure 3.6 and the optimized is drawn by using design analysis; and for the existing conventional leaf spring is performed by collecting data from TOYOTA MOENCO company Bahir Dar branch.

b. Mesh Generation

FEA analysis was performed on leaf springs for improving fatigue life, and boundary conditions are applied according to leaf spring mounting condition. Tetrahedral elements were used to mesh the leaf spring finite element geometry. FE models were done for conventional and optimized one, to analyze stress, and improve life of leaf spring with different mesh size or by using mesh optimization to obtain good result which is closer to analytical result.

This step involves subdividing the domain into elements and nodes. For continuous system, this step is very important and the answers obtained are only approximate. In this case, the accuracy of the solution depends on the discretization used. Import Parasolid format model in ANSYS WORKBENCH simulation module. Tetrahedral elements are used to mesh the leaf spring finite element geometry.

c. Loading and Boundary Conditions

Boundary conditions are playing the important role in finite element analysis. Load is applied on the component; when the spring is at the position of maximum bending stress or at the overloading position; but, loading data is taken from the calculated result. Boundary condition is based on under supporting condition of leaf spring then obtaining results.

1.8. Organization of the Paper

This thesis is organized into five main chapters. In the first chapter introduction, background, statement of the problem and objective to be achieved are discussed. In chapter two a review of literature appropriate to this thesis is stated. In chapter three material selection, dimensions, conditions, design analysis, and methods used were discussed. In chapter four results and discussions from FEM and analytical method for existing and modified leaf spring are analyzed and discussed. Finally, in chapter five conclusions, recommendation and future works are presented.

CHAPTER TWO

2. LITERATURE REVIEW

2.1. Introduction

Review of literatures reveals that different previous work which helps for the guidance of this work. These previous related works may be journals, conference papers, design works and books related to this paper. Selection of appropriate conditions, approaches and methodologies with these journals and other materials will strengthen for the successful accomplishment of the paper. And also, this study is mostly about design of composite leaf spring for estimating fatigue life and reduces weight; finite element modeling research journal concerning these issues are also summarized and discussed in this chapter.

2.2. Composite materials for leaf spring

To meet the need of natural resources conservation, automobile manufacturers are attempting to reduce the weight of vehicles in recent years. The interest in reducing the weight of automobile parts has necessitated the use of better material, design, and manufacturing processes. The suspension leaf spring is one of the potential elements for weight reduction in automobiles as it leads to the reduction of un-sprung weight of automobile. The elements whose weight is not transmitted to the suspension spring are called the unsprung elements of the automobile. This includes wheel assembly, axles, and part of the weight of suspension spring and shock absorbers. The leaf spring accounts for 10-20% 0f the un-sprung weight. The reduction of unsprung weight helps in achieving improved ride characteristics and increased fuel efficiency. The cost of materials constitutes nearly 60-70% 0f the vehicle’s cost and contributes to the better quality and performance of the vehicle. The introduction of fibre reinforced plastics (FRP) made it possible to reduce the weight of machine element without any reduction of the load carrying capacity. Because of FRP material’s high elastic strain energy storage capacity and high strength-to-weight ratio compared with those of steel, multi-leaf 33steel springs are being replaced by mono-leaf FRP springs. FRP springs also have excellent fatigue resistance and durability. But the weight reduction of the leaf spring is achieved not only by material replacement but also by design optimization [6].

Shankar et al. [10] studied a mono composite leaf spring with varying width and varying thickness is designed and manufactured. Computer algorithm using C-language has been used for the design of constant cross-section leaf spring. The results showed that a spring width decreases hyperbolically and thickness increases linearly from the spring eyes towards the axle seat. The experimental test is carried on both steel and composite leaf spring and compared the result. It is observed that composite leaf spring is more superior than steel with a large weight reduction.

Ekbote et al. [13] studied a double tapered mono leaf fiber reinforced composite leaf spring is developed to replace the nine-leaf steel spring. The main consideration is given to the optimal design of the geometry of E-glass/epoxy composite leaf spring. A comparative study has been made between composite and steel leaf spring with respect to weight and strength. The comparison of analysis results with analytical values from basic design equations of multi leaf steel spring will yield a maximum variation of 6% in deflection and 4% in stress (Von-Misses) values.

Palaskar et al. [25] studied that if the thickness is varied parabolic ally along the length of the leaf spring, better results can be obtained. Material saving up to 25% can be achieved by parabolic variation in thickness along the length of the leaf spring. In other words, distribution of stress of leaf spring can be improved (it becomes more uniform) after optimization. Utilization of material is improved due to optimization. Riding comfort of passengers is improved due to reduction in unsprung mass of the vehicle. More deflection can be obtained after optimization thus increasing the riding comfort further.

Karlus et al. [22] showed that as per the outcome obtain, by substituting the usual (55Si2Mn90) steel material by composite material (Carbon- Epoxy) we can decrease the stress produced in the leaf spring and moreover we anticipate that by substituting the material the enhanced comfort level throughout the spring can be accomplished or in other word it concentrated the total deflect ion of the leaf spring. By the reduction of weight and the less stresses, the fatigue life of Carbon- Epoxy composite leaf spring is to be higher than that of steel leaf spring.

Prasad et al. [16] showed that with development of analytical formulation for composite leaf spring and comparing the obtained results with the conventional steel leaf spring with 4 leaves. When maximum load is applied on the steel leaf spring, the maximum stress is greater than that of FRP leaf spring. Even under the maximum load, the maximum stress in the FRP is within the allowable limit. The weight reduction has greater influence in noise and vibration characteristics. Glass fibers are for manufacturing instead of carbon due to low cost.

Hargude et al. [24] showed that the objective is to review various techniques used to analyses of composite mono leaf spring for the load carrying capacity, stiffness and weight savings of composite leaf spring. The dimensions of an existing conventional steel leaf spring of a heavy commercial vehicle are taken same dimensions of conventional leaf spring are used to fabricate a composite multi leaf spring using e-glass/epoxy, c- glass/epoxy, s- lass/epoxy unidirectional laminates. Under the dynamic load conditions natural frequency and stresses of steel leaf spring and composite leaf spring are found with the great difference. The natural frequency of composite material is high than the steel leaf spring. Reductions in weight about 85 to 90% in composite leaf spring are observed than conventional with same level of performance. E-glass epoxy is better than using mild-steel as though stresses are little bit higher than mild steel, E-glass epoxy is having good yield strength value and also epoxy material components are easy to manufacture and this having very low weight while comparing with traditional materials. S-glass is having better results while comparing with E-glass and mild steel. CAE tools provides a cost effective and less time-consuming solution in comparison with the experimental testing but the results may vary in the specified range.

Bhanage et al. [17] showed comparative simulation results of E-glass/Epoxy mono composite leaf spring for different layup as well for different thickness condition. First, simulation results have been performed for SAE 1045-450-QT steel material from weight saving and stress reduction point of view. Secondly, comparative simulation analysis performed between [0-45-(-45)-90-0], [0-45-(-45)-0], [0-0-45-(-45)-0] and [0-45-90] lay-up with different thickness from 9 mm, 10 mm, 12 mm, 13mm and 15 mm, considered according to selection of each layup thickness. The design and comparative simulation analysis was done in ANSYS Software. Similar mechanical properties for E-Glass/Epoxy composite material were considered for all simulation procedure. The design constraints and meshing were also being similar for all conventional and composite models of leaf spring.

Mahesh et al. [15] studied the achievement of weight reduction with adequate improvement of mechanical properties has made composite a very replacement material for convectional steel. From the comparative study, it is seen that the composite leaf spring are higher and more economical than convectional leaf spring. After prolonged use of conventional metal Coil Spring, its strength reduces and vehicle starts running back side down and also hits on the bump stoppers (i.e. Chassis). This problem is entirely removed by our special purpose Composite leaf Springs.

Deshmukh et al. [11], in this paper gives the brief look on the suitability of composite leaf spring on vehicles and their advantages. The material selected is glass fiber reinforced plastic

(GFRP) and the epoxy resin can be used which is more economical to reduce total cost of composite leaf spring with similar mechanical and geometrical properties to the multi leaf spring. The weight of the leaf spring is reduced considerably about 74 % by replacing steel leaf spring with FRP leaf spring. Besides the reduction of weight, the fatigue life of composite leaf spring is predicted to be satisfactory. Thus, the objective of reducing the un-sprung mass is achieved to a larger extent. The stresses in the GFRP leaf spring are much lower than that of the steel spring.

Muhsin et al. [14] showed experimental and theoretical study of composite materials reinforcement fiber types are presented. The main conclusions of this work are the best modulus of elasticity for composite materials are unidirectional composite materials in longitudinal direction and woven composite materials in any direction. The powder, particle, mats, and short fiber composite materials may be give isotropic properties of composite materials. With depend on resin materials properties. The variable of matrix materials affects the properties in powder, short, and mats composite materials more than in properties of woven and unidirectional composite materials. Unidirectional composite materials give minimum modulus of elasticity in transverse direction compares with other composite materials types. Weight reduction has been the main focus of automobile manufacturers in the present scenario. The leaf spring suspension accounts for about 10-20% of vehicle un-sprung weight. Thus, it becomes a potential unit for weight reduction. The weight reduction can be achieved by choosing better materials and optimized design etc. The replacement of steel with optimally designed composite leaf spring can provide 93% weight reduction. Moreover, the composite leaf spring has lower stresses compared to steel spring. All these will result in fuel saving which will make countries energy independent because fuel saved is fuel produced [14].

Composite leaf springs in particular in light trucks deal with the cargo load, comfort, and safety aspects. Fibre reinforced epoxy coil springs have been known for years. Now a process for mass production has been developed. The era of electrically driven cars requires a change of thinking. It will be essential to reduce the weight of the vehicle. The question, “Battery or Passenger?” would be answered in a consumer-friendly manner, “Battery and Passenger”. A significant amount of today’s automotive composite applications are still parts which support the structure or are parts of the car body such as fenders, trunk lids, hoods. However, the new generation of electrically driven cars requires chassis and other load bearing structures made from CFRP (Carbon Fibre Reinforced Polymers). Epoxy carbon and glass composites have proven their outstanding mechanical, thermo-mechanical and fatigue resistance properties [18].

Jaydeep et al. [27], in their paper discussed the brief look on the suitability of composite leaf spring on vehicles and their advantages. Compared to the steel spring, the composite spring has stresses that are much lower, the natural frequency is higher and the spring weight is nearly 85 % lower with bonded end joint and with complete eye unit. The attempt has been made to fabricate the FRP leaf spring economically than that of conventional leaf spring. The leaf spring is design by considering as it is behaving like a cantilever beam. For the analysis purpose ANSYS software is selected as it gives good result. Advantages such as reduction in noise, increasing in comfort ride.

Sharif et al. [28] studied in this paper is to design and analyze composite mono leaf spring of constant width and thickness having the same bending stiffness of semi-elliptical laminated leaf spring. Stress analysis was done by using analytical method and results obtained by analytical methods are compared with ANSYS. The results obtained by analytical methods showed good agreement with ANSYS results. A Tsai-Hill failure criterion was used to check whether stresses are within reasonable levels for each ply. The stresses induced in the composite leaf springs were found to be 33.79% less compared to steel leaf spring. When steel leaf spring is replaced by composite leaf spring a weight reduction of 77.29% is obtained, 2.23 times higher natural frequency, 1.371 times more strain energy storage capacity, 33.79 % lesser stress and lesser value of spring rate is obtained in the composite leaf spring compared to steel leaf spring.

2.3. Design analysis of leaf spring

Different researchers approach for design, analysis, modelling and simulation of a composite leaf spring are reviewed as follows:

Kumar et al. [2], design and experimental fatigue analysis of composite multi leaf spring using glass fibre reinforced polymer are carried out using life data analysis, in this particular literature. Compared to steel spring, the composite leaf spring is found to have 67.35 % lesser stress, 64.95 % higher stiffness and 126.98 % higher natural frequency than that of existing steel leaf spring. The conventional multi leaf spring weighs about 13.5 kg; whereas the E-glass/Epoxy multi leaf spring weighs only 4.3 kg. Thus, the weight reduction of 68.15 % is achieved. Besides the reduction of weight, the fatigue life of composite leaf spring is predicted to be higher than that of steel leaf spring. Life data analysis is found to be a tool to predict the fatigue life of composite multi leaf spring. It is found that the life of composite leaf spring is much higher than that of steel leaf spring. The 3D FEM model of leaf spring is simulated using ANSYS [2].

Amare [29], in this paper he focused on reducing the weight of vehicle simultaneously increase or maintain the strength of their spare parts. So, this work considered leaf spring because of it contributes considerable amount of weight to the vehicle and needs to be strong enough. This work considered for light weight three-wheeler vehicles and a single E-glass/epoxy leaf spring were designed, simulated by following design rules of the composite materials and fabricated by hand lay-up method. The leaf spring was tested and analyzed for static load only. It was concluded that E-glass/epoxy leaf spring designed and simulated in this work having stresses much below the strength properties of the material satisfying the maximum stress failure criterion. It has observed that the fatigue life of the single E-glass/epoxy leaf spring of $221.16*10^3$ cycles.

Venkatesan et al. [35] also used three-dimensional finite element method of analysis. They pointed that the leaf spring behaves like a simply supported beam and the flexural analysis is done considering it as a simply supported beam. The simply supported beam is subjected to both bending stress and transverse shear stress. Flexural rigidity is an important parameter in the leaf spring design and test out to increase from two ends to the Centre. They tried to access three design approaches. These are:

- ✓ constant thickness, varying width design,
- ✓ constant width, varying thickness design and
- ✓ constant cross-section design.

Out of the above-mentioned design concepts; the constant cross-section design method is selected due to the following reasons: due to its capability for mass production and accommodation of continuous reinforcement of fibres. Since the cross-section area is constant throughout the leaf spring, same quantity of reinforcement fibre and resin can be fed continuously during manufacturing. It is also quite suitable for filament winding process.

Ravindra et al. [19] in this paper describes design and analysis of composite mono leaf spring. A composite mono leaf spring with Carbon/Epoxy composite materials is modelled and subjected to the same load as that of a steel spring. Compared to mono steel leaf spring the laminated composite mono leaf spring is found lesser stresses and weight reduction of 22.5% is achieved. Based on the results, it was inferred that carbon/epoxy laminated composite mono leaf spring has superior strength and stiffness and lesser in weight compared to steel material considered in this investigation.

Bhanage et al. [17] studied the objective to present a design and simulation study on the fatigue performance of a glass fibre/epoxy composite leaf spring through design and finite element method and prove the reliability of the validation methods based only on simulation. Due to weight reduction and stress, stiffness criteria, multi steel leaf spring is proposed to be replaced with E- Glass Epoxy composite leaf springs. This paper will help to understand linear static behavior of the composite leaf spring and simulation data to improve the fatigue life of the leaf spring using Computer Aided Engineering tool.

Shivashankar et al. [12] stated that taking the advantages of mass production and continuous fibre accommodation, composite leaf spring with constant cross sectional area is designed using Genetic Algorithm (GA) method. The weight of the composite leaf spring can be reduced by 53.5% by applying the GA optimization technique. Composite mono leaf spring reduces the weight by 85% for E-Glass/Epoxy over conventional leaf spring. The reduction of 93%weight is achieved by replacing conventional steel spring with an optimally designed composite mono-leaf spring. Here experimental and numerical methods of analysis are employed. The element SHELL 99, SOLID 46 is the best suited for modelling of composite material. SHELL 99 is an 8 – node, 3D shell element with six degree of freedom at each node. The advantage of SOLID 46 is that we can stack several elements to model more than 250 layers. Here selected element was SOLID 46. Static analysis is performed and the procedure consists of, build the model and defining parameters. The parameters for building the composite leaf spring are; young's modulus, (E_{XX}) value, poison ratio, Y(PR_{XY}) value, length of cantilever beam, width of cantilever beam, and height of cantilever beam. Experimental results from testing the leaf springs under static loading containing the stresses and deflection are calculated. These results are also compared with FEA. The weight of the leaf spring is reduced considerably about 85 % by replacing steel leaf spring with composite leaf spring. Thus, the objective of reducing the un-sprung mass is achieved to some extent. Also, the stresses in the composite leaf spring are much lower than that of the steel spring [12].

2.4. Failure Analysis

Failure prediction in large-scaled structures that are subjected to extreme loading conditions has been of utmost interest in the scientific and engineering community over the past century. Failure of mechanical assembly component is a common phenomenon due to fracture that occurs almost everywhere in mechanical structures. Leaf spring studies, suggest that leaf spring failures often occurs in the eye areas due to fatigue failure and large bending behavior [30]. Different researcher was discussed the literature review as:

Fuentes et al. [9] did a study causes of leaf spring failure subjected to fatigue loading. They concluded that the premature fracture failure on the leaf springs was as a result of mechanical fatigue and was caused by a combination of design, metallurgical and manufacturing deficiencies.

Kumar et al. [8] studied static and fatigue analysis of steel leaf springs and composite multi leaf spring made up of glass fiber reinforced polymer using life data analysis. The dimensions of existing conventional steel leaf springs of a light commercial vehicle were taken and verified by design calculations. Static analysis of 2-D model of conventional leaf spring was also performed using ANSYS 7.1 and compared with experimental results.

Ramakanth et al. [1] have carried out the fatigue and static analysis for steel, composite, and hybrid leaf springs. They finally concluded that, under the same static load conditions the stresses in leaf springs were found with great difference. Stresses in composite leaf springs were found out to be less as compared to the conventional steel leaf springs. In their study, the fatigue analysis of the steel leaf springs was carried out with four approaches; Soderberg’s approach showed better results for the analysis of life data for leaf spring.

Aher et al. [5] performed Prediction of Fatigue life of multi leaf spring used in the suspension system of light commercial vehicle along with analytical stress and deflection calculations. This present work describes static and fatigue analysis of a modified steel leaf spring of a light commercial vehicle (LCV). The fatigue life of the leaf spring is also predicted. They finally conclude that, the fatigue life prediction is performed based on finite element analysis and fatigue life simulation method.

Lee et al. [3], they proposed that fatigue life prediction is based on knowledge of both the number of cycles that the part will experience at any given stress level during that life cycle and another influential environmental and use factor. The local strain-life method can be used pro-actively for a component during early design stage.

Johanneson et. al [4] performed the prediction of fatigue life cycle of the leaf spring by representing different road surface and driving condition of the vehicle. Using constant amplitude loading, test have been performed in controlled condition. But they proposed in variable amplitude testing, a lot of parameter need to be consider like cycle range and sampling frequency.

2.5. Finite Element Method

Since the leaf spring has a complex geometry for analysis, finite element models have been considered to give an accurate and reasonable solution whenever laboratory testing is not available. In the FEM, different researchers have been reviewed as:

Ahmad et al. [7], investigated a parabolic spring under cyclic strain loading using numerical modeling with variable amplitude loading for the fatigue life analysis. Service loading of parabolic spring was collected using data acquisition system. The finite element method (FEM) was performed on the spring model to observe the distribution of stress and extent of damage. The experimental works were performed in order to validate the FEM result.

Parkhe et al. [19] studied that mono composite leaf spring with Carbon/Epoxy composite materials is modelled and subjected to the same load as that of a steel spring. The design constraints were stresses and deflections. The stresses induced in the Carbon/Epoxy composite leaf spring are 42% less than that of the steel spring nearly. The finite element solutions show the good correlation for total deformation with analytical results. Study demonstrates that the composite can be used for leaf spring for the light vehicle and meet the requirement, together with the sustainable weight reduction. A weight reduction achieved in mono composite leaf spring is about 22.15%.

CHAPTER THREE

3. ANALYTICAL METHODS AND CONDITIONS

3.1. Material Selection

Materials constitute nearly 60%-70% of the vehicle cost and contribute to the quality and the performance of the vehicle. Even a small amount in weight reduction of the vehicle, may have a wider economic impact [28]. Composite materials are proved as suitable substitutes for steel in connection with weight reduction of the vehicle. Hence, the composite materials have been selected for leaf spring design are: -

3.1.1. Fiber Selection

The commonly used fibers are carbon, glass, etc. Among these, the glass fiber has been selected based on the cost factor and strength. The types of glass fibers are C-glass, S-glass and E-glass. The C-glass fiber is designed to give improved surface finish. S-glass fiber is design to give very high modular, which is used particularly in aeronautic industries. The E-glass fiber is a high-quality glass, which is used as standard reinforcement fiber for all the present systems well complying with mechanical property requirements. Thus, E-glass fiber are found appropriate for this application. Vertical vibrations and impacts are buffered by variations in the spring deflection so that the potential energy is stored in spring as strain energy and then released slowly. So, increasing the energy storage capability of a leaf spring ensures a more compliant suspension system. The material used directly affects the quantity of storable energy in the leaf spring. As the energy storage capacity of E-glass/epoxy is much higher; it is the best material for the application selected. Also, the material with maximum strength and minimum modulus of elasticity is the most suitable material for the leaf spring application. Due to this, the material used in this study is E-glass/Epoxy [28].

3.1.2. Resin Selection

In a FRP leaf spring, the inter laminar shear strengths is controlled by the matrix system used. Since, these are reinforcement fibers in the thickness direction, fiber do not influence inter laminar shear strength. Therefore, the matrix system should have good inter laminar shear strength characteristics compatibility to the selected reinforcement fiber. Many thermoset resins such as polyester, vinyl ester, epoxy resin is being used for fiber reinforcement plastics (FRP) fabrication. Among these resin systems, Parametric Analysis of Composite Leaf Spring epoxies show better inter laminar shear strength and good mechanical properties. Hence, epoxies are found to be the best resins that would suit this application. Different grades of epoxy resins and hardener combinations are classifieds based on the mechanical properties which in combination with hardener 758 cures into hard resin. It is characterized by:

- ✓ Good mechanical and electrical properties.
- ✓ Faster curing at room temperature.
- ✓ Good chemical resistance properties.

Matrix materials or resins in case of polymer matrix composites can be classified according to their chemical base i.e. thermoplastic or thermosets. Thermoplastics have excellent toughness, resilience and corrosion resistance but have fundamental disadvantage compared to thermosetting resins, in that they have to be molded at elevated temperature. Thermosetting plastics or thermosets are formed with a network molecular structure of primary covalent bonds. Some thermosets are cross-linked by heat or a combination of heat and pressure. Others may be cross-linked by chemical reaction, which occurs at room temperature. At present, epoxy resins are widely used in various engineering and structural applications such as aircraft, aerospace engineering, sporting goods, automotive, and military aircrafts industries. In order to improve their processing and product performances and to reduce cost, various fillers are introduced into the resins during processing. Epoxy resins are the most commonly used thermosets plastic in polymer matrix composites. Hence from the above listed advantages of epoxy resin it has been selected for this study [28].

3.2. Design Selection

The leaf spring behaves like a simply supported beam and the flexural analysis is done considering it as a simply supported beam. The simply supported beam is subjected to both bending stress and transverse shear stress. Flexural rigidity is an important parameter in the leaf spring design and test out to increase from two ends to the center [29].

Constant Thickness, Varying Width Design: In this design the thickness is kept constant over the entire length of the leaf spring while the width varies from a minimum at the two ends to a maximum at the center.

Constant Width, Varying Thickness Design: In this design the width is kept constant over the entire length of the leaf spring while the thickness varies from a minimum at the two ends to a maximum at the center.

Constant Cross-Section Design: In this design both thickness and width are varied throughout the leaf spring such that the cross-section area remains constant along the length of the leaf spring. Out of the above-mentioned design concepts, the constant cross-section design method is selected due to the following reasons: -

- ✓ Due to its capability for mass production and accommodation of continuous reinforcement of fibers.
- ✓ Since the cross-section area is constant throughout the leaf spring, same quantity of reinforcement fiber and resin can be fed continuously during manufacture.
- ✓ Also, this is quite suitable for filament winding process.

3.3. Design Parameters and Material Properties of Leaf Spring

The design parameter for Toyota single cabin land cruiser vehicle model with rear leaf spring is used in this study and the design parameters and material properties are given below in Table 1 and Table 2. The dimensions are measured from Toyota service center and material properties are taken from the research paper.

Table 1: Design Parameters of Existing Steel Leaf Spring [39]

Parameters	**Values**	**Unit**
Total length (eye-to-eye)	1600	mm
Arc height at axle seat (Camber)	143	mm
Number of full length leaves	2	-
Number of graduated leaves	6	-
Width of the leaves (b)	70	mm
Thickness of the leaves (h)	7	mm
Full bump loading	9487.5	N
Spring weight	49	Kg

Table 2: Material Properties of Conventional Steel Leaf Spring [8].

Properties	**Values**	**Unit**
Existing Material - Steel	65Si7	-
Young’s Modulus – E	$2 * 10^5$	MPa
Poisson’s Ratio- ν	0.3	-
Ultimate Tensile Strength	1272	MPa
Yield Tensile Strength	1158	MPa
Density	7850	Kg/m^3
BHN	400 – 425	-
Behavior	Isotropic	-

From the material point of view, a unidirectional E-glass/Epoxy composite material is selected. It is selected due to its relative advantages stated in the material selection for leaf spring above, mainly high strength to weight ratio and high capacity of storing strain energy in the longitudinal direction of the fibres. The properties of E-glass/Epoxy composite material are given as follows in Table 3 [20].

Table 3: Mechanical Properties of E-Glass/Epoxy [20]

Properties	**Values**	**Unit**
Modulus of elasticity along the longitudinal direction (E_1)	54	GPa
Modulus of elasticity along the transverse direction (E_2)	18	GPa
Tensile strength of the material	900	MPa
Compressive strength of the material	450	MPa
Shear modulus (G_{12})	21.6	GPa
Density (ρ)	2600	Kg/m^3
Major Poisson's ratio (ν_{12})	0.25	-
Longitudinal tensile strength (σ_{u1})	1035	MPa
Transverse tensile strength (σ_{u2})	28	MPa
Longitudinal compressive strength ($\sigma_{u1'}$)	1035	MPa
Transverse compressive strength ($\sigma_{u2'}$)	138	MPa
Shear strength (τ_{u12})	41	MPa

3.4. Conditions

- ✓ All non-linear effects are assumed to be excluded in this study.
- ✓ Leaf spring is assumed to a beam member.
- ✓ Force is acting on the center of leaf spring.
- ✓ The stress-strain relationship for composite material is linear and elastic; hence Hooke's law is applicable for composite materials.
- ✓ The leaf spring has a uniform, rectangular cross section.
- ✓ The optimized composite leaf spring is assumed with large deformations and small strains behavior.

3.5. Design Specification of Leaf Spring

Here weight and measurements of four-wheeler model of “Toyota Single Cabin Land Cruiser HZJ79L Model Vehicle” are taken [37].

Kerb weight of the vehicle = 2065 kg

Load Carrying capacity of the vehicle = 1235 kg

Gross weight of the vehicle (W_{Gr}) =2065 + 1235 = 3300 kg

Taking factor of safety (FS) =1.15

Acceleration due to gravity (g) $= 10\ {}^{m}/_{s^2}$

Hence, total weight of the vehicle (W) = W_{Gr} * g * FS

= 3300 * 10 * 1.15 = 37950 N

Since the vehicle is 4-wheeler, a single leaf spring corresponding to one of the wheels takes up one fourth of the total weight.

$$F = \frac{W}{4} = \frac{37950}{4} = 9487.5\ N$$

3.6. Methods

3.6.1. Analytical Calculations of Steel Leaf Spring

Stress Analysis of Steel Leaf Spring

Leaf spring is the type of cantilever beam, it is simple rectangle shape. One side force is applied and other side is fixed.

There are two types of stresses are in the leaf spring.

- ✓ Bending stress
- ✓ Shear stress

The leaf spring is subjected to various forces but generally needs to be analyzed under bending loading condition and the normal stresses are important. Since analyzing half of the leaf spring is enough, half of the applied force would have been taken. But here we took as it is to account overloading of the vehicle and flexures of the leaf spring.

Hence, $\frac{L}{2} = 800\ mm$, $F = 9487.5$ N.

From equations of strength of materials, we have equations of maximum stress and maximum deflection for leaf spring:
The maximum stress applied in the steel leaf spring is given as:

$$\sigma_{max} = n\,\frac{6FL}{bh^2} \qquad (3.1)$$

We know that,

F is the applied load on leaf spring = *9487.5 N*,

L is straight length of leaf spring = *1600 mm,*

n is total number of leaves = *8,*

b is width of leaves = *70 mm,* and

h is thickness of the spring = *7 mm*.

$$\sigma_{max} = n\,\frac{6FL}{bh^2}$$

$$\sigma_{max} = n\,\frac{6F\left(\frac{L}{2}\right)}{bh^2} = \frac{8*6*9487.5*800}{70*7^2} = 1062\ MPa$$

The bending stress (σ_b) for conventional spring is given as:

$$\sigma_b = \frac{M}{I} y$$

But, since half of the spring is analyzed; the bending moment can be calculated as:

$$M = n * F * \frac{L}{2} = 8 * 9487.5 * 0.8 = 60720\ N.m$$

And,

$$y = n * \frac{h}{2} = 8 * \frac{7}{2} = 28\ mm$$

The conventional spring cross-sectional area is rectangular; the moment of inertia, I is given as follows:

$$I = \frac{nbh^3}{12} = \frac{8 * 0.070 * 0.007^3}{12} = 1.6 * 10^{-8}\ m^4$$

Therefore, the bending stress can be obtained:

$$\sigma_b = \frac{M}{I} y = \frac{60720}{1.6 * 10^{-8}} * 0.028 = 1062\ MPa$$

We use pure tensile loading nature of the leaf spring is considered, we took plane stress condition as the leaf is thin beam with multiple number of leaves. Thus, the bending stress is completely responsible to the lengthwise stress; $\sigma_1 = \sigma_b$.

$$\sigma_1 = 1062\ MPa$$

Stress along the crosswise direction can be obtained as;

$$\sigma_2 = \frac{F}{A} = \frac{F}{n * b * \left(\frac{L}{2}\right)} = \frac{9487.5}{8 * 0.07 * 0.8} = 0.2118\ MPa$$

The shear stress for existing spring can be calculated as follows;

$$\tau_{12} = \frac{F}{A_S} = \frac{F}{n*b*h}$$

$$\tau_{12} = \frac{9487.5}{8*0.07*0.007} = 2.42\ \text{MPa}$$

Deflection of Conventional Leaf Spring

The deflection of conventional leaf spring may be seeming that the leaf spring is not under small deflection conditions and has large deformation and more strains behavior. Applying this behavior in the spring design causes a high degree of nonlinearly to the problem.

Because of this, the optimization process for leaf spring was proposed. The maximum deflection for conventional leaf spring that given from strength of materials are [32]:

$$\delta_{max} = \frac{4FL^3}{nEbh^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.2)$$

$$\delta_{max} = \frac{4F\left(\frac{L}{2}\right)^3}{nEbh^3} = \frac{4 * 9487.5 * (800)^3}{8 * 2 * 10^5 * 70 * 7^3}$$

$$\delta_{max} = 506\ mm$$

3.6.2. Analytical Method of Composite Leaf Spring

Design Analysis of E-glass/Epoxy Leaf Spring

Based on the specific strain energy of steel spring and some composite materials, the E-glass/Epoxy is selected as the spring material. Many attempts have been made to substitute more economic resins for the epoxy but all attempts to use polyester or vinyl ester resins have been unsuccessful to date. The stored elastic strain energy in a leaf spring varies directly with the square of maximum allowable stress and inversely with the modulus of elasticity both in the longitudinal and transverse directions.

$$S = \frac{1}{2}\,\frac{{\sigma_t}^2}{\rho E}$$

Since the leaf spring is fixed with the axle at its center, only half of it is considered for analysis purpose.

Design Constraints

From Shiva and Vijayarangan for E-glass/ Epoxy [10]:

Maximum stress (σ_{max}) $= 473$ MPa

Maximum deflection (δ_{max}) $= 105$ mm

The measured data of light weight four-wheeler Single Cabin Toyota Land Cruiser HZJ79L model vehicle are measured as [39]:

Straight length of the leaf spring (L) = 1600 mm

The ratio of camber length to leaf span given by Manas Patnaik (2012) is [34]:

$$C/_{L} = 0.089$$

Where;

C is the camber length and

L is the leaf span.

Thus,

$$C = 0.089 * L = 0.089 * 1600\ mm$$

$$C = 142.4\ mm \approx 143\ mm$$

Figure 3.1: Important dimensions and free body diagram to analyze half of the leaf spring [29]

As it mentioned in the existing analysis of leaf spring; we are analyzing half of the spring only; here also in composite leaf spring analysis, we consider half of the spring for analysis purpose, so half of the applied force would have been taken. But here I took as it is; to account overloading of the vehicle and flexures of the leaf spring.

Hence, $\frac{L}{2} = 800\ mm,$

F = 9487.5 N,

Calculating for *'h'* and *'b'* dimensions which are capable of withstanding the loading behavior of the composite (E-glass/ Epoxy) leaf spring is the result of this design.

Solving equations (3.1) and (3.2) simultaneously for calculating the thickness *'h'* and width *'b'* of the leaf spring can be formulated, respectively, as follows:

$$h = \frac{\sigma_{max} L^2}{E\delta_{max}} \quad \text{..........} \quad (3.3)$$

And,

$$b = \frac{6F\left(\frac{L}{2}\right)}{\sigma_{max} h^2} \quad \text{..........} \quad (3.4)$$

Since we consider half of the leaf spring, it is substituted '$\frac{L}{2}$' instead of *'L'* to calculate *'h'* and *'b'*. As the ends of the leaf spring are hinged, the entire leaf spring will only be loaded under tension. Therefore, we have to consider only the longitudinal properties.

Equation (3.3) will be written and calculated as:

$$h = \frac{\sigma_{max}\left(\frac{L}{2}\right)^2}{E\delta_{max}} = \frac{473 * 10^6 * (800)^2 * 10^{-6}}{54 * 10^9 * 105 * 10^{-3}} = 0.0534\, m \approx 54\, mm$$

From equation (3.4); solve for the width 'b' and obtained as:

$$b = \frac{6F\left(\frac{L}{2}\right)}{\sigma_{max} h^2} = \frac{6*9487.5*800*10^{-3}}{473*10^6*(54*10^{-3})^2} = 0.034\, m = 34\, mm$$

Stress Analysis of E-Glass/Epoxy Leaf Spring

Calculating the bending stress (σ_b);

$$\sigma_b = \frac{M}{I} y \quad \text{..........} \quad (3.5)$$

Where;

M is the bending moment,

y is the distance from the neutral axis = $\frac{h}{2}$, and

I is the moment of inertia.

But, the bending moment, *M* of the component will be obtained; and which leads us to find out the bending stress of the single leaf spring.

$$M = F * \left(\frac{L}{2}\right) = 9487.5 * \frac{(1600 * 10^{-3})}{2}$$

$$= 9487.5 * 0.8 = 7590 \; N.m$$

And;

$$y = \frac{h}{2} = \frac{54}{2} = 27 \; mm$$

Since, the leaf spring cross-sectional area is rectangular; the moment of inertia, I is given as follows:

$$I = \frac{bh^3}{12} = \frac{0.034 * 0.054^3}{12} = 4.46 * 10^{-7} \; m^4$$

Therefore; bending stress becomes:

$$\sigma_b = \frac{M * y}{I} = \frac{7590 * (27 * 10^{-3})}{4.46 * 10^{-7}}$$

$$\sigma_b = 459331881 \; {}^{N}/_{m^2} = 459.3 \; MPa$$

Since, we use unidirectional orientation of fibres and pure tensile loading nature of the leaf spring is considered, we took plane stress condition as the leaf is thin beam. Thus, the bending stress is completely responsible to the longitudinal stress; $\sigma_1 = \sigma_b$.

$$\sigma_1 = 459.3 \; MPa$$

Stress along the transverse direction can be obtained;

$$\sigma_2 = \frac{F}{A} \quad \text{……………………………………………………………………} \quad (3.6)$$

$$\sigma_2 = \frac{F}{b * \left(\frac{L}{2}\right)} = \frac{9487.5}{34 * 800 * 10^{-6}} = 0.349 \; MPa$$

The shear stress will also be calculated as follows;

$$\tau_{12} = \frac{F}{A_S} \quad \text{……………………………………………………………………} \quad (3.7)$$

$$\tau_{12} = \frac{9487.5}{34 * 54 * 10^{-6}} = 3.17 \; MPa$$

Now we need to calculate the strains of the model of the leaf spring.

$$\begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{Bmatrix} = \begin{bmatrix} C_{11} & C_{12} & 0 \\ C_{21} & C_{22} & 0 \\ 0 & 0 & C_{66} \end{bmatrix} \begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{Bmatrix} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.8)$$

Where;

ε_{ij} = Strain matrix,

C_{ij} = Compliance matrix, and

σ_{ij} = Stress matrix.

$$C_{11} = \frac{1}{E_1} = \frac{1}{54 * 10^9} = 18.52 * 10^{-12}\, Pa^{-1}$$

$$C_{22} = \frac{1}{E_2} = \frac{1}{18 * 10^9} = 55.6 * 10^{-12}\, Pa^{-1}$$

$$C_{12} = \frac{-\nu_{12}}{E_1} = \frac{-0.25}{54 * 10^9} = -4.63 * 10^{-12}\, Pa^{-1}$$

$$C_{66} = \frac{1}{G_{12}} = \frac{1}{21.6 * 10^9} = 4.63 * 10^{-11}\, Pa^{-1}$$

Substituting the values of compliance and stress matrices' elements in to equation (3.8);

$$\begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{Bmatrix} = \begin{bmatrix} 18.52 * 10^{-12} & -4.63 * 10^{-12} & 0 \\ -4.63 * 10^{-12} & 55.6 * 10^{-12} & 0 \\ 0 & 0 & 4.63 * 10^{-11} \end{bmatrix} \begin{Bmatrix} 459.3 * 10^{6} \\ 0.349 * 10^{6} \\ 3.17 * 10^{6} \end{Bmatrix}$$

Solve for $\varepsilon_1, \varepsilon_2$, and γ_{12} from the matrix shown above:

$$\varepsilon_1 = (18.52 * 10^{-12}) * (459.3 * 10^{6}) + (-4.63 * 10^{-12}) * (0.349 * 10^{6}) = 8.5 * 10^{-3}$$

$$\varepsilon_2 = (-4.63 * 10^{-12}) * (459.3 * 10^{6}) + (55.6 * 10^{-12}) * (0.349 * 10^{6}) = 2.11 * 10^{-3}$$

The shear strain of composite leaf spring can be obtained from the above matrix equation becomes:

$$\gamma_{12} = (4.63 * 10^{-11}) * (3.17 * 10^{6}) = 1.47 * 10^{-4}$$

Deflection of E-glass/Epoxy Spring

Due to the large deflection of spring it may be seem that the leaf spring is not under small deflection conditions and has large deformation and small strains behavior. Applying this behavior in the spring design causes a high degree of nonlinearly to the problem, so the optimization process will be computationally intense and may not converge. To ensure that the results of linearly elastic assumption are reliable, the optimized leaf spring is analyzed assuming the large deformations and small strains behavior.

The deformation of the E-glass/Epoxy composite leaf spring for half of the component only is analyzed here. Thus, the maximum deflection for leaf spring that given from strength of materials [Khurmi and Gupta, (2007)] can be obtained by:

$$\delta_{max} = \frac{4FL^3}{Ebh^3}$$

$$\delta_{max} = \frac{4 * 9487.5 * 800^3}{54000 * 34 * 54^3} = 67.2\ mm$$

The deflection of the composite leaf spring along its transverse direction is 67.2 mm, which is very small compared to the considered maximum deflection δ_{max} (105 mm).

3.7. Failure Criteria for Isotropic Leaf Spring Materials

Isotropic materials can fail in a variety of ways. Failure in such materials can be predicted by using one of the following failure criteria.

- ✓ For brittle materials
  - Maximum principal stress criterion
  - Maximum principal strain criterion
  - Coulomb-Mohr criterion
- ✓ For ductile materials
  - Maximum shear stress or Tresca criterion
  - Von-Mises stress or distortion energy criterion

To use these criteria, especially in multi- axial loading situation, the concept of principal stress and principal strains is quite frequently invoked.

3.7.1. Failure Criteria for Isotropic Composite Lamina

However, none of the failure criteria used for isotropic materials are of much use for predicting failure in composite lamina. This so, because the planes along which the lamina may be possibly the weakest, may not be necessarily aligned with the direction of "principal" stresses in a lamina. Thus, for the same reason, the concept of principal stresses is of little use in case of composite materials. For this reason, several alternative failure theories have been developed, which may be used to predict failure of composite lamina.

3.7.2. Different Strengths of a Composite Lamina

From Table 3, the strength of a unidirectional composite lamina may be characterized by five different material parameters. These are:

- Longitudinal tensile strength, σ_{u1}.
- Longitudinal compressive strength, $\sigma_{u1'}$.
- Transverse tensile strength, σ_{u2}.
- Transverse compressive strength, $\sigma_{u2'}$.
- In- plane shear strength, τ_{u12}.

Failure theories for composite lamina can be classified into three distinct groups. These are:

Non- interactive or limit theories: Here, failure modes are predicted by comparing individual stresses or strains respective to their ultimate stresses and strains. However, such theories do not account for inter play between different stress components. Examples of such theories are maximum stress criteria, and maximum strain criteria.

Interactive theories: These theories go a step further than limit theories, and also account for interaction between various stress/strain components. Examples of such theories are those of Tsai-Wu and Tsai-Hill. They are able to predict overall failure, but cannot predict the exact failure mode.

Failure mode based theories: These theories provide separate criteria for failure of matrix, fiber and interface. Examples of such theories are those of Puck, and Hashim-Rotem.

Here, we discussed non-interactive or limit theories in particular, maximum stress theories, maximum strain theories, and Tsai-Hill theories. But in this study, we are going to use only maximum stress theory.

3.7.3. Maximum Stress Theory

To use this theory:

We obtain stresses in material directions on a layer-by-layer basis. It is done; so, we have longitudinal, transverse and shear stresses as $\sigma_1 = 459.3\ MPa,\ \sigma_2 = 0.349\ MPa$ and $\tau_{12} = 3.17\ MPa$ for a unidirectional composite leaf spring.

Failure occurs, if at least one of the following conditions in any layer is satisfied.

- ✓ $\sigma_1 > \sigma_{u1'}$ if $\sigma_1 < 0$
- ✓ $\sigma_2 > \sigma_{u2'}$ if $\sigma_2 < 0$
- ✓ $\tau_{12} > \tau_{u12}$

When we combine with our system, the failure conditions become:

- ✓ $459.3\ Mpa < 1035\ Mpa;$ if $459.3\ Mpa > 0$
- ✓ $0.349\ Mpa < 138\ Mpa;$ if $0.349\ Mpa < 0$
- ✓ $3.17\ Mpa < 41\ Mpa$

Since the failure conditions for E-glass/Epoxy composite leaf spring material is not satisfied as shown above, the design is safe.

3.7.4. Fatigue Life of E-glass/Epoxy

A life data analysis method is analyzed using *Hwang and Han* relation. Two constants in their relation on the basis of experimental results are proposed. It is proved that the analytical formula predicts the fatigue life of component with E-glass/Epoxy composite material is given as follows:

$$N = \{ B\ (1 - r) \}^{1/c} \quad \text{(3.9)}$$

But, $r = \frac{\sigma_{max}}{\sigma_u}$

Where;

N is number of cycles to failure

B and *C* are constants,

B = 10.33 and *C = 0.14012*

r is applied stress level,

σ_{max} is maximum stress, and

σ_u is ultimate tensile strength.

The applied stress level is given by the following analytical formula:

$$r = \frac{\sigma_{max}}{\sigma_u} = \frac{473\ MPa}{1035\ MPa} = 0.457$$

Therefore, the estimation of fatigue life for the E-glass/Epoxy composite leaf spring material becomes:

$$N = \{\ 10.33\ (\ 1 - 0.457\ )\}^{1/_{0.14012}} = 2.2 * 10^{\ 5}\ \ cycles$$

The fatigue life of the designed single E-glass/Epoxy composite leaf spring is predicted and obtained as $N = 2.2 * 10^5\ cycles$. This shows the acceptable life or good resistance of the material to failure under fatigue loading.

3.8. Weight Optimization for Leaf Spring

The leaf spring suspension holds about 10-20% of vehicle un-sprung mass. Thus, it becomes an essential component for weight minimization [4]. The mass minimization can be accomplished by selecting better materials and optimized design of leaf spring [8]. An ideal composite leaf spring should be light in weight, low cost to manufacture, and reliable for service. Also, in recent year's improvement in reinforcing fibres and resin properties, fibre-reinforced composites are replacing an increasing number of metal components.

Therefore, to preserve natural resources and cost effects, weight minimization has been the major focus of automotive industries. Now a day's weight minimization can be achieved generally by the replacement of better material, design optimization and enhanced manufacturing process.

3.8.1. Mass of Leaf Spring

The design process for optimization, which is the task of designing the best structure to meet a particular goal, such as minimum weight. The objective for the design of the composite leaf spring is the minimization of weight. So, the objective function of the problem is given as weight of the leaf spring,

$$m = \rho AL = \rho\ x\ (n\ x\ b\ x\ h\ x\ L) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.10)$$

Where; ρ is density of the material, *n* is number of leaves,

b is width of the leaf spring,

h is thickness of the leaf springs, and

L is span length of spring.

Therefore, by using equation (3.10); mass of steel leaf spring is calculated as:

$$m_{Steel} = 7850\ x\ (8\ x\ 0.07\ x\ 0.007\ x\ 1.6) = 49\ kg$$

And also, mass of the composite E-Glass/Epoxy is given by:

$$m_{composite} = 2600\ x\ (1\ x\ 0.034\ x\ 0.054\ x\ 1.6)$$

$$m_{composite} = 7.64\ kg$$

3.8.2. Weight of Leaf Springs

The main reason for calculating mass was significant saving in weight of leaf spring; the results showed that the steel and E-glass/Epoxy leaf spring have a mass of 49 kg and 7.64 kg, while when we compare the mass of leaf spring; the E-glass/Epoxy composite leaf spring is much less in mass than the conventional steel leaf spring. It is known that weight reduction is one of the critical issues in vehicle design to reduce fuel consumption and to fulfill strict environmental regulations. The weight of an object is the force exerted upon it by gravity. Designating the weight as W and the acceleration due to gravity as g, we have: (taking assume $g = 10\ m/s^2$)

$$W = mg \qquad (3.11)$$

For steel;

$$W_{steel} = mg = 49\ kg\ \times 10\ m/s^2 = 490\ N$$

For E-glass/Epoxy;

$$W_{composite} = mg = 7.64\ kg\ \times 10\ m/s^2 = 76.4\ N$$

Finally, percent of weight saving (reduction) by the composite E-Glass/Epoxy leaf spring is calculated as follows:

% of less weight =100 – (Weight of E-glass/Epoxy spring ÷ Weight of steel spring) x 100

= 100 – (7.64 ÷ 49) x 100 = 84.4 %

3.9. Finite Element Method

The basis of FEA relies on the decomposition of the domain into a finite number of subdomains (elements) for which the systematic approximate solution is constructed by applying the variational or weighted residual methods. It is not always possible to obtain the exact analytical solution at any location in the body, especially for those elements having complex shapes or geometries. Always the most important are the boundary conditions and material properties. In such cases, the analytical solution that satisfies the governing equation or gives extreme values for the governing functional is difficult to obtain. Hence for most of the practical problems, the engineers resort to numerical methods like the finite element method to obtain approximate but most probable solutions. To analyze steel leaf spring, a stress analysis is performed using the finite element method and ANSYS WORKBENCH software. Also, analysis is carried out for composite leaf spring and the results are compared with existing steel leaf. The maximum deflection and shear stresses along the adhesive layer were measured. Finite element procedures at present day are widely used in engineering analysis.

Discretizing the domain: This step involves subdividing the domain into elements and nodes. For continuous system, this step is very important and the answers obtained are only approximate. In this case, the accuracy of the solution depends on the discretization used.

a. **Writing the element stiffness matrices*:*** The element stiffness equations need to be written for each element in the domain.
b. **Assembling the global stiffness matrix:** This is done using the direct stiffness approach to obtain the stiffness matrix for the entire system.
c. **Applying the boundary conditions:** This involves specifying the load conditions and restraints like supports and applied loads and displacements. In this study this step is performed manually.
d. **Solving the equation:** This is done by partitioning the global stiffness matrix and then solving the resulting equation using Gaussian elimination and the reactions and element stresses and visualization of the resulting solution.

Types of elements used:

- ✓ Linear bar element
- ✓ Truss element
- ✓ Beam element

In this study beam element is used since the leaf spring is assumed to a beam element with two or more supports. The beam element is a two-dimensional finite element where the local and global coordinates coincide. It is characterized by linear shape functions.

In the case of transverse loading, the beam element has modulus of elasticity E, moment of inertia I, and the length L. Each beam element has two nodes and is assumed to be horizontal as shown in Figure below.

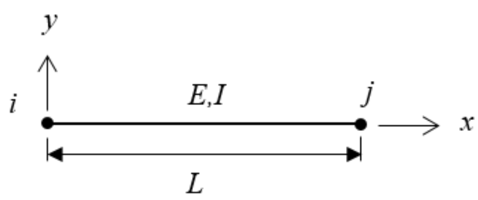


Figure 3.2: Representation of single beam element with its nodes

In this case the finite element matrix is given by the following matrix:

$$\frac{EI}{L^3}\begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix}\begin{Bmatrix} v_1 \\ \theta_1 \\ v_2 \\ \theta_2 \end{Bmatrix} = \begin{Bmatrix} F_1 \\ M_1 \\ \mathrm{F}_2 \\ \mathrm{M}_2 \end{Bmatrix} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.12)}$$

It is clear that the beam element has four degrees of freedom two at each node (a transverse displacement and a rotation). The global stiffness matrix K is obtained; the structure equation is given as follows:

$\{F\} = \{K\}\ \{U\}$

Where, $\{U\}$ is the global nodal displacement vector, and

$\{F\}$ is the global nodal force vector.

At this step the boundary conditions are applied manually to the vectors $\{U\}$ and $\{F\}$. Then the matrix equation is solved by partitioning and Gaussian elimination. The beam element developed in this chapter is based on the Euler–Bernoulli beam theory that is applicable for thin beams.

3.9.1. Finite Element Analysis of Leaf spring

The leaf spring is like a simply supported beam with a concentrated load at its center of the beam along its length. But, since the leaf spring is fixed with the axle at its center, only half of it is considered for analysis purpose. This means FEA is performed for cantilever beam with concentrated load at the free-end of leaf spring.

a) **Load analysis:** When the vehicle is loaded, the maximum load on the vehicle is distributed to four places of vehicle leaf springs. Since, we analyzing only one leaf spring, and it assumed to be uniformly concentrated load acting at the free-end along the half length of the component.
b) **Boundary conditions:** The leaf spring is assumed to be acting as a fixed-free beam under FEM conditions. Since the leaf spring center is fixed with axle, it is assumed that a uniformly concentrated load is acting on all the nodes except the end nodes which are fixed.
c) **Finite element analysis:** The conventional deign method gives the geometry of the leaf spring, and the load conditions are obtained after load analysis. These results are finally used to carry out the finite element analysis of the leaf spring on the basis of the assumed boundary conditions.

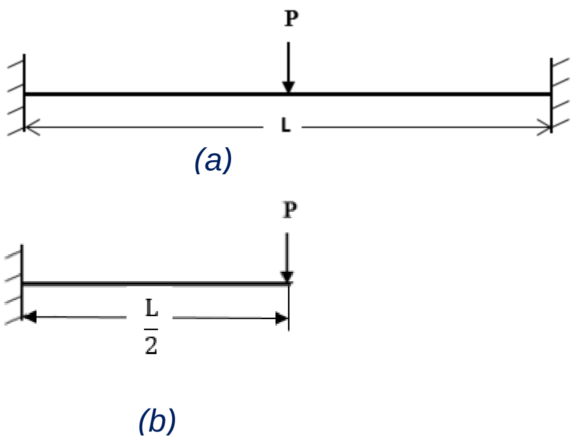


Figure 3.3: Beam element for analyzing the leaf spring :(a) Simply supported type, (b) Cantilever type

First, the analysis is carried out with overloading of the vehicle. The leaf spring is discretized using beam elements. Beam members can support applied concentrated load and nodal reaction load. Therefore, we must be able to account for concentrated loading. Consider the cantilever beam subjected to a uniformly concentrated loading. In general, cantilever reactions are those reactions at the one end fixed and in the other end is free loaded. If the one fixed ends of the element are assumed to be fixed that means, displacements and rotations at one ends are prevented.

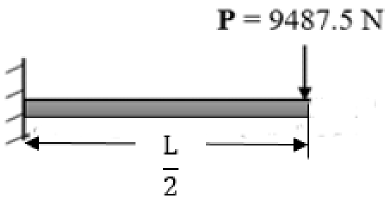


Figure 3.4: Loading conditions of cantilever beam

3.9.2. FEM Equations for Beam Member

In planar beam elements, there are two degrees of freedom (DOFs) at a node in its local coordinate system. They are deflection in the *y* direction, *v*, and rotation in the *x*–*y* plane, θ with respect to the *z*-axis. Therefore, each beam element has a total of four DOFs. The beam element developed in this FEM is based on the Euler-Bernoulli beam theory that is applicable for thin beams [38].

3.9.3. Shape Function Construction for Beam

Consider a beam element of length, *L* with nodes 1 and 2 at each end of the element, as shown in Figure below. The local *x*-axis is taken in the axial direction of the element with its origin at the middle section of the beam. Similar to all other structures, to develop the FEM equations, shape functions for the interpolation of the variables from the nodal variables would first have to be developed [38].

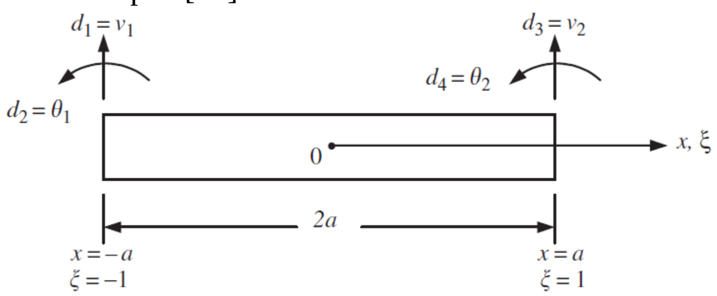


Figure 3.5: Beam element and its local coordinate systems: physical coordinates x, and natural coordinates ξ [38].

There are four DOFs for a beam element, there should be four shape functions. It is often more convenient if the shape functions are derived from a special set of local coordinates, which is commonly known as the *natural coordinate system*. This natural coordinate system has its origin at the center of the element, and the element is defined from −1 to +1, as shown in Figure above for finite element analysis of leaf spring. The relationship between the natural coordinate system and the local coordinate system can be simply given as [38]:

$$\xi = \frac{x}{a} \quad \ldots\ldots (3.13)$$

To derive the four shape functions in the natural coordinates, the displacement in an element is first assumed in the form of a third order polynomial of ξ that contains four unknown constants:

$$v(\xi) = \alpha_0 + \alpha_1\xi + \alpha_2\xi^2 + \alpha_3\xi^3 \quad \ldots\ldots (3.14)$$

where α_0 to α_3 are the four unknown constants. The third order polynomial is chosen because there are four unknowns in the polynomial, which can be related to the four nodal DOFs in the beam element. The above equation can have the following matrix form:

$$v(\xi) = [1 \quad \xi \quad \xi^2 \quad \xi^3]\begin{Bmatrix}\alpha_0\\ \alpha_1\\ \alpha_2\\ \alpha_3\end{Bmatrix} \quad \ldots\ldots (3.15)$$

or

$$v(\xi) = p^T(\xi)\alpha \quad \ldots\ldots (3.16)$$

Where; **p** is the vector of basis functions and $\boldsymbol{\alpha}$ is the vector of coefficients. The rotation θ can be obtained from the differential of equation (3.14) with the use of equation (3.13):

$$\theta = \frac{\partial v}{\partial x} = \frac{\partial v}{\partial \xi}\frac{\partial \xi}{\partial x} = \frac{1}{a}\frac{\partial v}{\partial \xi} = \frac{1}{a}\left(\alpha_1\xi + 2\alpha_2\xi + 3\alpha_3\xi^2\right) \quad \ldots\ldots (3.17)$$

The four unknown constants α_0 to α_3 can be determined by utilizing the following four conditions:

At $x = -a$ or $\xi = -1$:

$$\left.\begin{aligned}&(1)\ v(-1) = v_1\\ &(2)\ \left.\frac{dv}{dx}\right|_{\xi=-1} = \theta_1\end{aligned}\right\} \quad \ldots\ldots (3.18)$$

At $x = +a$ or $\xi = +1$:

$$\left.\begin{aligned}&(1)\ v(1) = v_2\\ &(2)\ \left.\frac{dv}{dx}\right|_{\xi=1} = \theta_2\end{aligned}\right\} \quad \ldots\ldots (3.19)$$

The application of the above four conditions gives:

$$\begin{Bmatrix} v_1 \\ \theta_1 \\ v_2 \\ \theta_2 \end{Bmatrix} = \begin{bmatrix} 1 & -1 & 1 & -1 \\ 0 & 1/a & -2/a & 3/a \\ 1 & 1 & 1 & 1 \\ 0 & 1/a & 2/a & 3/a \end{bmatrix} \begin{Bmatrix} \alpha_0 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \end{Bmatrix} \quad \text{.......................... (3.20)}$$

or

$$\mathrm{d}_e = \mathrm{A}_e\, \alpha \quad \text{.. (3.21)}$$

Solving the above equation for α gives:

$$\alpha = \mathrm{A}_e{}^{-1}\ \mathrm{d}_e \quad \text{.. (3.22)}$$

Where

$$\mathrm{A}_e = \frac{1}{4} \begin{bmatrix} 2 & a & 2 & -a \\ -3 & -a & 3 & -a \\ 0 & -a & 0 & a \\ 1 & a & -1 & a \end{bmatrix} \quad \text{.. (3.23)}$$

Hence, substituting equation (3.22) into equation (3.16) will give:

$$v = N_{(\xi)}\, d_e \quad \text{.. (3.24)}$$

where **N** is a matrix of shape functions given by:

$$N_{(\xi)} = P\mathrm{A}_e{}^{-1} = \begin{bmatrix} N_{1(\xi)} & N_{2(\xi)} & N_{3(\xi)} & N_{4(\xi)} \end{bmatrix} \quad \text{...................... (3.25)}$$

in which the shape functions are found to be:

$$\left.\begin{aligned} N_{1(\xi)} &= \frac{1}{4}(2 - 3\xi + \xi^3) \\ N_{2(\xi)} &= \frac{1}{4}a\,(1 - \xi - \xi^2 + \xi^3) \\ N_{3(\xi)} &= \frac{1}{4}(2 + 3\xi - \xi^3) \\ N_{4(\xi)} &= \frac{a}{4}(-1 - \xi + \xi^2 + \xi^3) \end{aligned}\right\} \quad \text{.. (3.26)}$$

It can be easily confirmed that the two translational shape functions N_1 and N_3 satisfy conditions shape functions of delta function which is linearly independent and partitions of unity. However, the two-rotational shape functions N_2 and N_4 do not satisfy the delta function and partitions of unity. This is because these two shape functions relate to rotational degrees of freedom, which are derived from the deflection functions. Satisfaction of N_1 and N_3 has already ensured the correct representation of the rigid body movement of the beam element.

3.9.4. Strain Matrix

Having now obtained the shape functions, the next step would be to obtain the element strain matrix, so we have:

$$\varepsilon_{xx} = \frac{\partial v}{\partial x} \quad \text{...} \quad (3.27)$$

Substituting equation (3.24) into equation (3.27), which gives the relationship between the strain and the deflection, we have:

$$\varepsilon_{xx} = \mathrm{B} \; d_e \quad \text{...} \quad (3.28)$$

where the strain matrix B is given by:

$$B = -y\,L\,N = -y\,\frac{\partial^2}{\partial x^2}\,N = -\frac{y}{a^2}\,\frac{\partial^2}{\partial \xi^2} = -\frac{y}{a^2}\,N'' \quad \text{..................} \quad (3.29)$$

Where; $L = \frac{\partial^2}{\partial x^2}$ is the differential operator.

In deriving the above equation, equations (3.13) have been used. From equation (3.26), we have:

$$N'' = [N_1{}'' \quad N_2{}'' \quad N_3{}'' \quad N_4{}''\,] \quad \text{..} \quad (3.30)$$

Where

$$\left.\begin{aligned} N_1'' &= \frac{3}{2}\,\xi, & N_2'' &= \frac{a}{2}\,(-1+3\xi) \\ N_3'' &= -\frac{3}{2}\,\xi, & N_4'' &= \frac{a}{2}\,(1+3\xi) \end{aligned}\right\} \quad \text{.......................................} \quad (3.31)$$

3.9.5. Element Matrices

Having obtained the strain matrix, we are now ready to obtain the element stiffness and mass matrices. By substituting equation (3.30) into element stiffness matrix and will obtaining the following equation as:

$$\mathrm{k}_e = \int_V \mathrm{B}^T \mathrm{cB}\, d\mathrm{V} = E\int_A \mathrm{y}^2\, d\mathrm{A} \int_{-a}^{a} \left(\frac{\partial^2}{\partial x^2}\,N\right)^T \left(\frac{\partial^2}{\partial x^2}\,N\right) d\mathrm{x}$$

$$\mathrm{k}_e = EI\int_{-1}^{1} \frac{1}{a^4}\left[\frac{\partial^2}{\partial \xi^2}\,N\right]^T \left[\frac{\partial^2}{\partial \xi^2}\,N\right] a\,d\xi$$

$$\mathrm{k}_e = \frac{EI}{a^3}\int_{-1}^{1} N''^T\, N'' d\xi \quad \text{...} \quad (3.32)$$

where $I = \int_A y^2 \, dA$ is the second moment of area (or moment of inertia) of the cross section of the beam with respect to the *z*-axis. Substituting equation (3.30) into (3.32), we obtain:

$$k_e = \frac{EI}{a^3}\int_{-1}^{1}\begin{bmatrix} N_1''N_1'' & N_1''N_2'' & N_1''N_3'' & N_1''N_4'' \\ N_2''N_1'' & N_2''N_2'' & N_2''N_3'' & N_2''N_4'' \\ N_3''N_1'' & N_3''N_2'' & N_3''N_3'' & N_3''N_4'' \\ N_4''N_1'' & N_4''N_2'' & N_4''N_3'' & N_4''N_4'' \end{bmatrix} dx \quad \text{………………... (3.33)}$$

Evaluating the integrals in the above equation leads to get the stiffness matrix shown below.

$$k_e = \frac{EI}{2a^3}\begin{bmatrix} 3 & 3a & -3 & 3a \\ 3a & 4a^2 & -3a & 2a^2 \\ -3 & -3a & 3 & -3a \\ 3a & 2a^2 & -3a & 4a^2 \end{bmatrix} \quad \text{…………………………………….. (3.34)}$$

The other element matrix would be the force vector. The nodal force vector for beam elements can be obtained. Suppose the element is loaded by an external distributed force f_y along the *x*-axis, two concentrated forces f_{s1} and f_{s2}, and concentrated moments m_{s1} and m_{s2}, respectively, at nodes 1 and 2; the total nodal force vector becomes:

$$f_e = \int_V N^T f_b \, dV + \int_{S_f} N^T f_s \, dS_f$$

$$f_e = f_y a \int_{-1}^{1}\begin{bmatrix} N_1 \\ N_2 \\ N_3 \\ N_4 \end{bmatrix} d\xi + \begin{Bmatrix} f_{s1} \\ m_{s1} \\ f_{s2} \\ m_{s2} \end{Bmatrix} \quad \text{……………………………….……………. (3.35)}$$

Consider a leaf spring with uniform cantilever beam subjected to a downward force. The beam is fixed at one end, and it has a uniform cross-sectional area as shown. The beam undergoes static deflection by a downward load of *P* = 9487.5N applied at the free end. The dimensions of the beam are shown in the figure, and the beam is made of composite E-Glass/Epoxy whose properties are given in Table 3. To make clear the steps involved in solving this problem, we first used just one beam element to solve for the deflection.

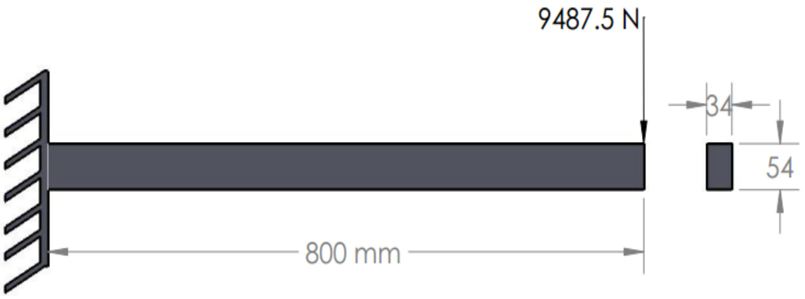


Figure 3.6: Cantilever beam under static load.

Obtaining the element matrices

The first step in formulating the finite element equations is to form the element matrices and, in this case, we have only one element matrices; that is, the element matrices are actually the global finite element matrices.

The second moment of area of the cross-sectional area about the *z*-axis can be given as:

$$\mathrm{I} = \frac{\mathrm{bh}^3}{12} = \frac{0.034 * 0.054^3}{12} = 4.46 * 10^{-7}\ \mathrm{m}^4$$

$\frac{\mathrm{L}}{2}$ = 0.8 m = 800 mm

P = 9487.5 N

Since only one element is used, the stiffness matrix of the beam is thus the same as the element stiffness matrix. From equation (3.34), we have the element stiffness matrix compatible to my system is given by:

$$\mathrm{k}_e = \frac{\mathrm{EI}}{2a^3}\begin{bmatrix} 3 & 3a & -3 & 3a \\ 3a & 4a^2 & -3a & 2a^2 \\ -3 & -3a & 3 & -3a \\ 3a & 2a^2 & -3a & 4a^2 \end{bmatrix}$$

$$\mathrm{K} = \mathrm{k}_e = \frac{(54 * 10^9 * 4.46 * 10^{-7})}{2 * 0.4^3}\begin{bmatrix} 3 & 1.2 & -3 & 1.2 \\ 1.2 & 0.64 & -1.2 & 0.32 \\ -3 & -1.2 & 3 & -1.2 \\ 1.2 & 0.32 & -1.2 & 0.64 \end{bmatrix}$$

$$\mathrm{k}_e = 0.188 * 10^6 \begin{bmatrix} 3 & 1.2 & -3 & 1.2 \\ 1.2 & 0.64 & -1.2 & 0.32 \\ -3 & -1.2 & 3 & -1.2 \\ 1.2 & 0.32 & -1.2 & 0.64 \end{bmatrix} \mathrm{Nm}^{-2}$$

The finite element equation becomes:

$$\underbrace{0.188 * 10^6 \begin{bmatrix} 3 & 1.2 & -3 & 1.2 \\ 1.2 & 0.64 & -1.2 & 0.32 \\ -3 & -1.2 & 3 & -0.9 \\ 1.2 & 0.32 & -1.2 & 0.64 \end{bmatrix}}_{\mathrm{K}} \underbrace{\begin{Bmatrix} v_1 \\ \theta_1 \\ v_2 \\ \theta_2 \end{Bmatrix}}_{\mathrm{D}} = \underbrace{\begin{Bmatrix} F_1 = ? \\ M_1 = ? \\ \mathrm{F}_2 = P \\ \mathrm{M}_2 = 0 \end{Bmatrix}}_{\mathrm{F}}$$

At node 1, the beam is clamped. Therefore, the shear force and moment at this node should be the reaction force and moment, which are unknowns before the FEM equation is solved for the displacements. To solve the unknown's reaction shear force and reaction moment, we need to impose the displacement boundary condition at the clamped node.

Applying boundary conditions: The beam is fixed or clamped at one end. This implies that at that end, the deflection, ν_1, and the slope, θ_1, are both equal to zero:

$\nu_1 = \theta_1 = 0$

The imposition of the above displacement boundary condition leads to the removal of the first and second rows and columns of the stiffness matrix:

$$0.188 * 10^6 \begin{bmatrix} 3 & 0.9 & -3 & 1.2 \\ 1.2 & 0.64 & -1.2 & 0.32 \\ -3 & -1.2 & 3 & -1.2 \\ 1.2 & 0.32 & -1.2 & 0.64 \end{bmatrix} \begin{Bmatrix} \nu_1 = 0 \\ \theta_1 = 0 \\ \nu_2 \\ \theta_2 \end{Bmatrix} = \begin{Bmatrix} F_1 \\ M_1 \\ \mathrm{F}_2 = P \\ \mathrm{M}_2 = 0 \end{Bmatrix}$$

The reduced stiffness matrix becomes a 2×2 matrix of:

$$\mathrm{K} = 0.188 * 10^6 \begin{bmatrix} 3 & -1.2 \\ -1.2 & 0.64 \end{bmatrix} \mathrm{Nm}^{-2}$$

The finite element equation, after the imposition of the displacement condition, is thus:

$$\mathrm{Kd} = \mathrm{F}$$

Where;

$$\mathrm{d}^T = [\nu_2 \quad \theta_2]$$

and the force vector **F** is given as:

$$\mathrm{F} = \begin{bmatrix} -9487.5 \\ 0 \end{bmatrix} \mathrm{N}$$

Solving the FE matrix equation: The last step in this problem would be to solve the finite element equation to obtain ν_2 and θ_2. In this case, finite element equation is actually two simultaneous equations involving two unknowns, and can be obtained:

$$0.188 * 10^6 \begin{bmatrix} 3 & -1.2 \\ -1.2 & 0.64 \end{bmatrix} \begin{Bmatrix} \nu_2 \\ \theta_2 \end{Bmatrix} = \begin{bmatrix} -9487.5 \\ 0 \end{bmatrix}$$

$$\begin{cases} 0.188 * 10^6 (3\nu_2 - 1.2\theta_2) = -9487.5 \\ 0.188 * 10^6 (-1.2\nu_2 + 0.64\theta_2) = 0 \end{cases}$$

$$\nu_2 = -0.06728 \text{ mm}, \qquad \text{and } \theta_2 = -0.1262 \text{ rad}$$

After ν_2 and θ_2 have been obtained, they are substituted back into the first two equations of finite element to obtain the reaction shear force at node 1:

$$F_1 = 0.188 * 10^6 \, [-3\nu_2 + 1.2\theta_2]$$
$$= 0.188 * 10^6 \, [-3 * (-0.06728) + 1.2 * (-0.1262)] = 9475 \text{ N}$$

and the reaction moment at node 1:

$$M_1 = 0.188 * 10^6 [-1.2\nu_2 + 0.32\theta_2]$$
$$= 0.188 * 10^6 [-1.2 * (-0.06728) + 0.32 * (-0.1262)]$$
$$= 7586 \text{ N.m}$$

We again observe the reproduction feature of the FEM. In this case, it is because the exact solution of the deflection for the cantilever thin beam is a third order polynomial, which can be obtained easily by solving the strong form of the system equation of beam with $f_y = 0$. On the other hand, the shape functions used in our FEM analysis are also third order polynomials. Therefore, the exact solution of the problem is included in the set of assumed deflections. The FEM based on Hamilton's principle has indeed reproduced the exact solution. This is, of course, also true if we were to calculate the deflection at anywhere else other than the nodes. To compute the deflection at the center of the beam, we can use equation (3.24) with *x* = 0.4, or in the natural coordinate system, $\xi = 0$, and substituting the values calculated at the nodes:

$$\nu_{(\xi=0)} = N_{(\xi=0)} \, d_e = \begin{bmatrix} N_{1(\xi=0)} & N_{2(\xi=0)} & N_{3(\xi=0)} & N_{4(\xi=0)} \end{bmatrix} \begin{Bmatrix} \nu_1 \\ \theta_1 \\ \nu_2 \\ \theta_2 \end{Bmatrix}$$

But, from equation (3.26), we have shape function equations and substitute *x* = 0.4 and $\xi = 0$ in this equation and get:

$$\nu_{(\xi=0)} = \begin{bmatrix} \frac{1}{2} & \frac{1}{10} & \frac{1}{2} & -\frac{1}{10} \end{bmatrix} \begin{Bmatrix} 0 \\ 0 \\ -0.06728 \\ -0.1262 \end{Bmatrix}$$

$$= \frac{1}{2}(-0.06728) - \frac{1}{10} \, (-0.1262) = -2.1 * 10^{-2} \, mm$$

To calculate the rotation at the center of the beam, the derivatives of the shape functions are used as follows:

$$\theta_{(\xi=0)} = \left(\frac{dv}{dx}\right)_{(\xi=0)} = \left(\frac{dN}{dx}\right)_{(\xi=0)} d_e$$

$$= \left[\left(\frac{dN}{dx}\right)_{(\xi=0)} \quad \left(\frac{dN}{dx}\right)_{(\xi=0)} \quad \left(\frac{dN}{dx}\right)_{(\xi=0)} \quad \left(\frac{dN}{dx}\right)_{(\xi=0)}\right] \begin{Bmatrix} v_1 \\ \theta_1 \\ v_2 \\ \theta_2 \end{Bmatrix}$$

$$= \left[-\frac{3}{4} \quad -\frac{1}{10} \quad \frac{3}{4} \quad -\frac{1}{10}\right] \begin{Bmatrix} 0 \\ 0 \\ -0.06728 \\ -0.1262 \end{Bmatrix}$$

$$= \frac{3}{4}(-0.06728) - \frac{1}{10}(-0.1262)$$

$$\theta_{(\xi=0)} = -3.78 * 10^{-2}\, rad$$

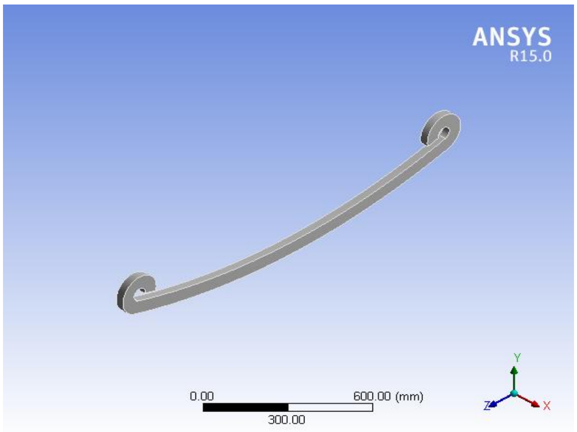
Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. In this section developing the 3D of original and optimized leaf spring model and the FEA technique are described in details. Von-Mises stress, shear stress, total deformation and life of original leaf spring and optimized leaf spring are done.

3.9.6. Procedure of static Analysis

First, I prepare a model of leaf spring in SOLIDWORK software and save as Parasolid file format for analysis of leaf spring in ANSYS WORKBENCH 15 and import the Parasolid file model into ANSYS WORKBENCH simulation module.

a. Applying material for Leaf Spring

Material type: - E-glass/Epoxy	Material type: - Steel 65Si7
Density: - 2600 Kg/m^3	Density: - 7850 Kg/m^3
Young's modulus: - 54 GPa	Young's modulus: - 200 GPa
Poisson ratio: - 0.25	Poisson ratio: - 0.3
Longitudinal tensile stress: - 1035 MPa	Yield tensile stress: - 1272 MPa
Transverse tensile strength: - 28 MPa	Ultimate tensile strength: - 1158 MPa
Longitudinal compressive stress: - 1035 MPa	
Transverse compressive stress: - 138 MPa	



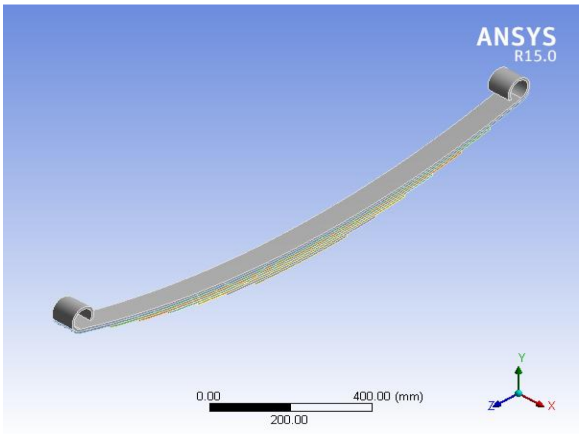


Figure 3.7: Model of leaf spring for Modified and Existing

b. Meshing of Leaf Spring:

The FE model of the leaf spring geometry is meshed with tetrahedral elements. Mesh refinement are done on the leaf spring for optimized material we are use, so that fine mesh is generally obtained for leaf spring.

Tetrahedral shape of element is used for meshing the imported complex geometries to the ANSYS WORKBENCH software. Three-dimensional model (3D) of leaf spring is performed in SOLIDWORK. After that, model is exported by ANSYS and profile is subdivided into nodes and elements. Collection of elements is called mesh and it is necessary to make mesh optimization to get more accuracy results. Mesh optimization is carried out until the FEA results and analytical solutions are close to each other.

Meshing of Model: We discretized the solid model into small elements. Depending upon the requirement of the accuracy of results the fineness of meshing varies. Finer is the meshing more we are closer to the actual results and when mesh size increases maximum stress on the component become decreased.

Mesh statics of leaf spring:

Table 4: Mesh statics for modified and existing leaf spring.

Description	**Mesh statics**	
	Modified leaf spring	**Existing leaf spring**
Type of element	Tetrahedral shape	Tetrahedral shape
Number of nodes	2141671	999727
Number of elements	499766	191064

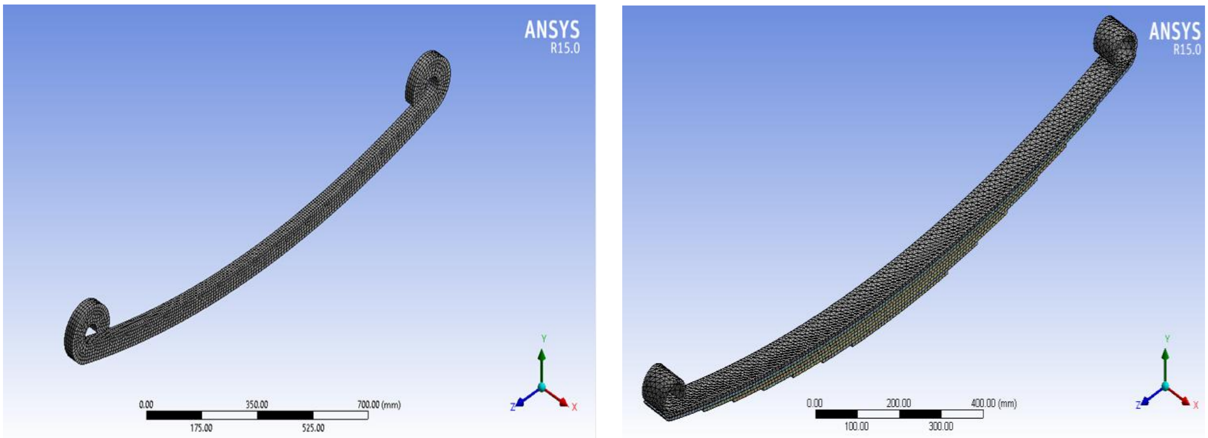


Figure 3.8: Meshing of leaf spring with rectangular shape of elements for Modified and Existing

The most commonly used finite element shapes are triangular and quadrilateral for two dimensional (2-D) problems and tetrahedral for three-dimensional (3-D) problems.

The triangular and tetrahedral edge elements have the advantage of being able to model very complex geometries. This meshing is performed by 3mm element size for original leaf spring and 2mm for optimized leaf spring. Finer is the meshing more we are closer to the actual results and when mesh size increases maximum stress on the component become decreased.

c. Defining boundary condition and applied load for analysis:

Load applied on the component is obtained by considering overloading position of the vehicle, which is load data were taken from the calculated result of the total weight of the vehicle. Boundary condition is based on under supporting condition of leaf spring.

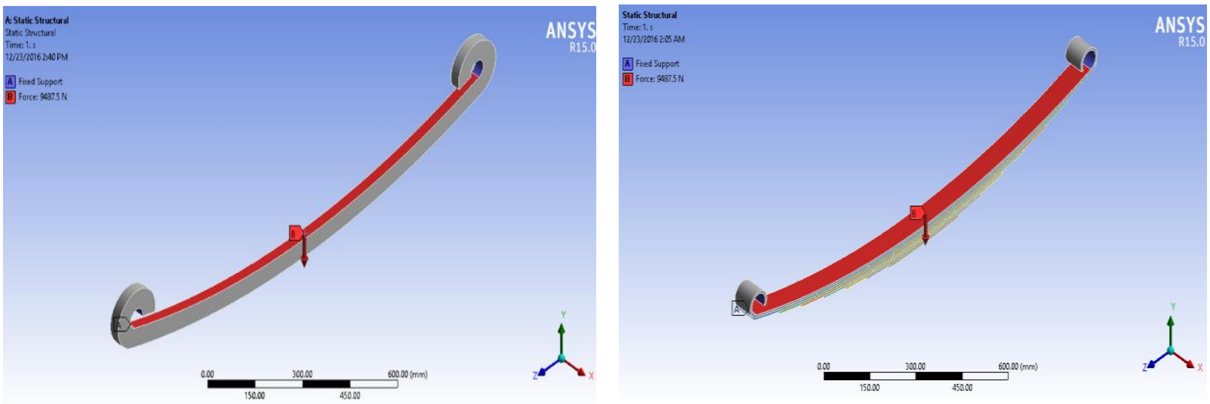


Figure 3.9: Loading and boundary conditions of Modified and Existing leaf spring

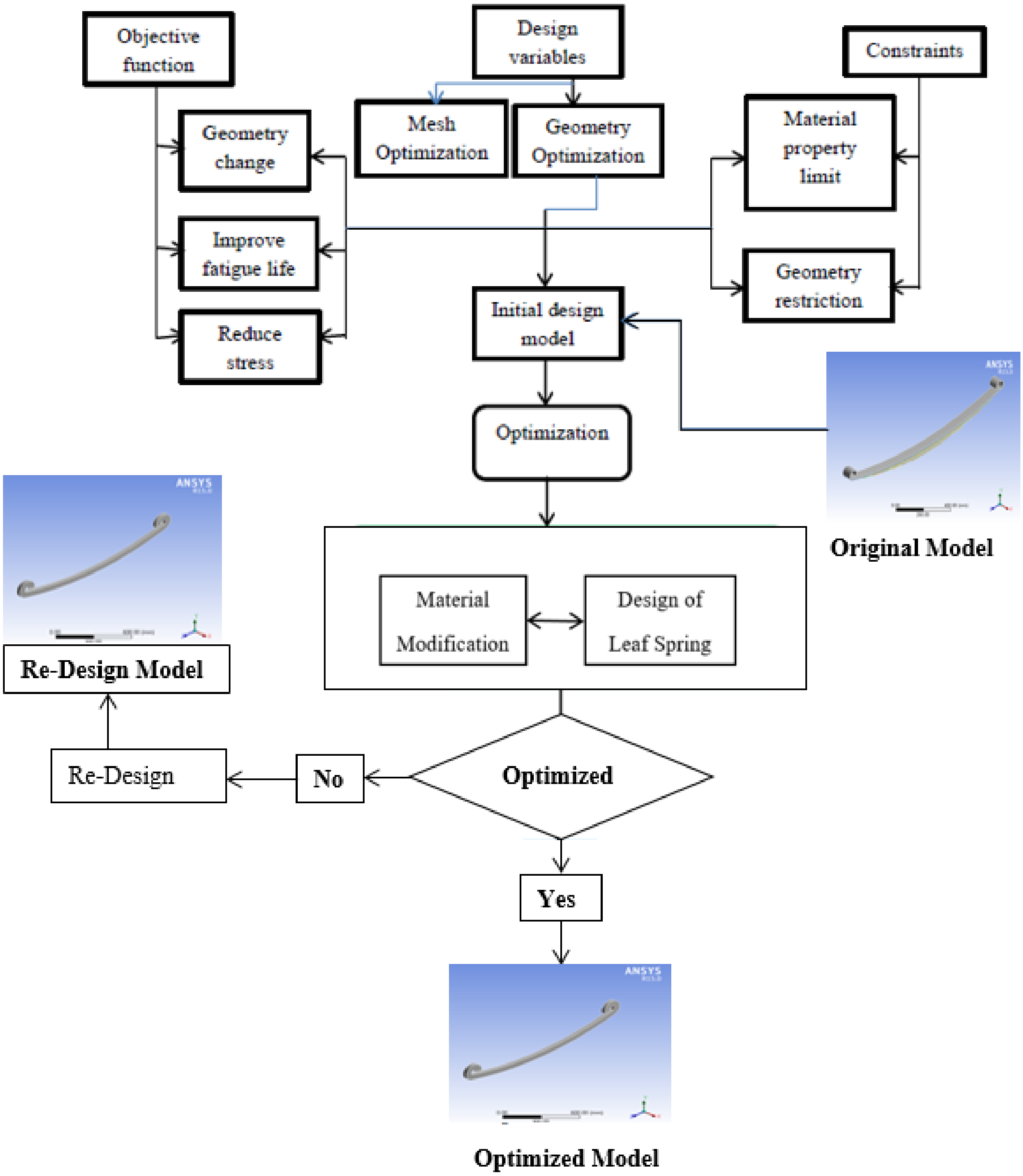


Figure 3.10: Flow Chart to Achieve the Objective Function

CHAPTER FOUR

4. RESULTS AND DISCUSSIONS

4.1. Results

In this present work, the modified and existing leaf spring model was created by Solid work software. Then, the model created by solid work was imported to ANSYS WORKBENCH software for analysis.

Stress for Modified and Existing Leaf Spring

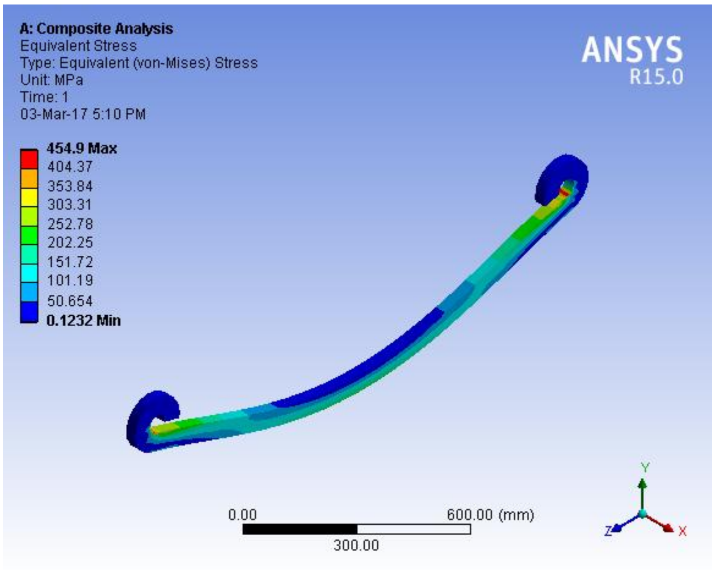


Figure 4.1: (a) Von-Mises stress of Modified model

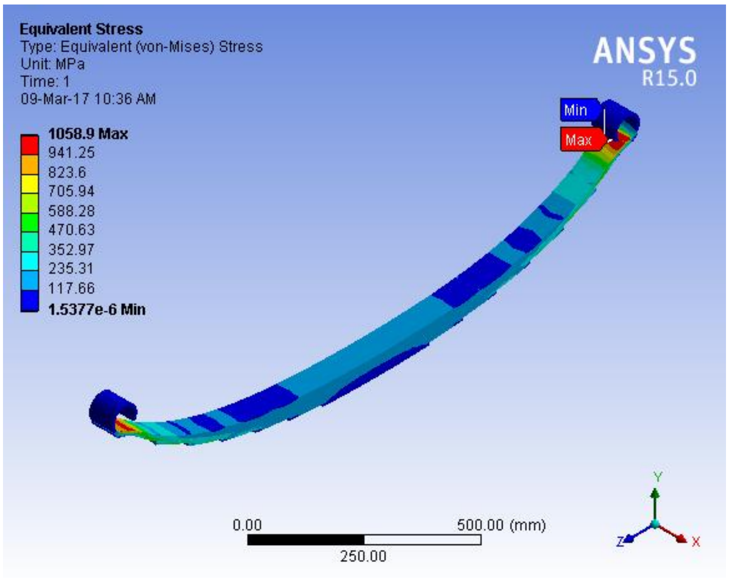


Figure 4.1: (b) Von-Mises stress of Existing model

The results shown above indicates the maximum Von-Mises stress generated in conventional steel leaf spring is $1058.9\ MPa$ and E-glass/Epoxy material is $454.9\ MPa$.

Shear stress for Leaf spring

The shear stresses in the E-Glass/Epoxy composite leaf spring are much lower than that of the steel spring.

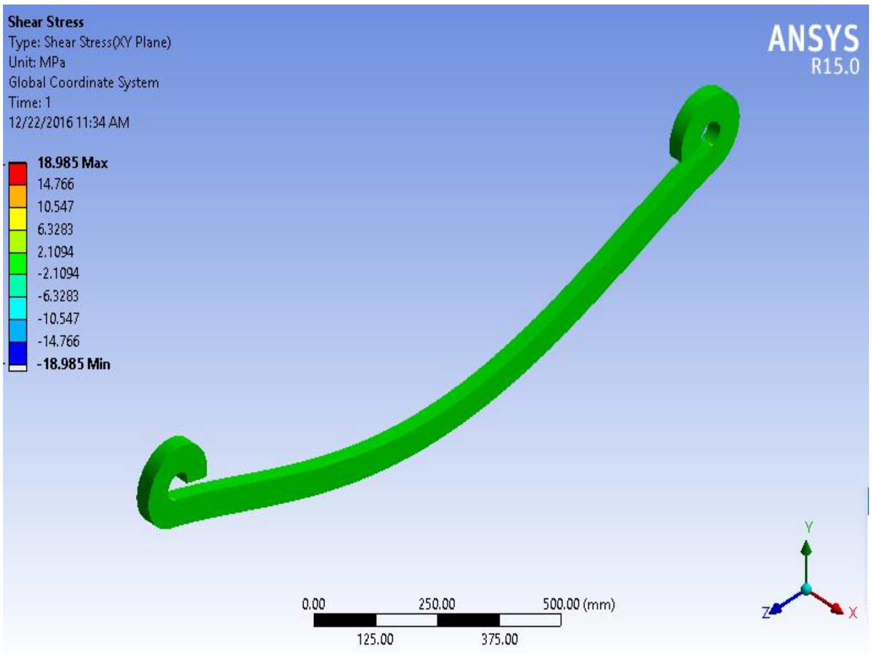


Figure 4.2: (a) Shear stress of Modified Spring

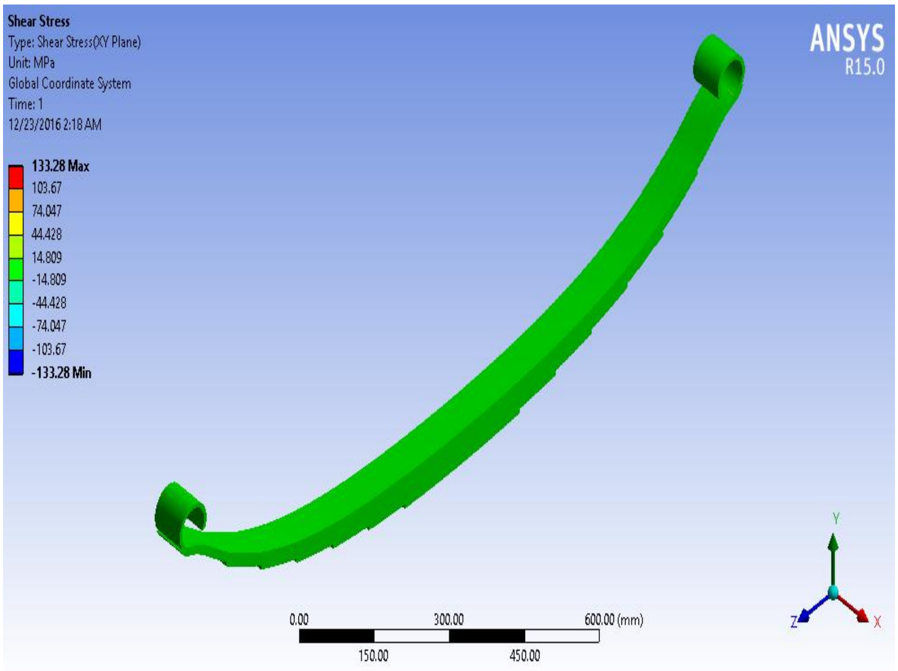


Figure 4.2: (b) Shear stress of Existing Spring

As per the results shown above the shear stress generated in conventional steel leaf spring is 133.28 MPa, E-glass/Epoxy composite material is 18.98 MPa.

Strain Energy of leaf spring

The potential energy stored in leaf spring as strain energy and then released slowly. So, here the strain energy for the E-glass/Epoxy is lower than the conventional steel leaf spring.

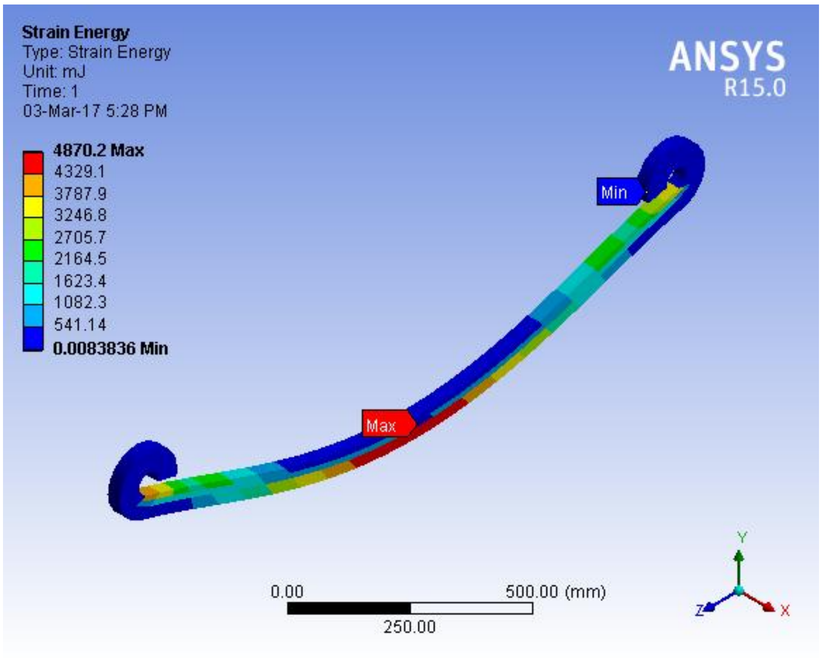


Figure 4.3: (a) Strain Energy of Modified Spring

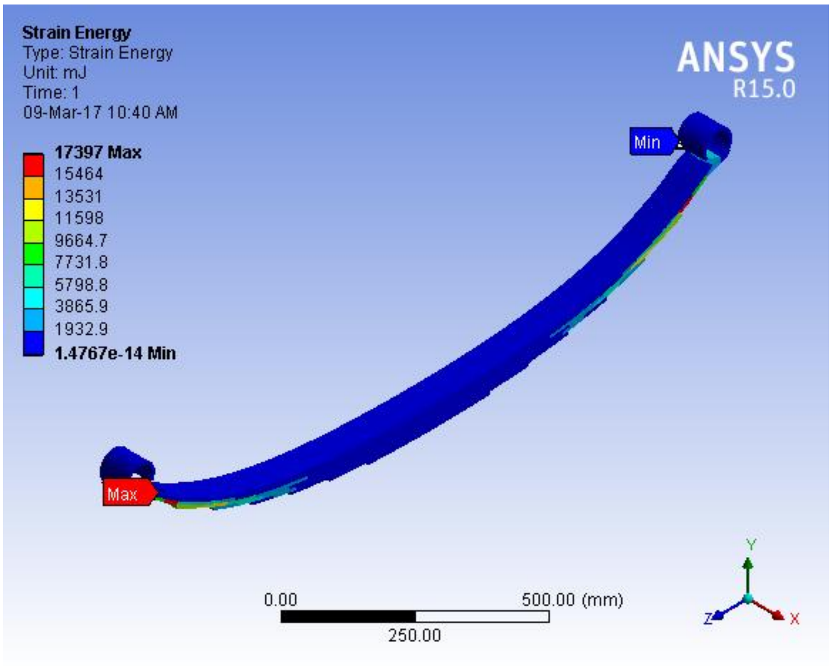


Figure 4.3: (b) Strain Energy of Existing Spring

Total deformation

The deformation of conventional leaf spring is much higher than that of E-glass/Epoxy leaf spring.

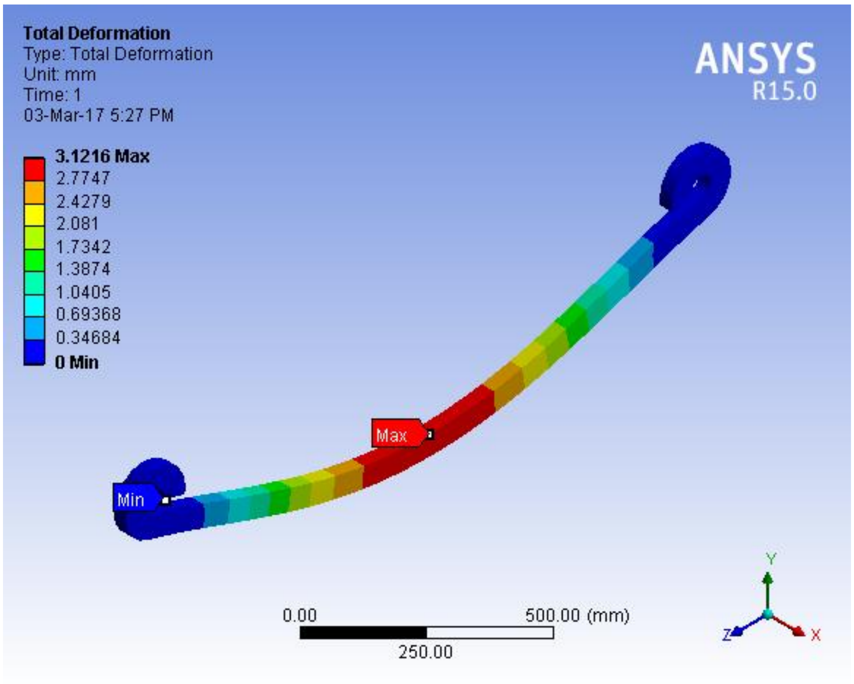


Figure 4.4: (a)Total Deformation generated for E-glass/Epoxy

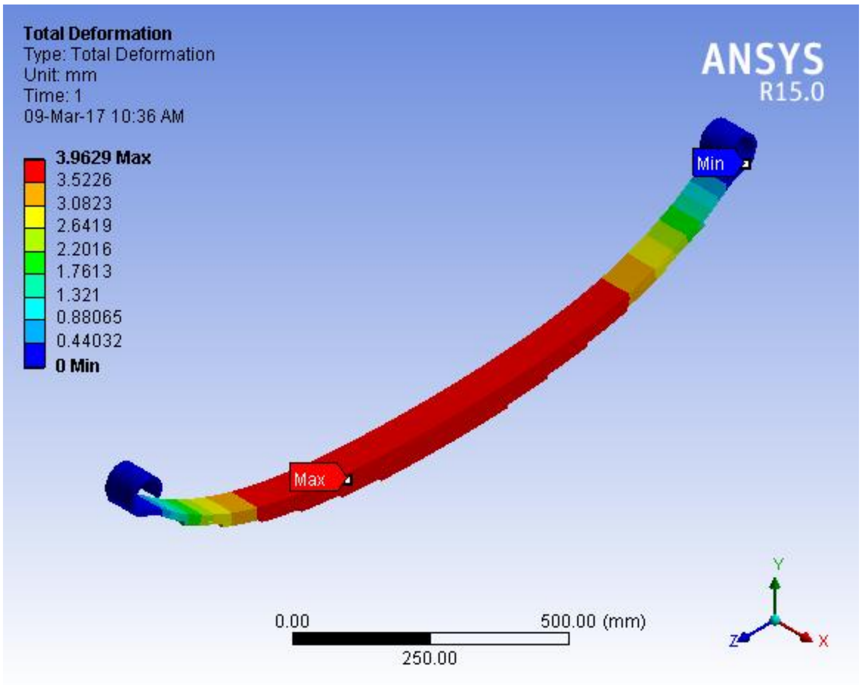


Figure 4.4: (b) Total Deformation generated for Steel Leaf Spring

The above result shows the improved strength and comfort level as low deflection for the leaf spring which is better in case of E-glass/Epoxy composite material, but we are still looking for the possible weight reduction. The mass of the conventional leaf spring is 49 kg and E-glass/Epoxy composite leaf spring is found as 7.64 kg, which means E-glass/Epoxy composite leaf spring reduce the weight by 84.4 % from conventional one.

Life

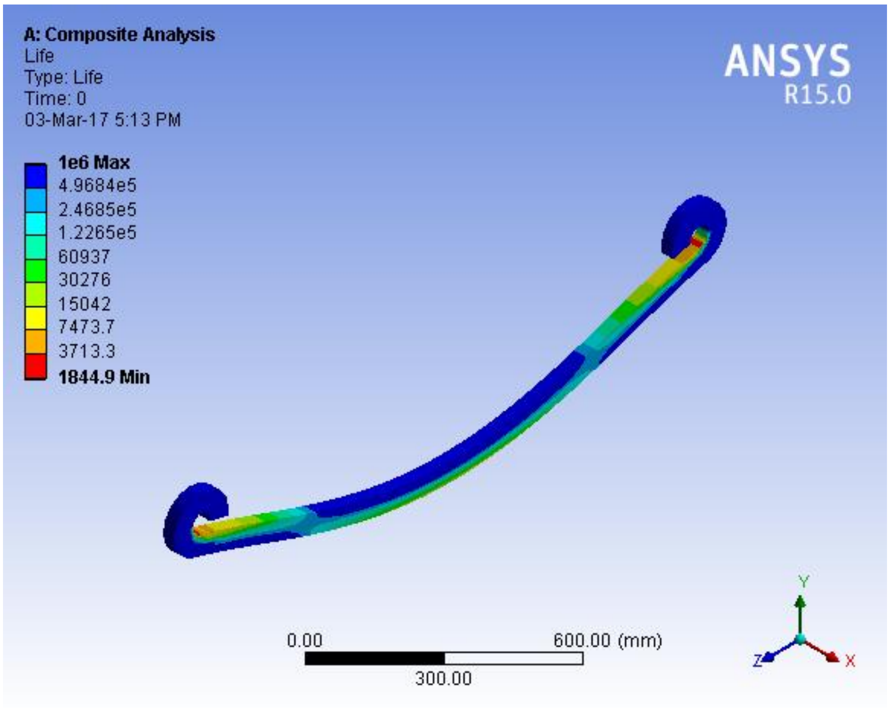


Figure 4.5: (a) Life generated for E-glass/Epoxy

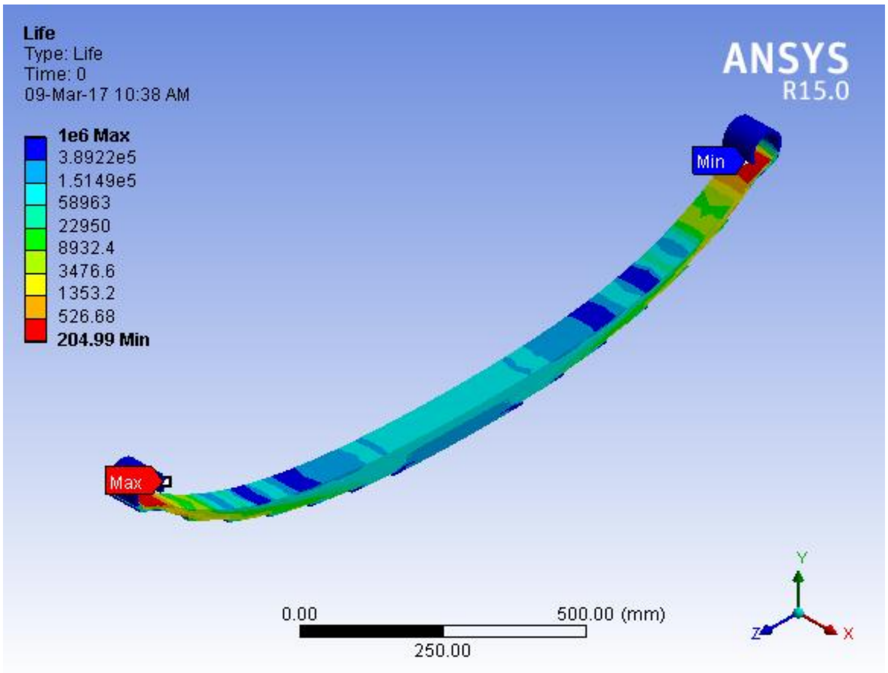


Figure 4.5: (b) Life generated for Steel

E-glass/Epoxy is the optimized one due to its stress is minimum and its life is more rather than the other one. So, it can be replaced by the original one.

Safety Factor

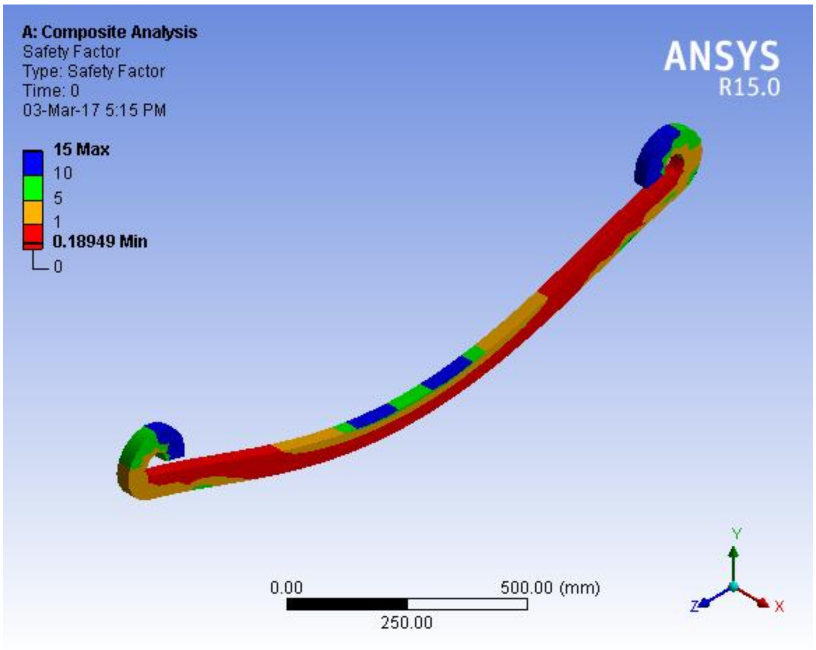


Figure 4.6: (a) Safety Factor for E-glass/Epoxy

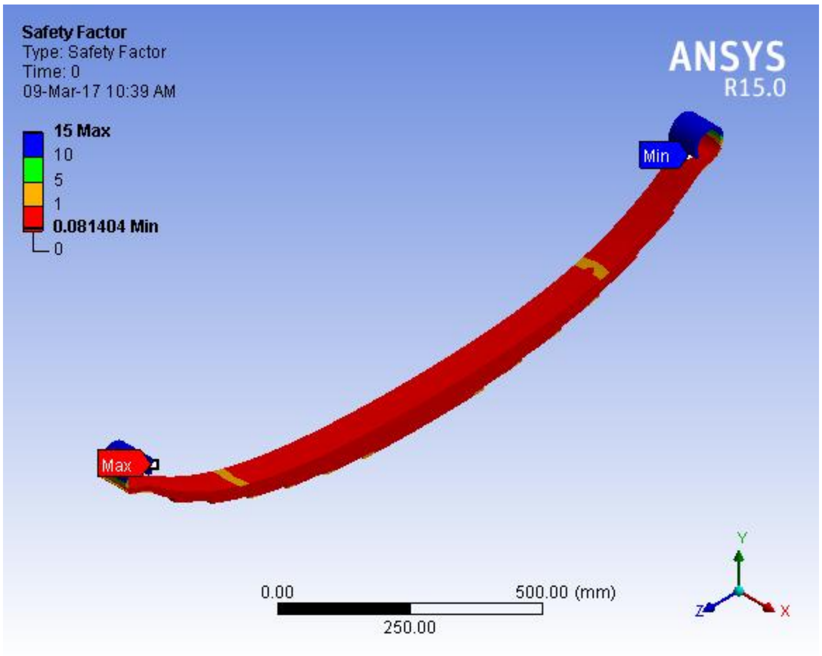


Figure 4.6: (b) Safety Factor for Existing Steel

Damage

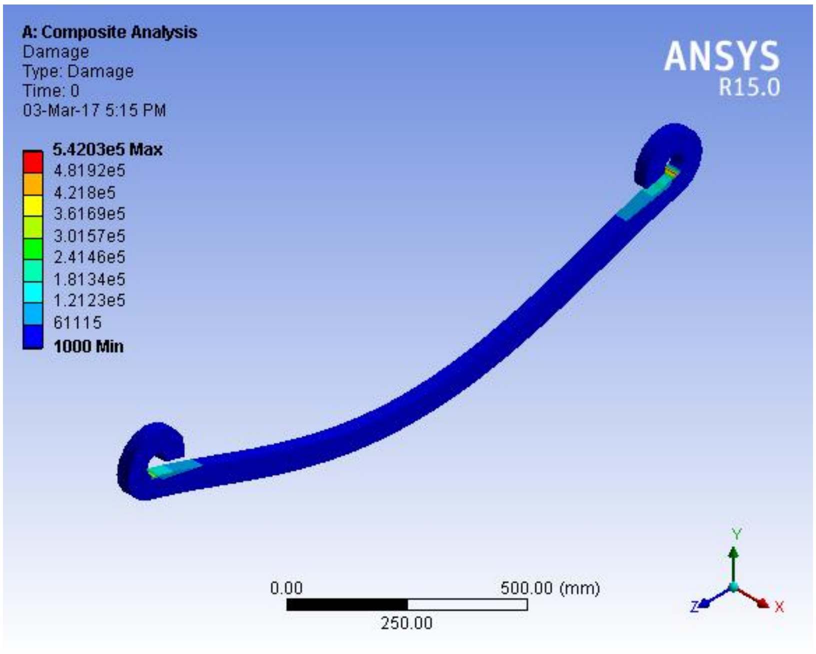


Figure 4.7: (a) Damage for E-glass/Epoxy

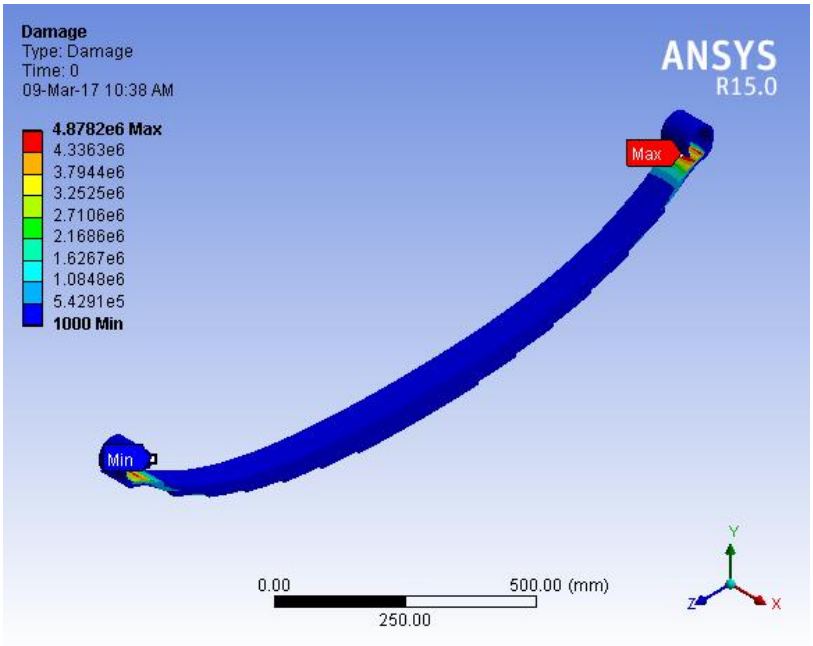


Figure 4.7: (b) Damage for Existing Steel

The damage in case of E-glass/Epoxy is much less than that of the conventional one. So, it will be safer than the original steel leaf spring.

4.2. Discussions

This paper focuses on the design analysis of composite leaf spring for improving fatigue life and weight optimization of the component. Leaf spring model in ANSYS WORKBENCH predicts that the maximum value of the equivalent alternating stress decreases, and fatigue life increases.

Table 5: Alternating stress versus Cycles

Alternating Stress (MPa)	Cycles
3999	10
2827	20
1896	50
1413	100
1069	200
441	2000
262	10000
214	20000
138	1.e+005
114	2.e+005
86.2	1.e+006

Fatigue Sensitivity

Figure 4.10 shows fatigue sensitivity curve and how the fatigue results change as a function of the loading at the critical location on the model.

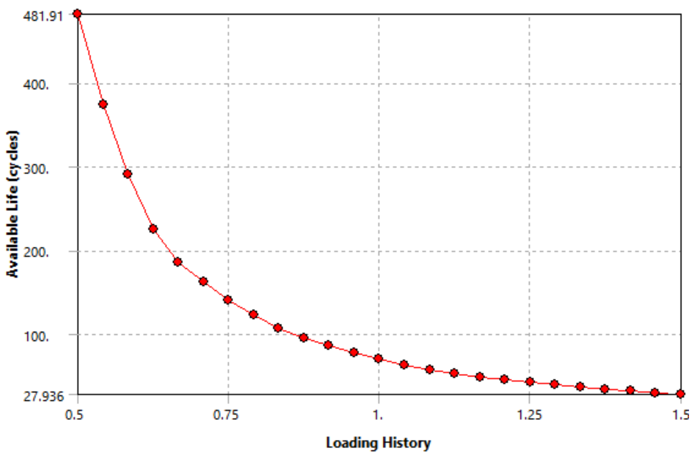


Figure 4.8: Fatigue Sensitivity Curve

Figure shows fatigue sensitivity curve and between 1.25 and 1.5 is dangerous region as load going to increase material get failure at critical location.

As presented above, we discussed the modelling and analysis of the conventional steel and E-glass/Epoxy composite leaf springs with the same loading and boundary conditions. The results of the analyses are shown in the previous chapter. The results are tabulated in the Table 6.

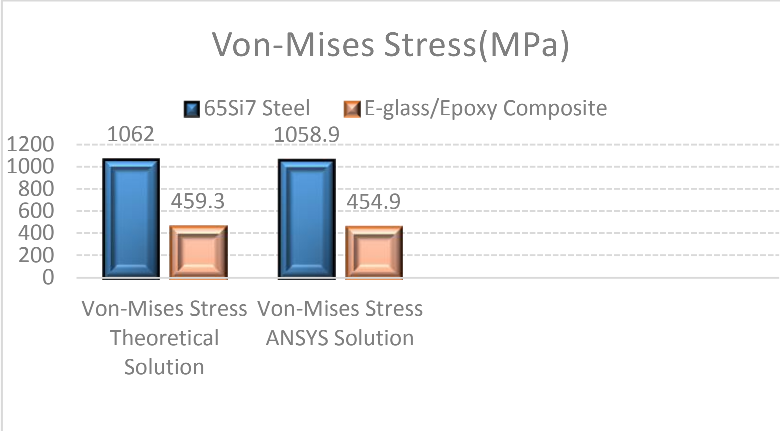
Table 6: Comparison Between Steel and E-glass/Epoxy Composite Leaf Springs

Parameters	**65Si7 Steel Leaf Spring**		**E-glass/Epoxy Composite Leaf Spring**		**Reduction of E-glass/Epoxy Composite Leaf Spring**
	Theoretical solution	**ANSYS solution**	**Theoretical solution**	**ANSYS solution**	
Von-Mises Stress (MPa)	1062	1058.9	459.3	454.9	57 %
Maximum Deflection (mm)	506	3.9629	67.2	3.1216	87 %
Total Mass (Kg)	49		7.64		84.4 %

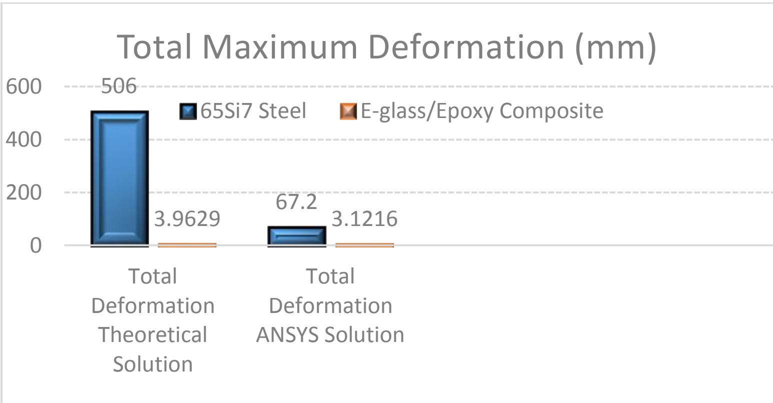
Table 7: Ansys Comparison of life between 65Si7 Steel and E-glass/Epoxy leaf spring

Parameter	**65Si7 Steel Leaf Spring**	**E-glass/Epoxy Composite Leaf Spring**	**Increment of life on E-glass/Epoxy Composite Leaf Spring**
Minimum Life by Ansys	204.99	1844.9	89%

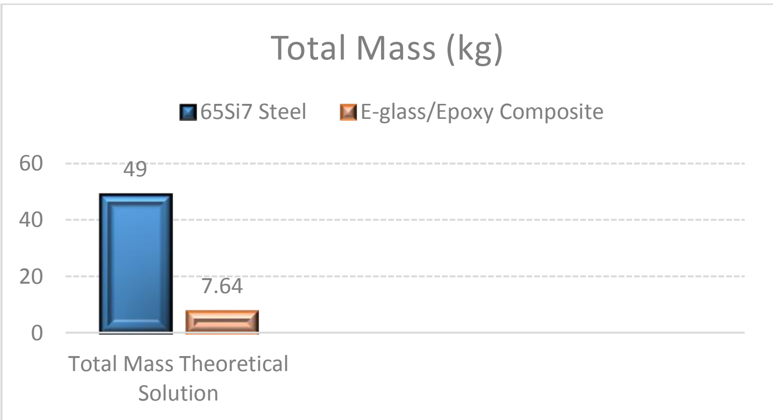
Through the comparative assessment of 65Si7 steel and E-glass/Epoxy composite material leaf spring the maximum total deflection is reduced by 87% through E-glass/Epoxy composite material, Von-Mises stress generation is reduced by 57%, and the weight is also reduced by 84.4% using the E-glass/Epoxy composite material as shown in figure 4.9. And life of E-glass/Epoxy composite leaf spring is increased by 89%.



(a)



(b)



(c)

Figure 4.9: Comparison of two materials on basis of (a) Von-Mises stress, (b) Total maximum deformation, and (c) Total mass

CHAPTER FIVE

5. CONCLUSION, RECOMMENDATION, AND FUTURE WORK

5.1. Conclusion

Reducing weight and increasing strength of products are high research demands in the world; composite materials are getting to be up to the mark of satisfying these demands. In this paper, a mono composite leaf spring for the vehicular suspension system was designed using E-glass/Epoxy composite leaf spring with the objective of minimizing weight of the leaf spring and improving fatigue life. And it is shown that the resulting design stress which is $459.3\ MPa$ is much below the strength properties of the material which is $473\ MPa$ satisfying the maximum stress failure criterion. So, the design is safe. And the leaf spring model was created by SOLIDWORK software. Then, the model created by SOLIDWORK was imported to ANSYS software. The analysis of the leaf spring will be done using material optimization like eye end segment regions. Finite element analysis using ANSYS and analytical methods are done. The eye two ends segment is maximum stress area in the leaf spring. The FE model of the leaf spring geometry is meshed with tetrahedral elements. Mesh refinement are done on the eye end segment regions, so that fine mesh is obtained on eye areas, which are generally critical locations on leaf spring. The failure in the leaf spring initiated at the eye end segment region of the spring, and fatigue is the dominant mechanism of failure. The comparison results of all different parameters will show the effect of stresses on leaf spring and this will help to select optimized one. Geometry or material optimization resulted in 57% stress reduction, total deformation is optimized by 87%, and weight of leaf spring is reduced by 84.4% which was achieved by changing material. The deflection of the leaf spring along its transverse direction is 67.2 mm, which is very small compared to the considered maximum deflection $\delta_{max}(105\text{ mm})$. As the stress of the leaf spring is decreased this will increase fatigue life of the leaf spring. Analytical stress and FEA results showed close agreement and fatigue life of modified leaf spring is improved rather than original and the optimized one can be replaced the original one.

5.2. Recommendation

From the result found from analytical and ANSYS result, the best way of analyzing fatigue life is in ANSYS result. In this paper, better fatigue life of leaf spring is done due to material optimization. We can see that as the stress decreases the life of the leaf spring increase. In this manner, there are some recommendations given for the leaf spring. Predicting and preventive maintenance will help to decrease the crack initiation and crack propagation and minimize the cost of maintenance of leaf spring change due to damage. Additionally, by using better strength material property the fatigue life of the leaf spring will be improved.

5.3. Future work

By considering the aspects presented in this study more work can also be done in the same area. The considerations for future work are as follows: -

- ✓ In this paper, only analysis of leaf spring was performed, so one can go for the production of composite leaf spring.
- ✓ As this analysis is under static load condition, so one can go for the analysis of dynamic.
- ✓ In Future parametric optimization, can be done by changing other parameters (varying width and varying thickness).
- ✓ According to FE analysis, blue locations shown in Figure have low stresses during service life and have the potential for material removal and weight reduction.

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