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Design of Fatigue Testing Machine for Composite Leaf Spring

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I. INTRODUCTION

The main purpose of this project is to understand the fundamental knowledge of design and mechanism by using manufacturing system and a simple mechanism property. The project also aims to evolve and improve performance, leaving no doubt in its design and concept. This project requires a lot of skills, information and knowledge. B. Use of Computer Aid Design software, Solid Works software, shearing machines, bending machines, and machines called vertical bend saw, bench work and welding process. This design definitely would help the user. This design therefore goes through many processes before moving to the prototype stage to achieve the goals and, of course, the customer's requirements.

Fatigue testers are used to measure the fatigue life and fatigue strength of materials. A look at most scrap pieces shows that almost all failures occur at stresses below the yield strength of the material. This complex phenomenon is known as "fatigue". 90% of industry failures are due to fatigue. Fatigue failure was shrouded in mystery in the 19th century as it did not exhibit visible plastic deformation, and fatigue was considered an engineering problem. Major breakthroughs in understanding the fatigue failure process came in the 20th century with the help of more powerful tools such as computers and powerful microscopy instruments. Fatigue was then viewed as a material and structural phenomenon rather than as a technical problem.

The idea behind this work is not to give answers to unanswered questions, but to solve the difficulty of answering them from another perspective. It includes defining the unpredictability of fatigue failure by describing known fatigue testing techniques. In materials science, fatigue is the weakening of a material caused by repeated loading. This is progressive, unpredictable structural damage that occurs when materials are subjected to cyclic loading.

The nominal maximum stress value that actually causes a material to fail can be much lower than the material's strength, which is usually reported as yield strength or ultimate strength.

When the applied load exceeds a certain threshold, micro cracks begin to form in stress concentration areas such as: B. where grain boundaries or surface defects exist. Eventually the crack reaches a critical size, the crack propagates faster and the structure fails.

II. LITERATURE REVIEW

1) A Review on Design and Fabrication of Fatigue Testing Machine, Shashidhar MBanavasi, Ravishankar K S,Padmayya S Naik, 2018
2) Design and Analysis of Leaf spring using various composites, K Ashwini CVM Rao 2018
3) Modeling and Analysis of leaf spring with different type of materials, B Mailesh, B Gupta, SK Kumar, and SP Jani 2021
4) Yahya Kara, worked on Fibre Reinforced Polymer Composite Helical Springs.

The FRP composite material allows the weight of the coil spring to be reduced without reducing the load capacity. With increased competition and technological innovation over the past decades, the automotive industry has shown interest in replacing traditional steel springs with his FRP composite coil springs.

III. RESEARCH GAP

Due to the dynamic effects occurring in the fatigue machine compared to other fatigue setups, a small scale study was conducted to modify the design with simpler assembly and lack of dynamic scattering calibration. Even fatigue testers available on the market are expensive and complex to design assemblies. Uncertainties associated with fatigue tester variance have created the need to perform statistical comparisons of machines in special cases where multiple fatigue testers are used in the same study. Laboratory experiments are an important last link to fully understanding and understanding scientific and engineering theories. Every effort should be made not to deprive future engineers and educators of this important part of their education. Therefore, there is a continuing need to develop effective and efficient teaching methods and techniques for engineering laboratory experience. The idea behind this paper is not to give answers to unsolved problems, but to approach them from another perspective. So the plan here was to design and manufacture a test at a lower cost that would help us better understand the problem.

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IV. OBJECTIVES

The main objective of this project work is to design and manufacture a fatigue tester capable of testing the fatigue life of various samples of composite leaf springs manufactured for automotive applications. In addition, the results of each test conducted with this machine can maintain the fatigue life of various fiber-reinforced composite materials and optimally prevent fatigue fracture.

A. Project Scope

The scope of this project gives us many advantages to learn new processes for making this product. Scope of this project. The fatigue tester is the most important part of the project.

The scope of work for this project are:

1) Design literature survey from all possible sources.
2) Design the model.
3) Create a design in the selected material.
4) Test your design with a demonstration.
5) Designing mechanical parts using Solid Work CAD software.
6) Development of a hassle-free fatigue tester.
7) Develop fatigue tester models using bending, welding, drilling, and cutting methods.

Most of today's vehicles, such as light vehicles, heavy trucks, and rail systems, use leaf springs to absorb vehicle impact loads. Withstands lateral loads, braking torques and drive torques in addition to shock absorption. In recent years, automobile manufacturers have been working to reduce the weight of vehicles in order to meet the need for resource conservation. Weight reduction can be achieved by implementing better materials, design optimizations, and manufacturing processes. Suspension leaf springs are one of the potential factors in reducing the unsprung weight of automobiles. The vehicle accomplishes this with more fuel efficiency and improved driving characteristics The introduction of composite materials has the potential to reduce the weight of leaf springs without reducing their load carrying capacity. Composite materials have been developed to reduce the weight of mechanical elements without reducing their load carrying capacity. FRP springs have excellent fatigue life and durability. FRP springs too have excellent fatigue life and durability. Glass fibres are strong as any of the newer inorganic fibres but they lack rigidity of on account of their molecular structure. Leaf spring weight reduction is achieved through material substitution and design optimization. In the current scenario, automakers are primarily focused on weight reduction. The emphasis is on reducing weight without sacrificing mechanical strength to achieve better performance. Replacing steel with composite leaf springs can reduce weight by 75% to 78%. In addition, composite leaf springs have lower stresses compared to steel springs.

V. THEORY

Examination of almost all scrap failed parts reveals that many failures occur at stresses below the yield strength of the part material. This complex phenomenon is known as "fatigue". Fatigue is responsible for up to 90% of component failures in the industry.

In the 19th century, it was considered cryptic that fatigue fractures did not exhibit visible plastic deformation, leading to the misconception that fatigue was merely an engineering problem.

Microscopic devices at the time had very limited capabilities, and few fatigue tests by reputable researchers have been done in this century, so they weren't so wrong. The most popular was the work of August Wöhler, who later came up with the idea of the load-life curve (Wöhler curve).

A major breakthrough in understanding the fatigue failure process occurred in the 20th century. Thanks to more powerful tools such as computers, powerful microscopy equipment, advanced numerical techniques, and much more research (John Mann cites 100,000 references in one of his books), fatigue is It was no longer seen as a technical issue, but as materials and design. phenomenon.

Despite extensive research on fatigue failure, its true nature is still unknown, and cyclic loading damage, cracks, and even complete failure are constantly being reported. If the problem persists after his 100 years of research in the last century, there is something to explain.

The idea behind this paper is not to give answers to unsolved problems, but to approach them from another perspective. This includes a description of the intricacies of fatigue failure, an introduction to the basic concepts of design against fatigue failure, a description of known techniques for fatigue testing, and validation testing of a servo hydraulic dynamic testing machine still under construction at the Mechatronics Laboratory settings in HAMK University of Applied Sciences and planning a laboratory fatigue test exercise. It is suitable for teaching materials in mechanical design or materials science courses.

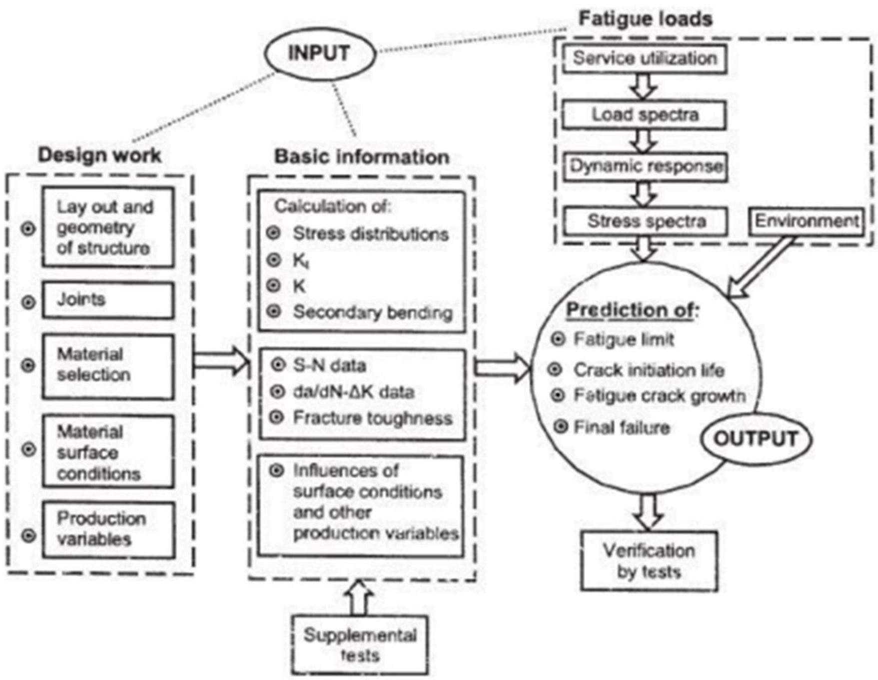
VI. FATIGUE AS A PHENOMENON IN THE MATERIAL

A. General

Fatigue is the condition in which a material cracks or fails due to repeated (repeated) application of stress below the ultimate strength of the material. Fatigue failure is often sudden and has devastating consequences.

When a structure is loaded, cracks form on a microscopic scale (crack nucleation) and continue to grow (crack growth) until the specimen completely fails. The entire process shapes the fatigue life of the component under consideration. According to Jaap Schijve, rational fatigue prediction for design or analysis is not only about fatigue as an engineering problem, but also as a materials phenomenon, a process that leads from invisible microscale cracks to macroscale fatigue failure.

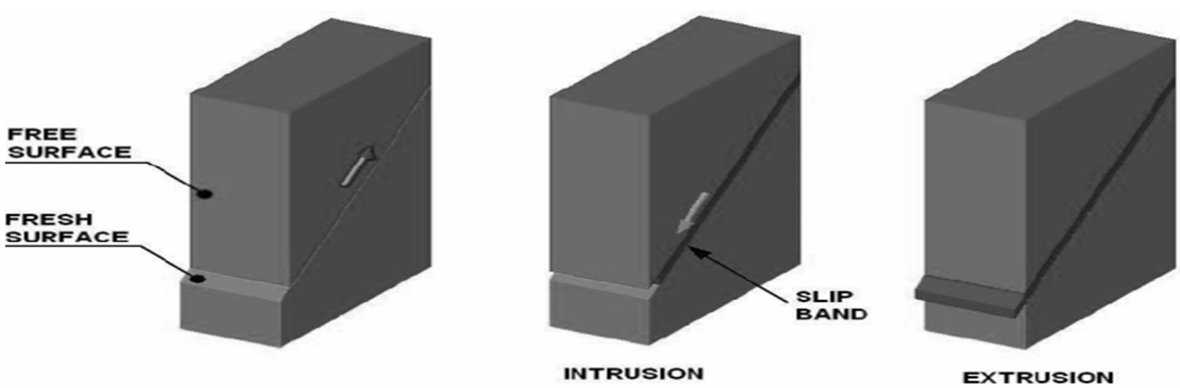
To this end, we discuss the stages that underlie the fatigue life of components, the critical fatigue properties of common materials, and the basic principles of design against fatigue failure.



Survey of the various aspects of fatigue of structures

B. Different Phases of Fatigue Life

Microscopy studies in the 20th century showed that fatigue crack nucleation occurs very early in fatigue life. Cracks start as slip bands within the grain. Cyclic slip is caused by cyclic shear stress. This slip leads to the formation of slip steps. In the presence of oxygen, the newly exposed surface of the material is oxidized in the slip step, preventing slip reversal. Slip reversal in this case occurs at adjacent slip planes, forming ridges and depressions on the surface of the material as shown in the figure below.



Formation of intrusion and extrusion marks on the material surface

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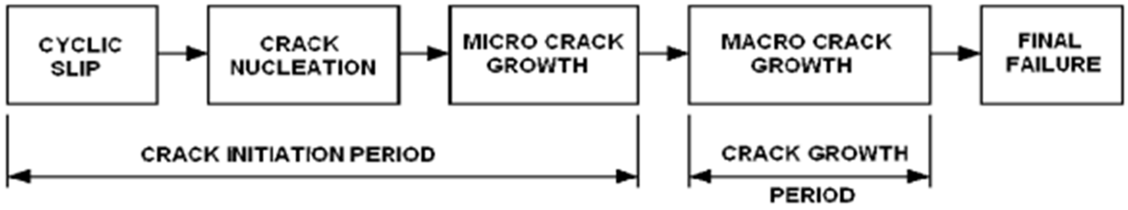
The fatigue life is generally divided into three stages/periods



Different phases of the fatigue life

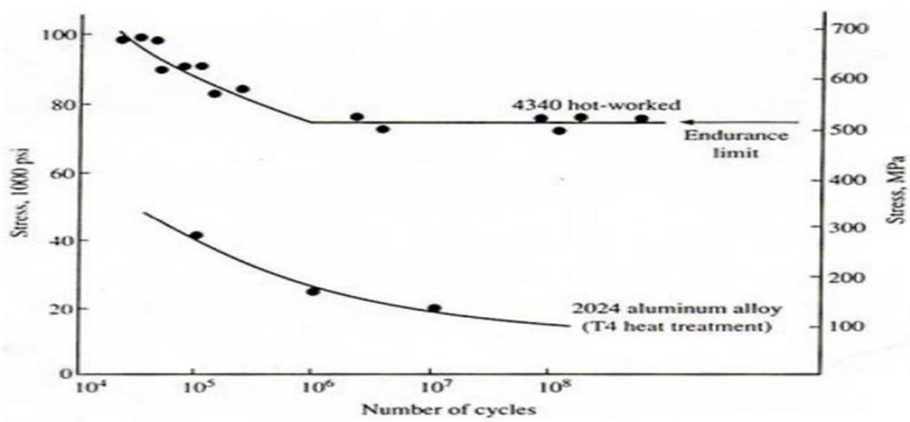
The fatigue life (N_f) of a component is defined by the total number of stress cycles required to cause failure. Fatigue life can be divided into three stages where:

$N_f = N_i + N_p$

1) *Crack Initiation (N_i):* This is the number of cycles required to initiate cracking. This is commonly caused by dislocation packing and defects such as surface roughness, voids, and scratches. During this period, fatigue is a material surface phenomenon. The stress concentration factor, K_t, is another factor that must be considered when predicting crack initiation.
2) *Crack Growth (N_p):* It is the number of cycles required to grow a crack steadily to a critical size and is generally controlled by the stress level. Prediction of crack growth is the most studied aspect of fatigue because most common materials contain defects. The resistance to crack growth when a crack penetrates a material is a material-dependent mass property. It is no longer a superficial phenomenon. The stress intensity factor is an important factor in predicting fatigue growth.
3) *Rapid Fracture:* Very rapid critical crack growth occurs when the crack length reaches a critical value. Rapid rupture occurs in a short period of time, so there is no period of rapid rupture during fatigue life. Material Fracture Toughness KIC is a key factor in rapid failure prediction or design against failure.

C. Fatigue Properties of Materials

Fatigue is generally understood to mean the gradual deterioration of materials subjected to cyclic loading. Fatigue testing subjects a specimen to a cyclically varying load of constant amplitude. The applied voltage alternates between equal positive and negative values from zero to a maximum positive or negative value, or between equal positive and negative values, or between unequal positive and negative values. A series of fatigue tests are performed on a large number of specimens of the material at various stress levels. The load endured is then plotted against the number of cycles endured. By choosing progressively lower stresses, one can find values that do not cause failure regardless of the number of cycles applied. This stress level is called the fatigue or endurance limit of the material. The two-term representation is called a voltage gradient diagram or Wöhler diagram. Fatigue limits can be specified between 2 and 10 million cycles for most steels. Non-ferrous metals such as aluminum generally do not have well-defined fatigue limits. (Mark's Standard-Handbook/Strength of Materials).



S-N Curve of a ferrous and non-ferrous metal

Surface imperfections such as roughness, scratches, nicks and ridges reduce the fatigue strength of the part. Different metals vary greatly in their susceptibility to roughness and concentration or notch sensitivity effects. For a particular material exposed to a particular stress condition and load type, notch sensitivity can be viewed as the ability of that material to resist stress concentrations associated with the presence of notches. Corrosion and galling (due to friction between mating surfaces) can cause a significant reduction in fatigue strength, sometimes reaching 90% of the original fatigue strength. Any corrosive agent will accelerate severe corrosion fatigue, but there are numerous differences in the effects of seawater or tap water from different locations.

This among others explains the complex nature of fatigue phenomenon. Shot-penning, nit riding and cold work usually improve fatigue properties.

There is generally not a very good correlation between fatigue properties and other mechanical properties of materials. The best correlation is between full reverse load fatigue limit and normal tensile strength. For many ferrous metals, the fatigue limit is approximately 0.40 to 0.60 times the tensile strength. For nonferrous metals, it is about 0.20 to 0.50 times the tensile strength.

Material	Tensile strength		Endurance limit		Endurance ratio
	(MPa)	(ksi)	(MPa)	(ksi)	
Ferrous alloys					
AISI 1010, normalized	364	52.8	186	27	0.46
1025, normalized	441	64	182	26.4	0.41
1035, normalized	539	78.2	238	34.5	0.44
1045, normalized	630	91.4	273	39.6	0.43
1060, normalized	735	106.6	315	45.7	0.43
1060, oil Q, tempered	1295	187.8	574	83.3	0.44
3325, oil Q, tempered	854	123.9	469	68	0.55
4340, oil Q, tempered	952	138.1	532	77.2	0.56
8640, oil Q, tempered	875	126.9	476	69	0.54
9314, oil Q, tempered	812	177.8	476	69	0.59
302, annealed	560	81.2	238	34.5	0.43
316, annealed	560	81.2	245	35.5	0.44
431, quenched, tempered	798	115.7	336	48.7	0.42
ASTM 20 gray cast iron	140	20.3	70	10.2	0.50
30 gray cast iron	210	30.5	102	14.8	0.49
60 gray cast iron	420	61	168	24.4	0.40
Nonferrous alloys					
AA 2011–T8	413	59.9	245	35.5	0.59
2024, annealed	189	27.4	91	13.2	0.48
6061–T6	315	45.7	98	14.2	0.31
6063–T6	245	35.5	70	10.2	0.29
7075–T6	581	84.3	161	23.4	0.28
214 As cast	175	25.4	49	7.1	0.28
380 Die-cast	336	48.7	140	20.3	0.42
Phosphor bronze, annealed	315	45.7	189	27.4	0.60
hard drawn	602	87.3	217	31.5	0.36
Aluminum bronze, quarter hard	581	84.3	206	29.9	0.35
Incoloy 901, at 650°C (1202°F)	980	142.1	364	52.8	0.37
Udimet 700, at 800°C (1472°F)	910	132	343	49.7	0.38
Reinforced plastics					
Polyester–30% glass	123	17.8	84	12.2	0.68
Nylon 66–40% glass	200	29	62.7	9.1	0.31
Polycarbonate–20% glass	107	15.5	34.5	5	0.32
40% glass	131	19	41.4	6	0.32

Fatigue strength and tensile strength of common materials

D. Design for Fatigue Failure

1) Corrected Fatigue Strength

The fatigue properties of materials can be said to be sensitive to many factors (size, surface area, test method, environment, probability). S-N curves obtained from laboratory tests must be related to actual design conditions by modifying some elements, at least the laboratory results cannot be directly used as it is.

Laboratory endurance strength (Se ’) of the materials obtain from S-N diagram (or the likes) are therefore corrected for actual conditions by using correction factors like;

Se = K_a x K_b x K_c x K_d x K_e x K_f x S_e

Where,
K_a = Surface Correction factor
K_b = Size Correction factor
K_c =Reliability Correction factor
K_d = Temperature Correction factor
K_e = Stress concentration Correction factor
K_f = Miscellaneous Correction factor
Se'= Endurance Strength of material specimen under laboratory condition
Se = Endurance Strength of material specimen under actual running condition

2) Selection of Materials for Fatigue Resistance

In many applications, component performance is affected by several factors other than the properties of the materials used in manufacturing. This is especially true when a component or structure is subject to fatigue loads, and fatigue strength can be greatly influenced by the operating environment, part surface condition, manufacturing process, and design details. In some cases, the role of the material in achieving satisfactory fatigue life is secondary to the above parameters, unless the material has gross defects. Below are the material types commonly used for fatigue design and their basic properties:

3) Steel and Cast Iron

1) Steel is a popular material of construction for fatigue applications because it offers high fatigue strength and good machinability at a relatively low cost.
2) The optimum steel structure for fatigue is tempered martensite which offers maximum homogeneity.
3) Highly hardenable steels exhibit high strength with relatively mild quenching, resulting in low residual stresses and are suitable for fatigue applications.
4) Compared to the coarse pearlite structure obtained by annealing, the normalized structure, which has a finer structure, is superior in fatigue resistance.

E. Properties Values

Density	7850 Kgm^3
Young's Modulus	2 * 10^{11} Pa
Poison's Ratio	0.3 Pa
Bulk Modulus	1.6667E+11 Pa
Shear Modulus	7.6923E+10 Pa
Tensile Yield Strength	2.5E+08 Pa
Compressive Yield Strength	2.5E+08 Pa
Tensile Ultimate Strength	4.6E+08 Pa

1) Nonferrous alloys

a) Unlike ferrous alloys, non-ferrous alloys, with the exception of titanium, generally do not have a defined fatigue life.
b) Aluminium alloys usually combine corrosion resistance, light weight, and reasonable fatigue resistance.
c) Alloys without fine-grained inclusions are best suited for fatigue applications.

2) Plastics

a) The viscoelasticity of plastics makes their fatigue behaviour more complex than that of metals.
b) The fatigue behaviour of plastic is affected by the type of loading, slight changes in temperature and environment, and the manufacturing process.
c) Due to the low thermal conductivity of, hysteresis heating can build up in the plastic, which can lead to failure of functionality due to thermal fatigue and loss of stiffness.
d) Heat generated increases with load and test frequency

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3) Composite Materials

a) The fatigue failure modes of reinforced materials are complex and can be affected by the manufacturing process when the difference in shrinkage between the fibers and the matrix induces internal stresses.
b) However from practical experiences, some fiber reinforced plastics are known to perform better in fatigue than some metal.
c) The advantage of fiber reinforced plastics is even more apparent when compared to aper weight basis.
d) Similar to static strength, fiber orientation affects fatigue strength of fiber reinforced composites.
e) For unidirectional composites, fatigue strength is significantly lower in directions other than fiber orientation.
f) Reinforcement with continuous unidirectional fibers is more effective than reinforcement with short random fibers.

VII. FATIGUE TESTING METHODS

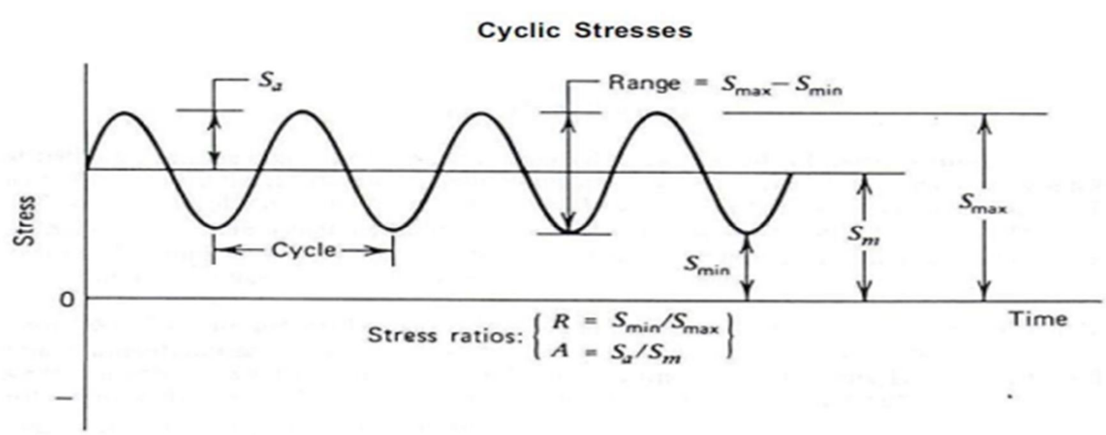
The purpose of fatigue testing is generally to determine fatigue life and/or danger points. H. Failure point of a specimen subjected to a specified series of stress amplitudes.
By simplifying and idealizing the experimental conditions, it is possible to vary one or more of the factors that affect fatigue life and determine their effects.
Even when these conditions are met, there are always many unknown and uncontrollable factors that lead to large variations in fatigue life, even for specimens that are considered identical.
There are two criteria for classifying different methods of fatigue testing.

1) The sequence of stress amplitude.
2) The nature of the test-piece.

A. Classification Based On The Sequence Of Stress Amplitudes

1) Constant-amplitude test

This is the simplest amplitude sequence obtained by applying a constant amplitude stress reversal to the specimen until failure occurs. Different specimens in a test series may experience different stress amplitudes, but the amplitudes do not change for individual elements.



Schematic Illustrating Cyclic Loading Parameters

The following parameters are utilized to identify fluctuating stress cycles:
Mean Stress, S_m
$S_m = (S_{max} + S_{min})/2$

Stress Range, S_r
$S_r = S_{max} - S_{min}$

Stress Amplitude, S_a
$S_a = (S_{max} - S_{min})/2$

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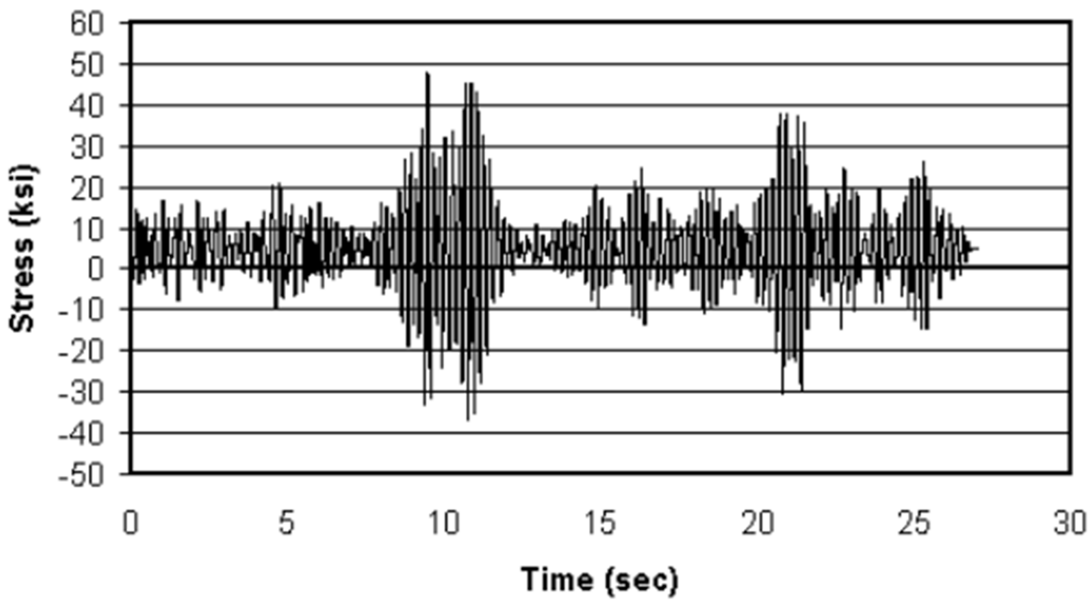
Stress Ratio, R
$R = S_{min} / S_{max}$

Depending on the choice of stress level, constant amplitude testing can be categorized into three types.

a) *Routine Test:* In a typical test, the applied voltage is chosen such that all samples are expected to fail after a reasonable number of cycles, say 10^4 to 10^7 cycles.
b) *Short-life Test:* Stress levels are appropriate above the yield point, and some specimens are expected to statically fail under load.
c) *Long-life Test:* For this type of testing, stress levels below or slightly above the fatigue limit are appropriate, and some samples do not fail after a given number of cycles, approximately 10^6-10^7 cycles.

2) Variable-amplitude tests

A more complex amplitude sequence is required to simulate the stresses the sample will experience in actual use. Realistic simulations are very complex. Sequences of stress amplitudes can be simplified to discover regularities associated with the accumulation of fatigue damage in samples subjected to stress reversals at different amplitudes. Regardless of the pattern used, such tests are called variable amplitude tests.



Variable amplitude fatigue stress loading

Variable amplitude testing can be further classified as:

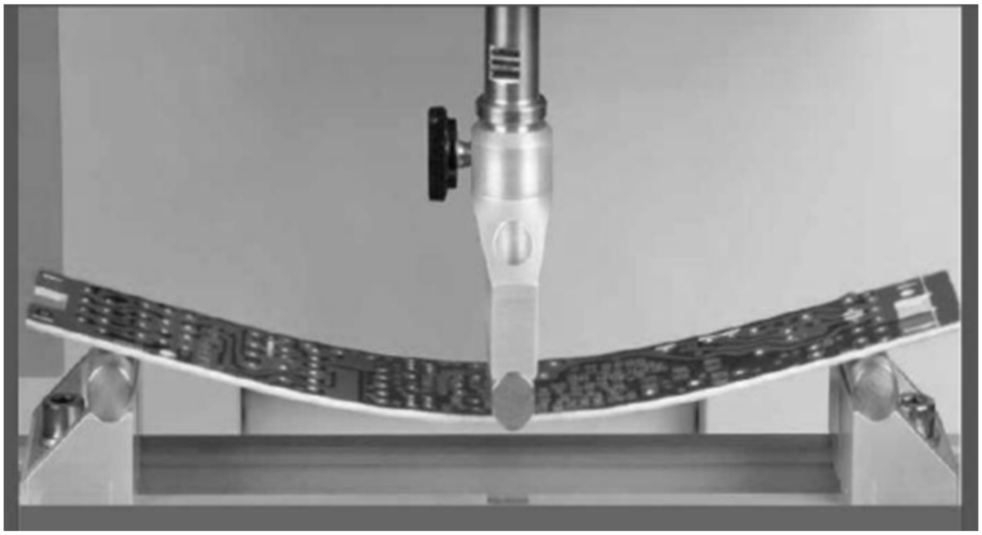
a) *Cumulative Damage Test:* These are tests aimed at examining theories of cumulative damage and often simplify procedures.
b) *Service Simulating Test:* These are tests that use more sophisticated patterns (closer to actual service loading) for simulation purposes.

B. Classification Based on the Nature of the Test-Piece

It is sufficient to classify candidates into two categories:

1) *Specimens:* The term "specimen" is commonly used to mean simple shaped specimens, often standardized, small in size, with good surface finish and carefully manufactured.

The purpose of simplification is not to lower prices, but to reduce product variability and control various influencing factors.
This type of specimen is commonly used to test materials and determine their fatigue behavior. It is also widely used for research purposes.



3-point bending test of a circuit board

C. General Classification of Fatigue Testing

Based on the three categories above, fatigue testing can be classified according to the purpose of the test. The three categories are:

1) *Material Type Test:* Material type testing includes a variety of simple tests such as different material behavior when subjected to cyclic loads, the effects of different manufacturing processes, material behavior in different environments, different sizes and shapes, notches, different surface finishes, etc. Useful for comparing geometric factors. It can also be used to investigate the effects of surface treatments such as hardening, decarburization, nitriding, shot peening, and plating on the fatigue properties of various materials.
2) *Structural Type Test:* This type of testing helps compare components made from different materials, designs, and structures manufactured by different processes. It can also be used to reveal stress concentrations and develop better designs and manufacturing processes.
3) *Actual Service Type Test:* These are tests typically used as reliability or quality tests and are primarily intended for finding or verifying faults in new components within a machine or structure.

Fatigue tests are completely different in nature from the above tests, and are intended to investigate the initiation and propagation of fatigue cracks, requiring complex knowledge of fracture mechanics and microscopic observation of material structures. , is beyond the scope of this document.

VIII. FATIGUE TESTING MACHINE

Fatigue testers can be classified according to various aspects such as purpose of the test, type of stressing, means of producing the load, operation characteristics, type of load etc.

The purpose of the investigation is the most important point of the investigation and is of course known at the beginning of the investigation. Therefore, it makes sense to categorize them based on their test purpose.

A. Classification of Fatigue Testing Machine

Fatigue testing machines can be classified as follows according to the purpose of the test.

1) General purpose fatigue testing machine
2) Special purpose fatigue testing machine
3) Equipment for testing parts and assemblies

In this work, the General class is further expanded, especially since it is a type that is widely used in academic settings.

B. General Purpose Testing Machine

In this work, the General class is further expanded, especially since it is a type that is widely used in academic settings.

They are following:

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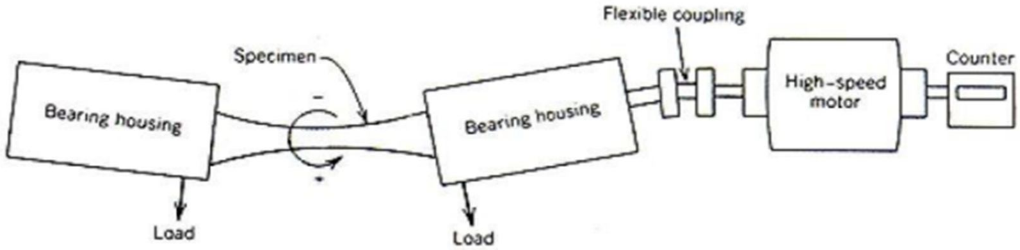
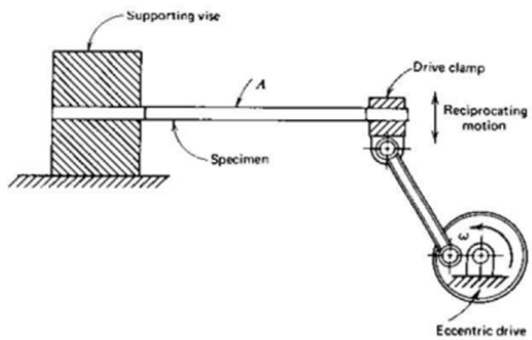
1) *Classification Based on Type of Stressing Method*

a) *Rotating Bending Testing Machine:* The type of SN curve produced by this machine is identified as a torsional bending, stress controlled fatigue data curve. A rotary flex tester is used to generate an S-N curve by rotating the motor at a constant rpm or frequency. To induce failure in a specimen, a constant steady-state force is applied to the specimen, creating a constant bending moment. When a rest moment is applied to a rotating sample, the stress at any point on the outer surface of the sample goes from zero to maximum tensile stress, back to zero, and finally compressive stress. Therefore, the state of stress is essentially a state of total reversal.



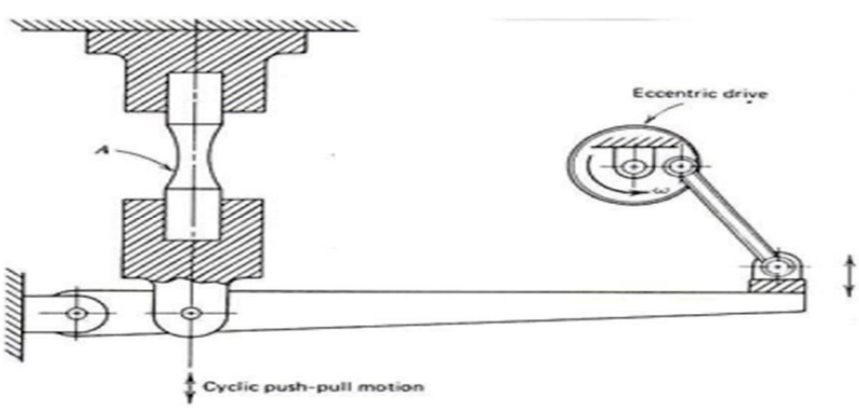
Rotating Bending Testing Machine [Callister, 1994].

b) *Reciprocating Bending Test Machine:* The type of SN curve generated is identified as a stress-compressive, strain-controlled fatigue data curve. This type of machine can be loaded by placing the specimen vise against the mid-displacement position of the crank mechanism.



Reciprocating Bending Testing Machine [Collins, 1981].

c) *Axial Loading (push-pull) Type Fatigue Tester:* In this type of test, the specimen is subjected to a purely axial (tensile or compressive) load rather than bending. Samples are held at both ends and loaded cyclically between the two extreme values (maximum and minimum).



Direct-Force Fatigue Testing Machine [Collins, 1981].

Axial load (push-pull) testers using conversion attachments can also perform bending and torsional fatigue tests if required. Most general purpose fatigue testers on the market have this capability.

Universal-testing Machine

Other possible types, although not commonly used for simplified testing, are:

- Torsion Load Fatigue Tester.
- Combined bending and torsional fatigue tester.
- 3 Fatigue tester with biaxial and triaxial loading.

2) Classification Based on Source of Stressing

The following classifications of fatigue testers are based on the principle of the source of the test force. Generated load:

a) Mechanical deflection
b) Dead weight or constant spring force
c) Centrifugal force
d) Electromagnetic force
e) Hydraulic force
f) Pneumatic force

The choice of stress source depends on many factors such as: B. Required frequency, amount of force required, available control systems, cost, and how much to simplify the test to the actual workload on the farm.

C. Design of the Testing Support Structure

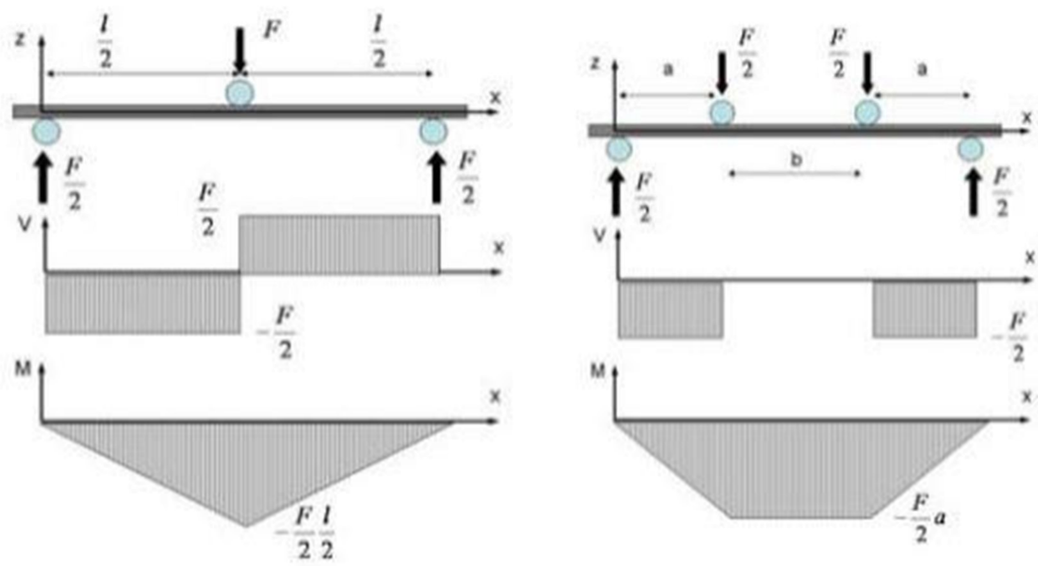
The original idea was to design a tensile test setup as a validation test for a dynamic fatigue tester. The main reason was that the machine was destined to test the behavior of welded TRIP steel from Innosteel Oy, Hämeenlinna when the welded part was exposed. Up to dynamic load tensile forces of 2 to 3 KN. To develop this design, several commercially available universal fatigue testers and international standards for basic mechanical test designs were analyzed. It was later found that the machine was unable to perform tensile fatigue tests because the dynamic testing machine's hydraulic cylinders could not provide the linear motion amplitude required for tensile fatigue testing, necessitating a new design (smaller linear amplitude is required).

Next, the bending setup was considered, aside from the fact that the first documented laboratory fatigue test by August Wöhler was actually a rotating bending fatigue test. A bent setup offers many advantages over other possible designs. Some of these benefits are listed below.

1) A simple way to characterize some of the mechanical properties of a material
2) No special grips required
3) Assembly and disassembly of specimens is very simple
4) Samples are usually very simple shape (rectangular cross). -section)

There are two options for flex life.

a) 3-point flex life.
b) Four point fatigue bend setup.



Transverse force and moment for the two bending setups

In a 3-point bending structure, each section of the beam has a shear force that creates interlaminar shear stresses and can lead to delamination. However, in 4-point bending, there is a constant cross-section of the beam and zero shear forces (that is, interlaminar shear stresses). Only normal loading occurs in this section.
In fatigue tests, specimens exhibit permanent deflection after thousands of cycles.
As a result, if the deflection of the indenter is less than the permanent deflection, the indenter will lose contact. In the next cycle, the indenter hits the surface of the specimen, causing impact damage, which distorts the fatigue data.
This problem can be solved if the permanent deflection is kept symmetrical, i.e. the deflection is zero. This is easily achieved by a full reverse bend with a displacement varying between $-U_{min}$ and $+U_{max}$.
Additionally, the sample rotates at both ends, so the outer supports should allow for this rotation. Failure to do so will introduce unwanted reaction forces on the sample and falsify the fatigue data.
A 4-point setup is required to perform a 4-point full inversion fatigue flex test. The figure below is the first proposed design for bending the support structure, considering all the above points to obtain reasonable fatigue data.

D. Types Of Spring

1) Helical springs
2) Conical and volute springs
3) Torsion springs
4) Disc or Belleville springs
5) Special purpose springs
6) Laminated or leaf springs

E. Dimensions Of Leaf Spring

Conventional design methods of leaf springs are largely based on the application of empirical and semi-empirical rules along with the use of available information in the existing literature. A spring's function is to absorb energy and release it depending on the function it needs to perform. Leaf spring design therefore depends on load capacity and deflection. Therefore, the Mahindra "Model Commander 650di" was designed with consideration given to the leaf springs.

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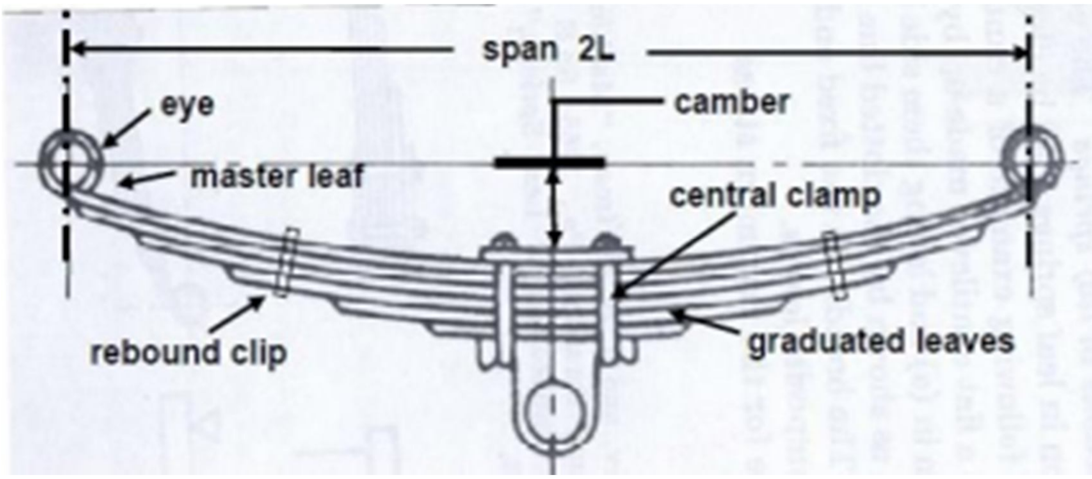
Volume 10 Issue VIII Aug 2022- Available at www.ijraset.com

F. Material Of Leaf Spring

1) E-glass 21xK43 Gevetex
2) AS4
3) Silenka E-glass 200tex
4) T300



Terminology of Leaf Spring

Properties of Composite Material:

Sr. no.	Properties	AS4	T300	E-glass 21xK43 Gevetex	Silenka E-glass 1200tex
1.	Ex	2.25E+11	1.38E+11	5.34E+10	4.56E+10
2.	Ey	1.5E+10	1.1E+10	1.77E+10	1.62E+10
3.	Ez	1.5E+10	1.1E+10	1.77E+10	1.62E+10
4.	PRxy	0.2	0.28	0.278	0.287
5.	PRyz	0.071	0.4	0.4	0.4
6.	PRzx	0.2	0.28	0.278	0.287
7.	Gxy	1.5E+10	5.5E+9	5.83E+9	5.83+9
8.	Gyz	7E+9	1.96E+9	6.32E+9	6032E+9
9.	Gzx	1.5E+10	5.5E+9	5.83E+9	5.83+9
10.	ρ	1790	1770	2550	2570

IX. METHODOLOGY

The fatigue tester consists of the following components:

- HP Electric Motor 2
- Flange, connecting rod.
- Dead weight
- Pillow block bearing
- Digital counter
- Speed controller
- Sample (leaf spring)
- Bearing flange and connecting rod

A. Flange and Connecting Rods
They are used to perform rotary motion and convert it to vertical linear motion. Therefore, you can achieve the desired stroke. For this purpose, an eccentric hole is made in the flange and a connecting rod is attached using a number of manufacturing processes (welding, threading, drilling). Eccentric length and connecting rod length affect stroke length. Therefore, it is important to pay close attention to these design parameters.

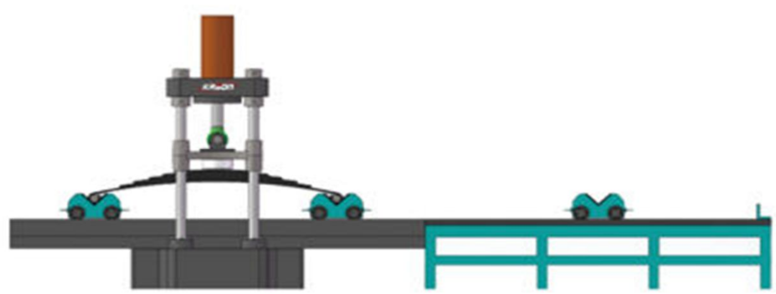
B. Bearing
To allow a connecting rod to move freely without interruption while realizing a leaf spring stroke.

C. Plummer Block
It balances rotating mass and high weight capacity by absorbing the vibrations induced during the lifting process.
The special bearing used in this project is a single row deep groove bearing that can withstand both radial and some axial loads. The bearings are mounted in cast iron housings to ensure proper bearing support and withstand maximum axial loads.

D. Digital Counter
These are important for ease of analysis. One is used to indicate the number of cycles completed and the other to indicate the stress generated at each cycle. These measurements make the machine work very accurately.

E. Variable Speed Control
The speed of the rotor is set by the arrangement of belts and pulleys. If necessary, it can be adjusted by rearranging belts and pulleys. However, the speed of the motor is predefined by the diameter of the pulley. Permission to change the speed is now complete.

X. WORKING
Power to the machine is cut off and the spring load is reduced to zero. Then unscrew the clamping mechanism and load the sample. Then tighten the bolts to prevent the sample from loosening during machine operation. The required load is set to the spring load by adjusting the nut under the spring load mechanism and the power is turned on. The digital counter resets to zero and the machine powers on. After a period of time, the sample fails, the digital counter stops, and the machine powers off. The failed sample is then removed, a new sample loaded and the test repeated for different values of spring load.



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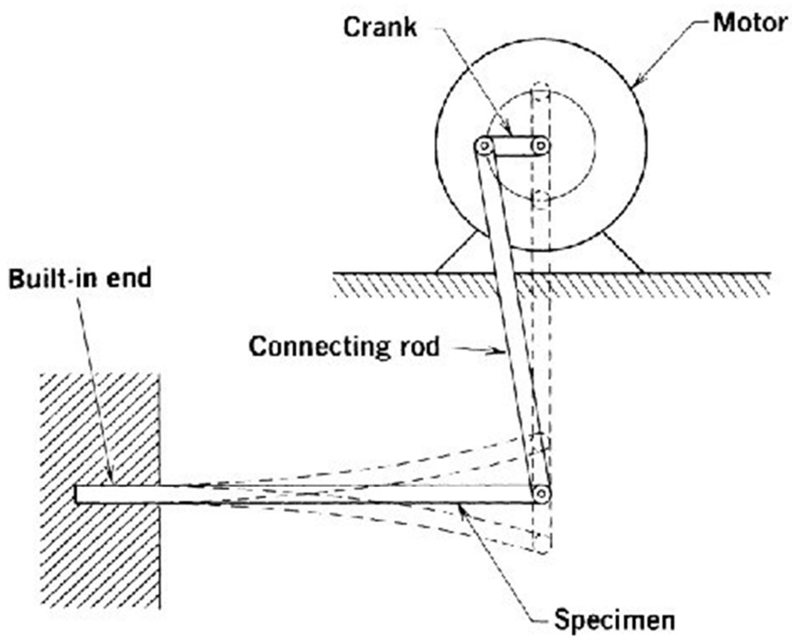


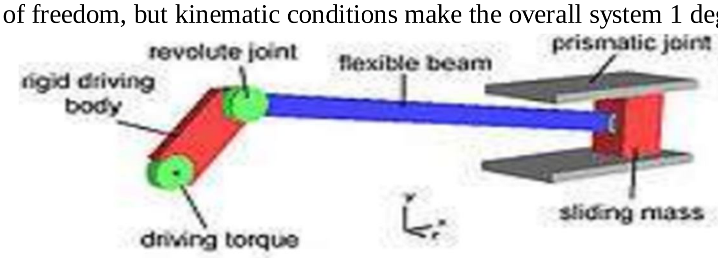
Fig 9.13 a reciprocating cantilever bending fatigue testing machine based on controlled deflections from a rotating eccentric.

Mechanisms for converting reciprocating motion to rotating motion or rotating motion to reciprocating motion have been used for many centuries. For example, from the third century AD Hierapolis Sawmill, in which a water wheel was used to power a horizontal reciprocating saw, to the mechanisms that form the modern internal combustion engine.

The most commonly used mechanism for converting between rotational and reciprocating motion is the crank and slider mechanism. Other known mechanisms include the scotch yoke, and others.

Slider-crank Mechanism is an arrangement of mechanical parts designed to convert straight line motion to rotary motion, as in a reciprocating piston engine, or it is used to convert rotary motion to straight line motion as in a reciprocating pump. Internal combustion engines are a common example of this mechanism, where combustion in a cylinder creates pressure which drives a piston. The linear motion of the piston is converted into rotational motion at the crank through a mutual link, referred to as the connecting rod. As the geometry of the crank forces the conversion of linear motion to rotational, shaking forces are generated and applied to the crank's housing.

The shaking forces result in vibrations which impede the operation of the engine. A slider-crank mechanism is used to convert rotary motion into translational motion through a rotary drive beam, connecting rod, and sliding body. In the present example, a flexible body is used for the connecting rod. The sliding mass must not rotate and uses swivel joints to connect the bodies. All bodies in space have 6 degrees of freedom, but kinematic conditions make the overall system 1 degree of freedom.



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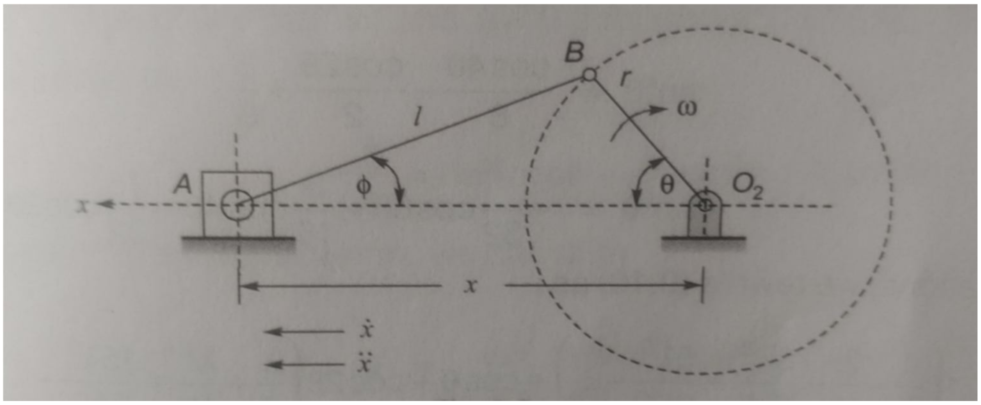
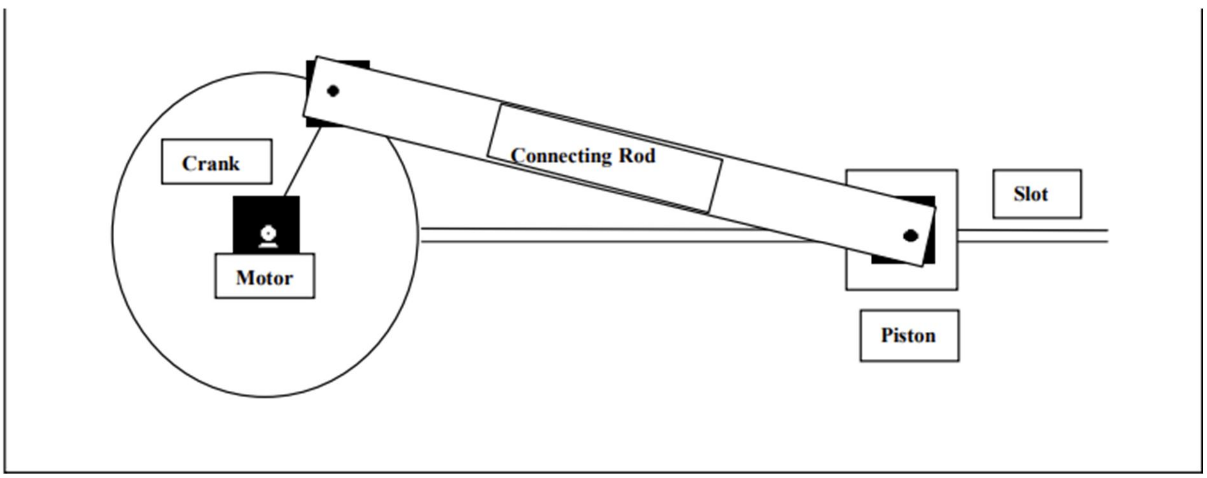
XI. NUMERICAL ANALYSIS



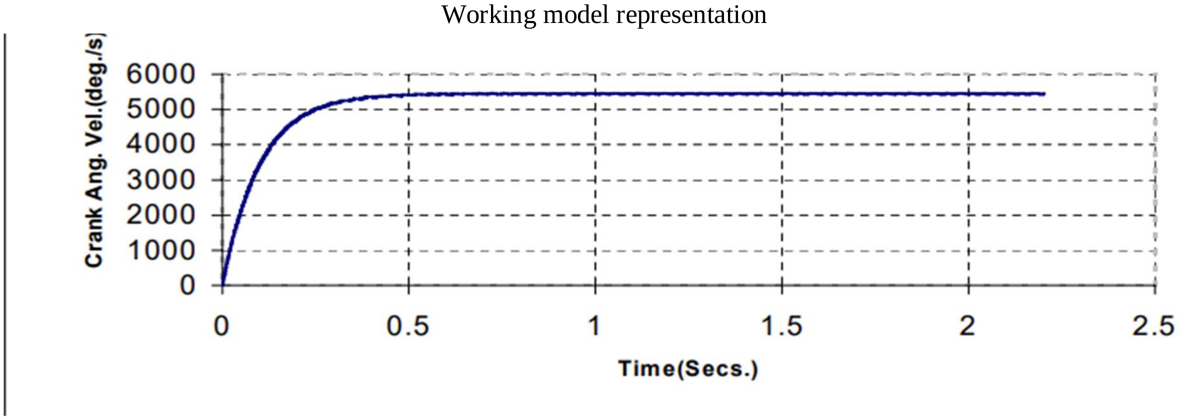
A. Material Selection

Connecting rod- medium carbon steel (AISI 1040)
Crank- medium carbon or alloy steel (AISI 1040)
Plunger- Steel S355-J2G3
Bearing-

B. Force Analysis of Machine Elements



Working model representation



Crank angle velocity vs time

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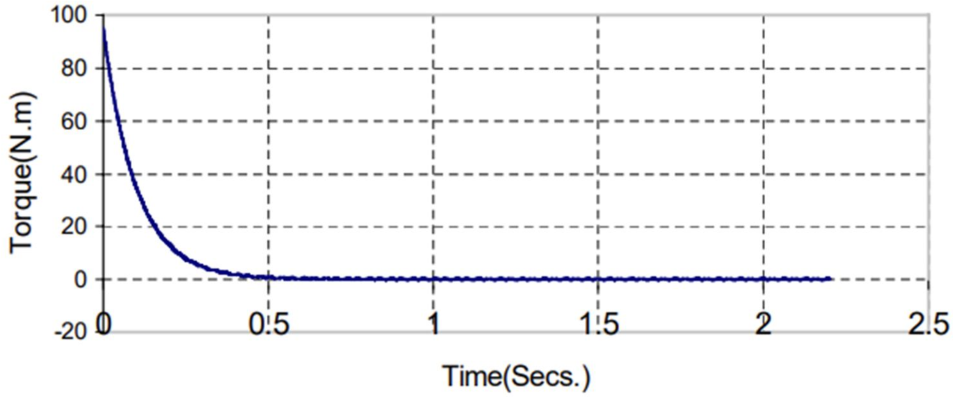
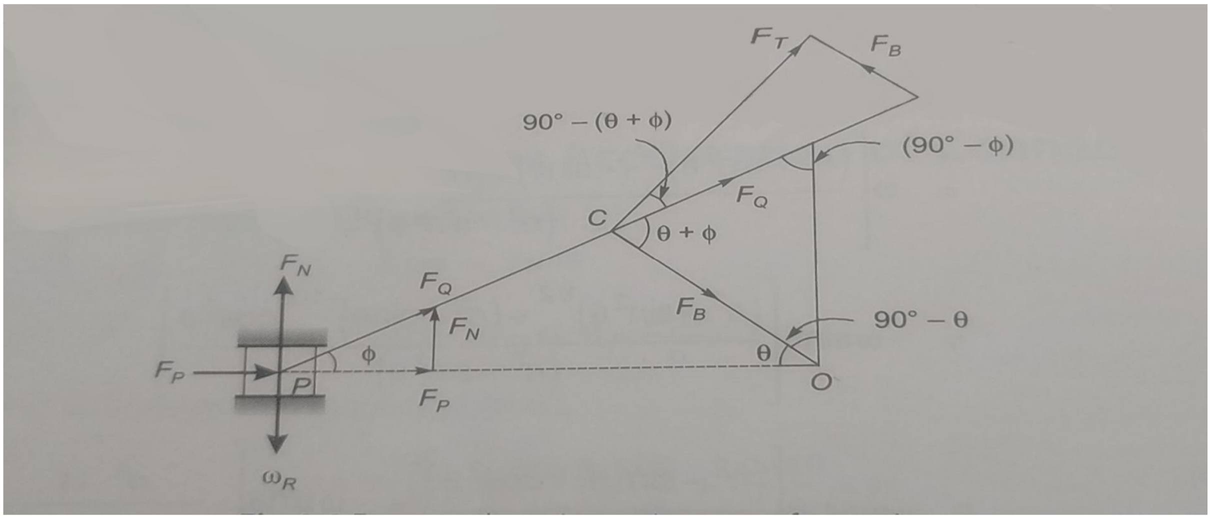


Motor torque variation with time



F_p = piston effort i. e. net force acting on the piston or crosshead pin along the line of stroke.
a_r or a_p = acceleration of reciprocating parts
F_L = net load on the piston due to spring
F_I = Inertia force
F_Q = Force acting along the connecting rod or connecting rod thrust
F_N = Thrust on the side of the cylinder walls or normal reaction on the guide bars
F_T =Crank-Pin effort i.e. thrust on crank shaft bearing
F_B = Component of F_Q along the crank
T =Crank effort
From the Figure:
$F_Q * \cos\varphi = F_P$
$F_Q = F_P/(\cos\varphi)$
$= F_P /(\sqrt{(1-(\sin^2\theta/ n^2)})$
$F_N = F_Q \sin\varphi$
$= F_P \sin\varphi / \cos\varphi$
$= F_P \tan\varphi$

Crank Pin Effort,
$FT = FQ \sin(\theta + \varphi)$
$= FP \sin(\theta + \varphi) / \cos\varphi$

Thrust on crank shaft bearings,
$FB = FQ \cos(\theta + \varphi)$
$= FP \cos(\theta + \varphi)/\cos\varphi$

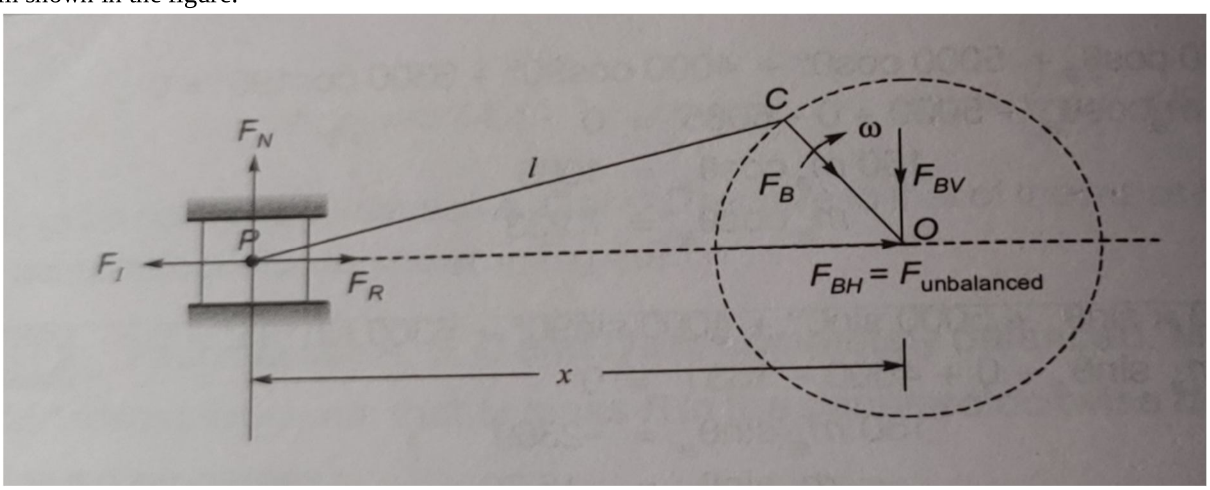
Crank Effort
$T = FT * r$
$= FP * r \sin(\theta + \varphi)/\cos\varphi$
$= FP * r (\sin\theta \cos\varphi + \cos\theta \sin\varphi)/\cos\varphi$
$= FP * r (\sin\theta + \cos\theta + \tan\varphi)$

Since,
$l \sin\varphi = r \sin\theta$
$\sin\varphi = r (\sin\theta)/l$
$= \sin\theta/n$ (n=l/r)
$\cos\varphi = \sqrt{(1 - \sin^2\varphi)}$
$= (1/n)*\sqrt{(n^2 - \sin^2\theta)}$
⇨ $\tan\varphi = \sin\theta / \sqrt{(n^2 - \sin^2\theta)}$

So we will get,
$T = FP * r (\sin\theta + \sin 2\theta / 2\sqrt{(n^2 - \sin^2\theta)})$
Since $\sin^2\theta$ is very small as compared to n^2
Therefore neglecting $\sin 2\theta$, we have
$T = FP * r (\sin\theta + \sin 2\theta/2n)$

C. *Balancing of Reciprocating Masses*

The various forces acting on the reciprocating parts of the machine, the resultant of all forces acting on the body due to inertia forces only is known as unbalanced or shaking force. Even if the resultant of all the forces due to inertia effects / forces is zero, then there will be no unbalanced force, but a unbalanced couple or shaking couple will be present. Let us consider a vertical reciprocating mechanism shown in the figure:



Reciprocating mechanism

F_R = force required to accelate the rotating parts
F_I = Inertia force due to reciprocating parts
F_N = Force on the sides of the cylinder walls or normal force acting on the crosshead guides
F_B = Force acting on the crank shaft bearing or the main bearing
F_R and F_I balances each other and similarly F_{BH} acts along the line of reciprocation and it is an unbalanced force or shaking force and required ti be properly balanced.

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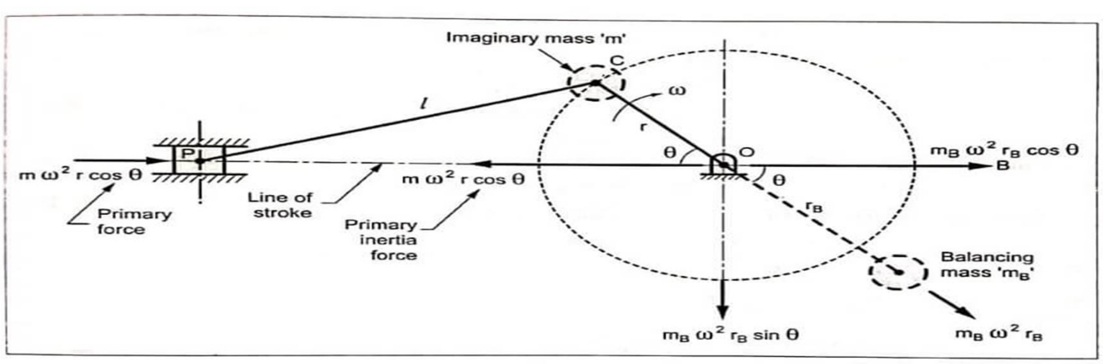
The force on the sides of the cylinder walls (F_N) and the vertical component of F_B are equal and opposite and thus form a shaking couple of magnitude F_N * x or F_{BV} * x. Thus, the shaking force and the shaking couple is produced by the effect of reciprocating parts. Since the shaking force and the shaking couple vary in magnitude and direction during the operational cycle, therefore they cause very objectionable vibrations.
Thus, the purpose of balancing the reciprocating masses is to eliminate the shaking force and shaking couple. In most of the mechanisms, we can reduce the shaking force and the shaking couple by adding appropriate balancing mass, but it is usually not practical to eliminate them completely. So, the reciprocating masses are only partially balanced. The masses rotating with the crankshaft are normally balanced and they do not transmit any unbalanced or shaking force on the body of the machine.

Primary and secondary unbalanced force of reciprocating masses
m = mass of reciprocating parts
l = length of connecting rod
r = radius of the crank
θ = angle of inclination of the crank with the line of stroke
ω = angular speed of the crank
n = ratio of length of connecting rod to the crank radius
approximate expression for the acceleration of the reciprocating parts is
$a_r = \omega^2 r(\cos\theta + \cos 2\theta/n)$
$F_R = F_I = m\omega^2 r(\cos\theta + \cos 2\theta/n)$
$F_{unbalanced} = F_U = F_R = F_I = m\omega^2 r\cos\theta + m\omega^2 r\cos 2\theta/n$
$= F_P + F_S$
Primary unbalanced force
$F_P = m\omega^2 r\cos\theta$
Secondary unbalanced force
$F_S = m\omega^2 r\cos 2\theta/n$
Important points :
1) $F_{P(max)} = m\omega^2 r$ for $\theta = 0°$ or $180°$
2) $F_{S(max)} = m\omega^2 r/n$ for $\theta = 0°, 90°, 180°$ and $360°$.
3) The primary force is maximum twice in one revolution of the crank.
4) The secondary force is maximum four times in one revolution of the crank.
5) Secondary unbalanced force is negligible in case of moderate speeds.
6) The unbalanced force due to reciprocating masses varies in magnitude but constant in direction while due to the revolving masses, the unbalanced force is constant in magnitude but varies in direction.

D. *Partial Balancing of Unbalanced Primary Forces in Reciprocating Machine*



The Primary unbalanced force ($m\omega^2 r\cos\theta$) may be considered as the component as the component of the centrifugal force produced by rotating mass m placed at the crank radius r, as shown in Figure.

The primary force acts from O to P along the line of stroke. Hence balancing of primary forces considered as equivalent of balancing of mass m rotating at the crank radius r. This is balanced by having a mass B at a radius b, placed diametrically opposite to the crank pin C. Horizontal component of centrifugal force of mass B is equals to $B\omega^2 b \cos\theta$.

The primary force is balanced if $B\omega^2 b \cos\theta = m\,\omega^2 r \cos\theta$

or Bb=mr.

but the rotating mass also has a component $m\,\omega^2 r \sin\theta$ perpendicular to the line of stroke which remains unbalalnced. This unbalanced force is zero at the end of stroke when $\theta = 0°$ or $180°$ and maximum at middle when $\theta = 90°$. The magnitude of the maximum unbalanced force remains same i.e. is equals to $m\,\omega^2 r$. Thus, instead of slide to or for on its mounting, the mechanism tends to jump up and down.

To minimize the effect of unbalanced force, a compromise is, usually made, i.e. 2/3 of the reciprocating mass is balanced (or a value between ½ and ¾). As a compromise let a fraction ‘c’ of the reciprocating is balanced, such that

cmr = Bb

Unbalanced force along the line of stroke

$= m\,\omega^2 r \cos\theta - B\,\omega^2 b \cos\theta$

$= m\,\omega^2 r \cos\theta - cm\omega^2 r \cos\theta$

$=(1-c)\, m\,\omega^2 r \cos\theta$

And unbalanced force along the perpendicular to the line

$= B\,\omega^2 b \sin\theta = cm\omega^2 r \sin\theta$

Resultant unbalanced force at any instant,

$\sqrt{[(1-c)\, m\,\omega^2 r \cos\theta]^2 + [cm\omega^2 r \sin\theta]^2}$

The resultant unbalanced force is minimum when c=1/2.

If the balancing mass is required to balance the revolving masses as well as reciprocating masses then,

$Bb = m_1 r + cmr = (m_1 + cm)\, r$

Where m_1 = magnitude of revolving masses

m= magnitude of reciprocating masses

E. Transmissivity

It is defined as the ratio of the force transmitted to the force applied. Transmitted force implies the one which is being transmitted to the foundation or to the body of a particular system. Applied force is the external agent that cause the force to be generated in the first place and be transmitted.

Transmissibility T= output/ input

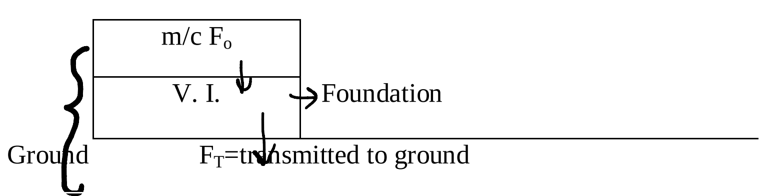
T>1 means amplification and maximum amplification occurs when forcing frequency and natural frequency of the system coincide.

There is no unit designation for transmissibility, although it may sometimes be referred to as the Q factor.

The transmissibility is used in calculation of passive efficiency.

The lesser the transmissibility the better is the damping or the isolation system.

T<1 is Desirable, T=1 acts as a rigid body, T>1 is Undesirable.



$F_T <<<<< F_o$,

Transmissibility $E = F_T/F_o$

$F_T = \sqrt{(sA)^2 + (c\omega A)^2}$

$F_T = sA\sqrt{1 + (2z\omega/\omega_n)^2}$

$F_o = sA\sqrt{(1-(\omega/\omega_n)^2)^2 + (2z\,\omega/\omega_n)^2}$

$E = F_T/F_o$

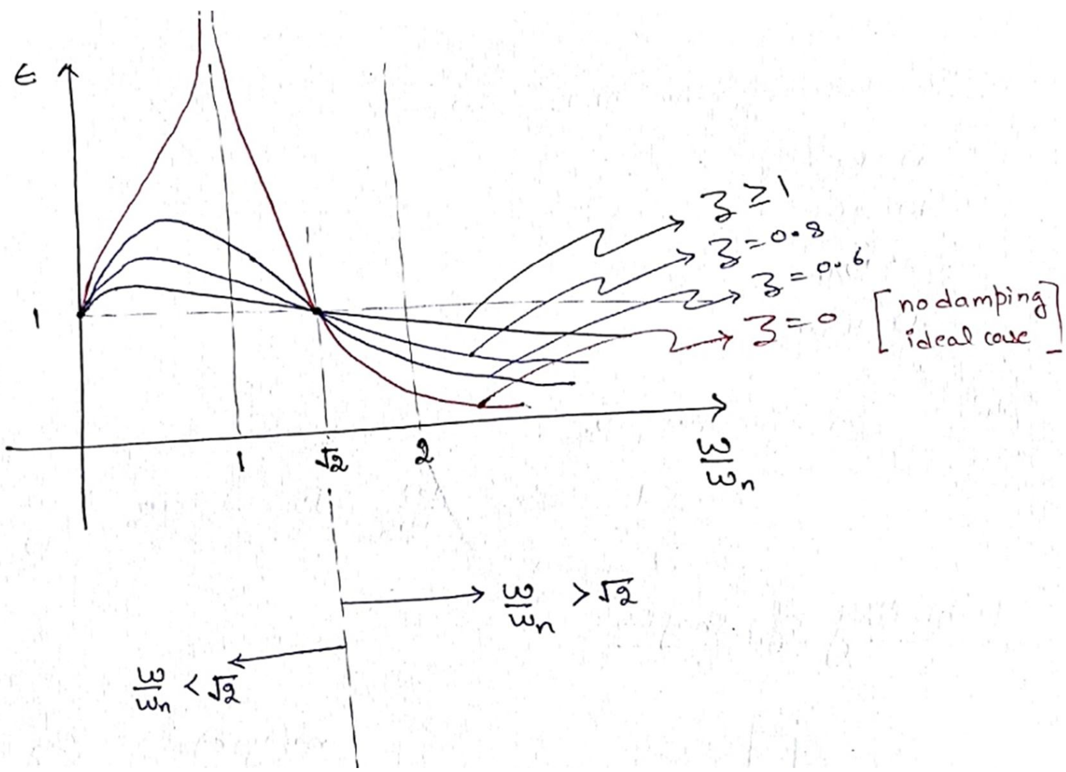
$$= \frac{\sqrt{1+(2z\omega/\omega_n)^2}}{\sqrt{(1-(\omega/\omega_n)^2)^2+(2z\,\omega/\omega_n)^2}}$$

E depends upon

1. ω/ω_n
2. z

If $\omega/\omega_n = 0 \Rightarrow E=1$

$\omega/\omega n = \sqrt{2} \Rightarrow E=1$ for all values of z



- increasing underdamping (z↓)

 E↑ if $\omega/\omega_n < \sqrt{2}$

E↓ if $\omega/\omega_n > \sqrt{2}$

E will remain same if $\omega/\omega_n = \sqrt{2}$

- Vibration isolation will be effective when E<1

  => if $\omega/\omega_n > \sqrt{2}$

- In effective V.I. zone

  $\omega/\omega_n > \sqrt{2}$

  - No damping is the best $E \to 0$
  - Damping is harmful.

The FEM method is used to analyse the stress state of an elastic body with a given geometry, such as leaf spring. In this paper the analysis of leaf spring in Maruti 800 is intended for study using FEM software using ANSYS.

F. Basic Data Of Leaf Spring

1) Total length of the spring (Eye to Eye) = 1120 mm
2) No. of full length leaves (n_f) = 1
3) Thickness of leaf (t) = 5 mm
4) leaf spring width (b) = 50 mm
5) Total load = 500 N
6) BHN = 420 – 430 HB with hardened and tempered

XII. CAD MODEL

Alto 800 paper mono leaf springs with a length of 700mm and a width of 40mm were chosen for this car. The CAD model was prepared with CATIA software. Fig. 1 shows CAD model of mono leaf spring under study.

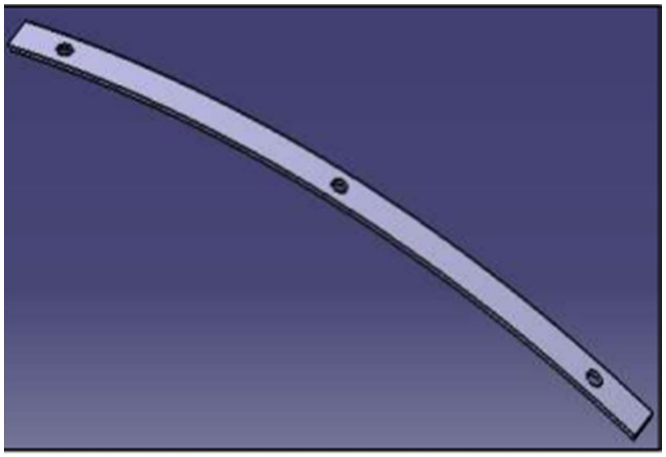


Fig. 1: CAD Model of Mono Leaf Spring of Alto800 Car

XIII. MESHING

Hyper mesh software was used for meshing. A Quad type of meshing was used. Fig. 2 shows Quad meshing on mono leaf spring.

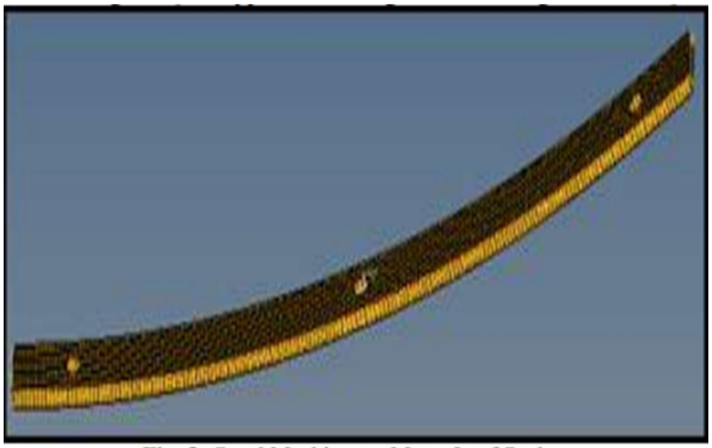


Fig. 2: Quad Meshing on Mono Leaf Spring

The following are the specifications of meshing:
Number of nodes: 1873
Number of elements: 1678
Element size = 4 mm

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Volume 10 Issue VIII Aug 2022- Available at www.ijraset.com

XIV. BOUNDARY CONDITIONS

After applying the boundary conditions, a vertical load is applied to joint 2 and fixed at the other two sides. Fig.3 shows Meshed model with applied boundary conditions. In static condition, the earth's gravitational pull (mg) acts through the centre of gravity and the reaction (remember: to every action there is an equal and opposite reaction) acts through the contact patches between the tyre and the road.

Total weight of the car = 25987.6 N

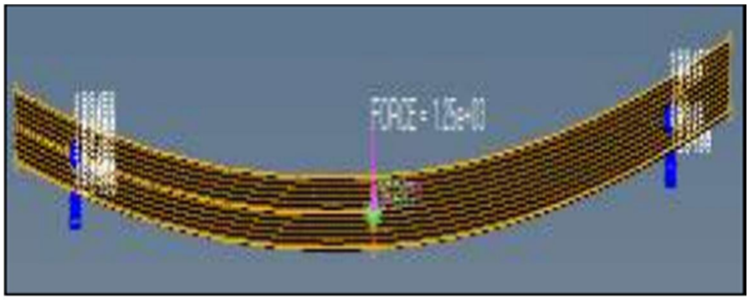

Fig. 3: Meshed Model with Applied Boundary Condition

This weight must be divided into front axle weight and rear axle weight. 52% of total weight is taken by front axle and 48% of total weight is taken by rear axle.

1) Front axle weight = 13513.5 N
2) Reaction at one wheel = 6756.8 N
3) Rear axle weight = 12474.05 N
4) Axle weight on one wheel = 6237.02 N
5) Assuming 5 number of plates, Load on leaf spring= 1247.4 N
6) Deformation and Stress in leaf spring,
7) Stress at centre of constant cross section is given by 175.38 Mpa
8) (Same for both steel and Glass fibre)
9) Maximum Deflection at load is given by,
10) For Steel = 1.4 mm
11) For Glass fibre = 0.658 mm

Table - 1
Mechanical Properties of Steel

Property	Value
Young's modulus	$2*10^5$
Poisson's Ratio, ν	0.3
Density, ρ	7.85×10^{-6} kg/mm^3
Tensile Yield Strength	250 Mpa
Compressive Yield Strength	460 MPa

Table - 2
Mechanical Properties of GFRP

Property	Value
Young's modulus in z-direction	$4*10^5$
Poisson's Ratio, ν	0.36
Density, ρ	6×10^{-6} kg/mm^3
Tensile Yield Strength	2500 Mpa
Compressive Yield Strength	3150 MPa
Spring Constant (N/mm)	4.83
Maximum Compression (mm)	83
Shear Stress (N/mm^2)	83

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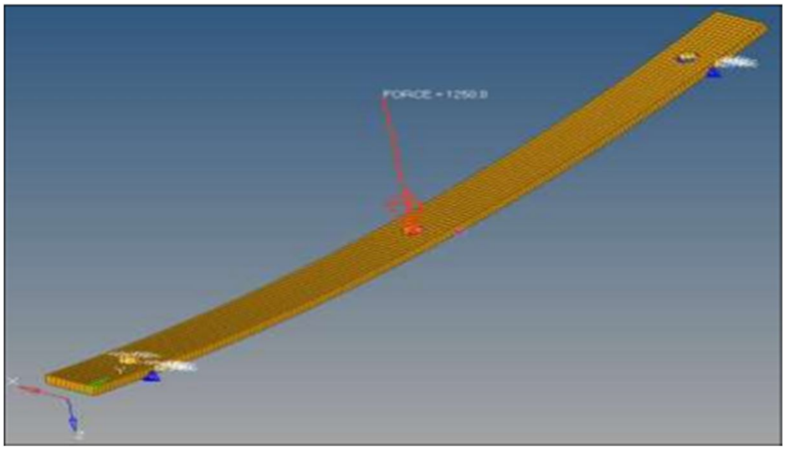
The calculated load was applied at joint 2 of leaf spring or existing material and glass fibre. Fig. 4 shows Meshed model of Glass fibre leaf spring and applied boundary conditions.

XV. RESULTS

Deformation stress and von Mises stress were obtained by applying the calculated loads to the leaf spring while the other end was fixed.. Table-3 shows results for loading conditions on existing material and glass fibre leaf spring respectively. Fig. 5 to Fig. 8 shows deformation and Von-mises Stresses for existing material and glass fibre respectively.

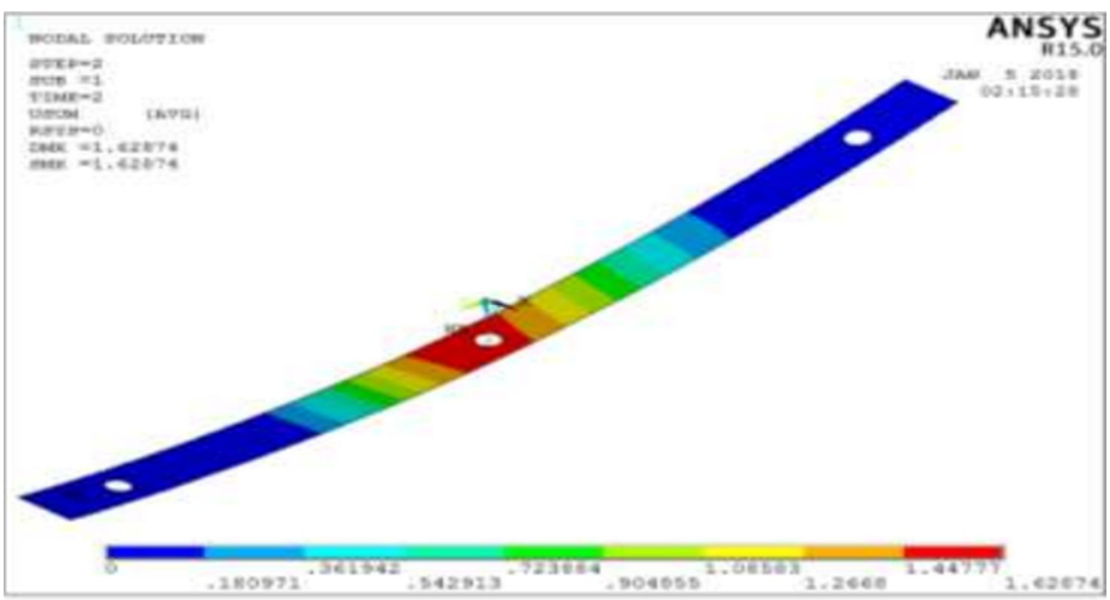


Meshed Model of Glass Fibre Leaf Spring & Applied Boundary Conditions

From deformation and Von-mises Stresses plot given below it was observed that leaf spring with glass fibre material is most feasible for considered loading conditions. Thus leaf spring with glass fibre material is highly recommended for fabrication.

Table - 3
Deformation and Von-Misses Stresses

Type of Analysis	Steel		Glass Fibre	
	Deformation (mm)	Stress (Mpa)	Deformation (mm)	Stress (Mpa)
Analytical	1.4	175.3	0.655	175.38
FEA	1.6	175.7	0.655	175.73



Deformation Plot for Existing Material

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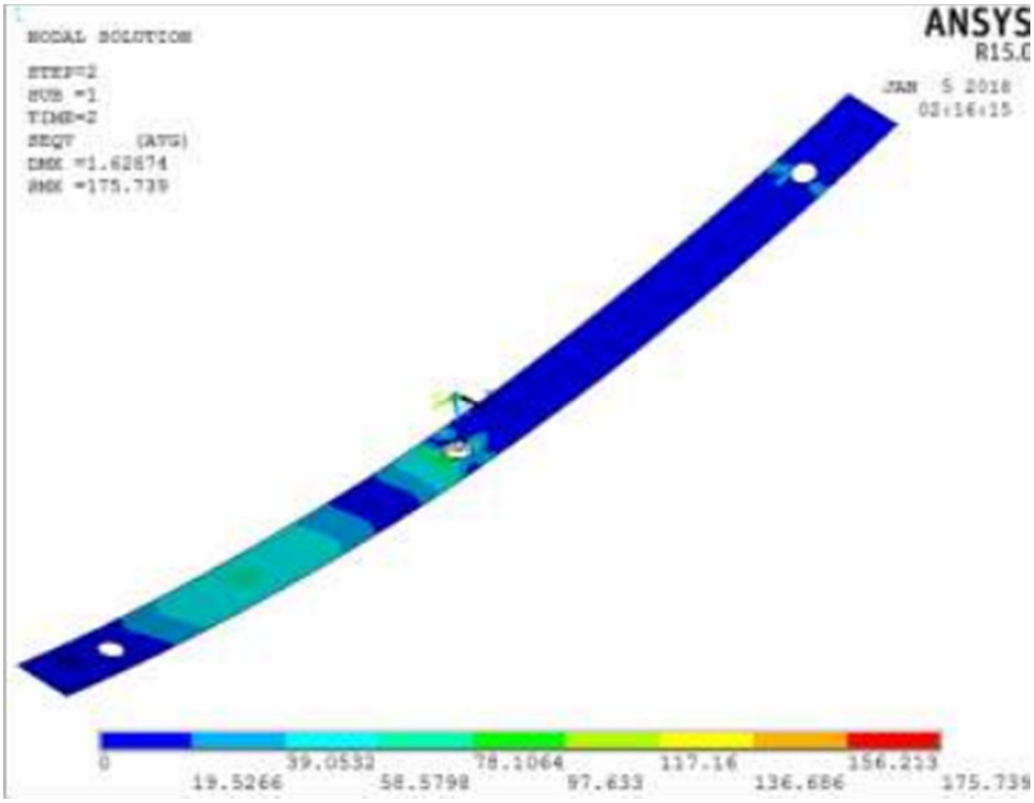
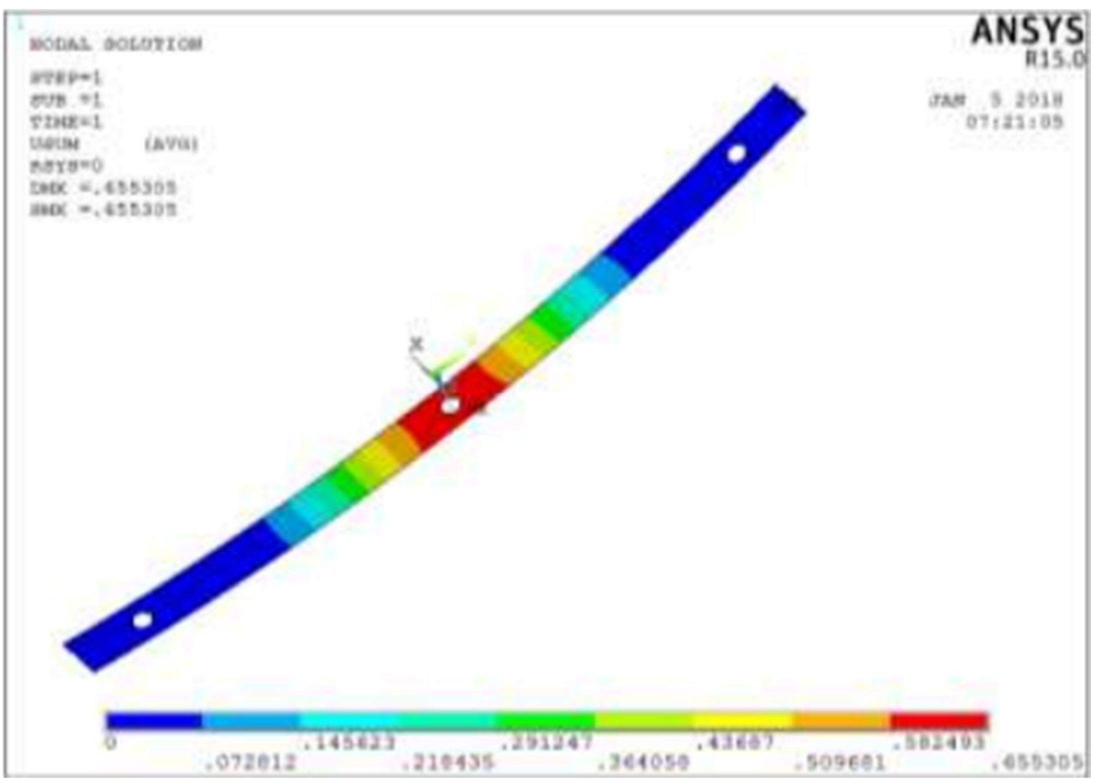


Von-Misses Stresses for Existing Material



Deformation Plot for Glass Fibre Material

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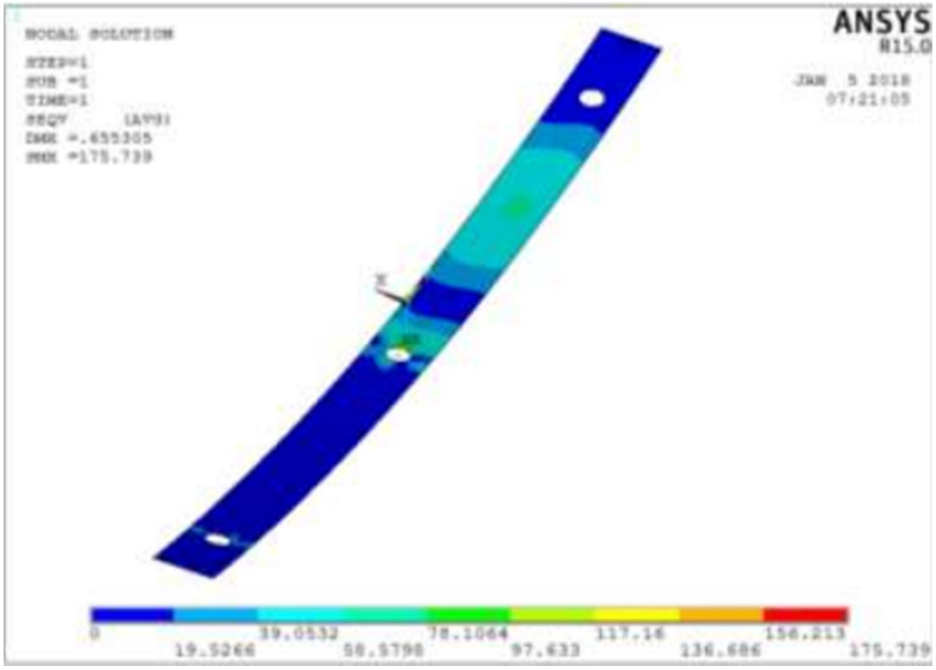


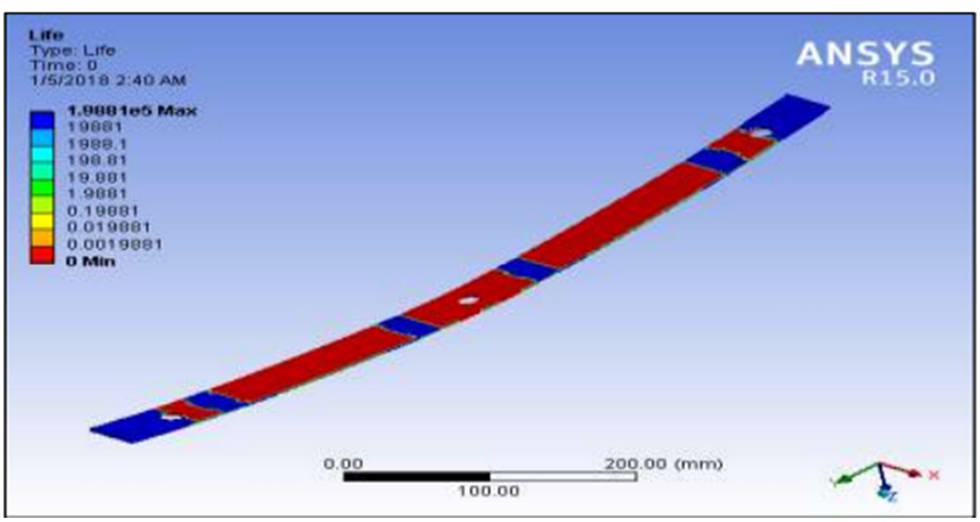
Von-Mises Stresses for Glass Fibre Material

XVI. FATIGUE ANALYSIS

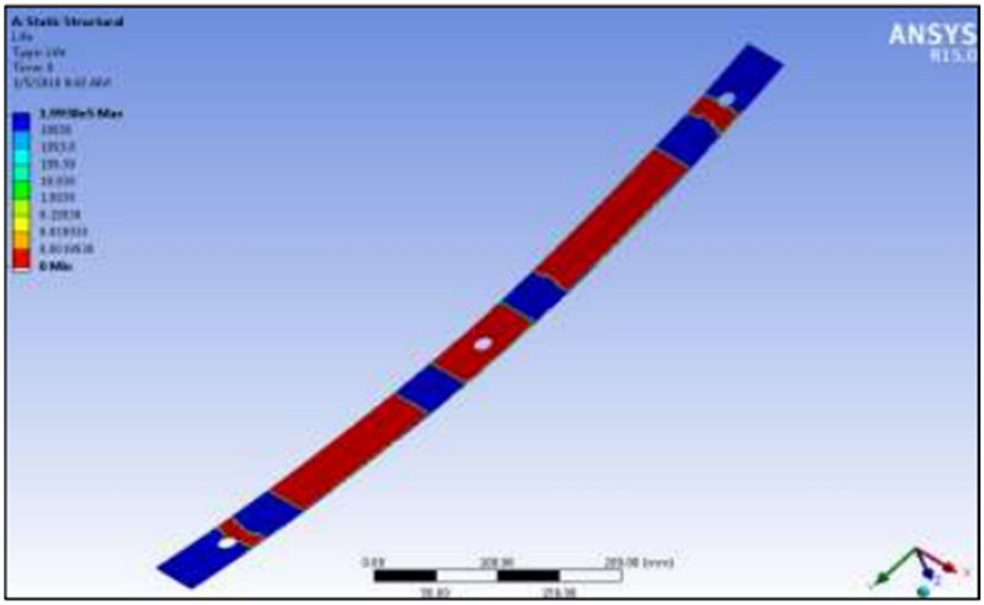
Fatigue is generally understood to mean the gradual deterioration of materials subjected to cyclic loading. In fatigue testing, a specimen is subjected to periodically varying constant amplitude stress and Life of the existing Mild Steel & Glass fibre leaf spring Specimen obtained by FEA (ANSYS) Method. Table-4 shows results for Fatigue life of mild steel and Glass Fibre leaf spring respectively. Fig. 9 and 10 shows Fatigue life of mild steel and Glass Fibre leaf spring.

Table - 4
Fatigue Life of Leaf Spring

Sr. No.	Material of leaf Spring	Fatigue life (cycles)
1.	Mild Steel (Existing Material)	1, 98,810
2.	Glass Fibre	1, 99,380



Fatigue Life of Mild Steel Leaf Spring



Fatigue Life of Glass Fibre Leaf Spring

From Fatigue life plots it was observed that leaf spring with glass fibre material have more life than the Fatigue life of mild steel leaf spring. Hence, the newly material optimized glass fibre is recommended as high structurally stable than the steel mono leaf spring and considered for fabrication for further experimental validation.

XVII. FABRICATION OF TEST MODEL

From numerical analysis it was decided to fabricate leaf spring composite reinforced glass fibre for validating results obtained numerically. The fabrication of glass fibre leaf spring has been done layer by layer to achieve required thickness with help of epoxy resin as an adhesive with the added cobalt 10% cobalt and hardener 10% to increase the rate of drying process as shown in fig.11 below. Once after the reinforcement process is completed the assembly is allowed to dry for nearly 48 hours. Then the required glass fibre surface is removed for fine finishing.

XVIII. SCOPE FOR FUTURE WORK

The project you see today is a conceptual setup. This setup was intended to test the fatigue life of composite leaf springs where much research and development is underway. The future scope of this project is considered to be very extensive and, among other things, can be modified for different test parameters. Currently, the accuracy of machine output relies on strain gauge readings, but in the future, advanced micro-crack detection technology may improve it.

XIX. RESULT & DISCUSSION

Table- 5. Shows numerical and experimental results for tested composite glass fibre leaf spring. Fatigue testing is performed on glass fibre leaf spring rod. It shows that at 1250N load & frequency 100Hz, No Crack observed after 1,00,000 cycles of glass fibre leaf spring structure with composite as a material in Experimental Test. Fatigue analysis has been performed for glass fibre leaf spring structure by using FEA method and No. of cracks observed is 199380 cycles. Based on Experimental and FEA it can be concluded that the glass fibre leaf spring has infinite life because it can withstands above 1,00,000 cycles in both tests and results are correlated.

Table - 5
Numerical & Experimental Results

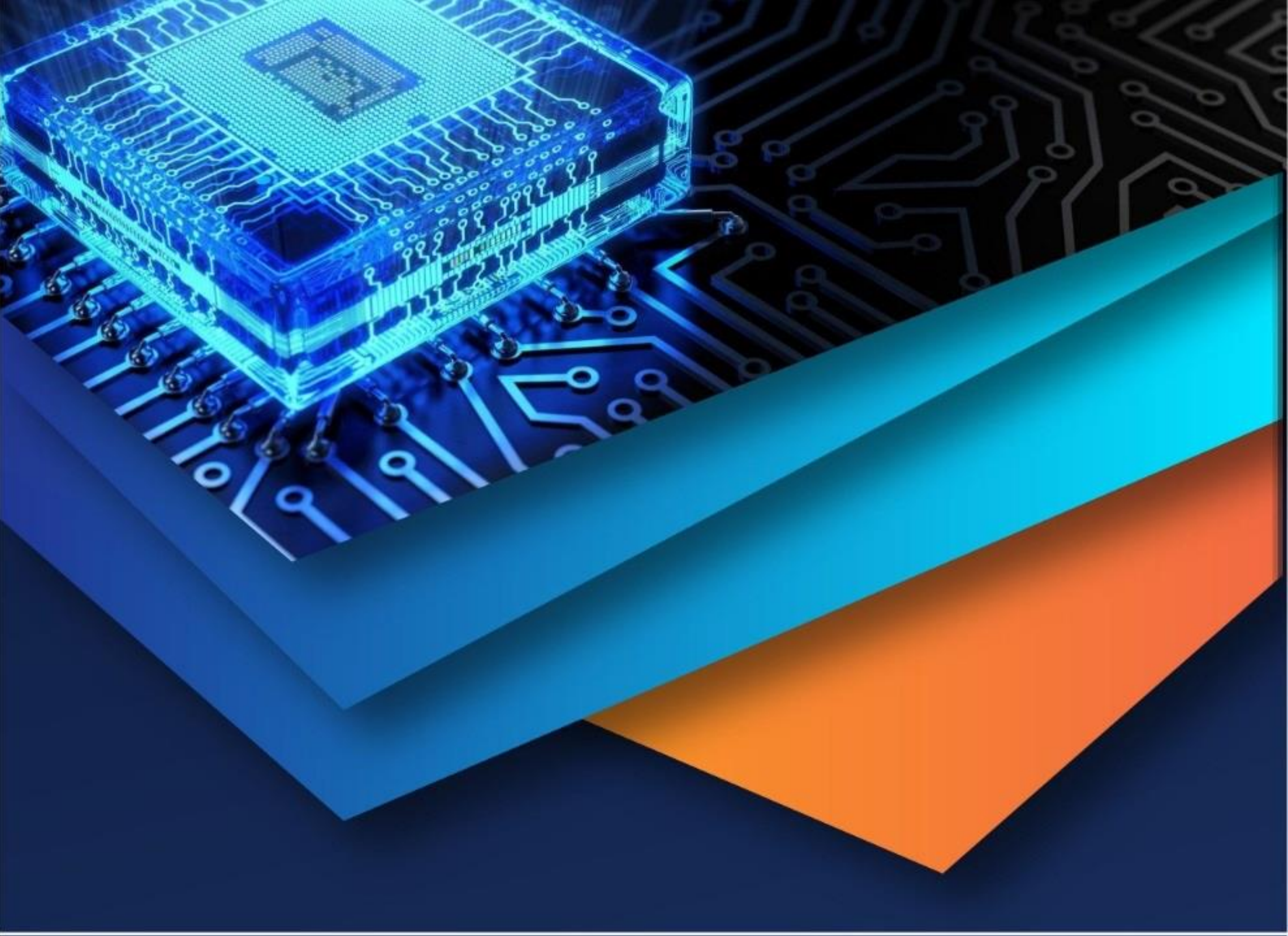
Method of Analysis	No. of cycles
Finite Element Analysis	1, 99,380 cycles
Experimentation	1,00,080 cycles (No Cracks)

XX. CONCLUSIONS

The design approach was simple, identifying the critical section of the machine, superimposing of forces and moments, followed design based on biaxial loading consideration. Choice of design and material selection was done on local availability, so any replacement of parts or machining operations can be done within the workshop itself. Small amount of vibrations was nullified due to the heavy structure and by using rubber pads on four corner. A provision provided at the pivot assembly is that by knowing the lateral difference in distance between the two horizontal pivots, manual adjustment is made by increasing or decrease in washer thickness with machining accuracy so some machine scatter (lateral misalignment / Co- Axially) can be adjusted which showed a better performance in run, due to this rise in temperature effects can be almost nullified. Gripping methods used in this machine showed that loosening of grips during dynamic conditions can be identified by the movement of unscrewing of locking nut from its locked position which was known before. Due to its symmetry of a load supporting member the choice of design is made such that, use of bolted connection, pin joint and locking its movement by a washer arrangement assembly, will help an operator to easily dismantle the assembly without damage and will help him in calibration of machines whenever necessary. Initial tightening of fastener arrangement required for better and stable operation. Bearing should be lubricated time to time. Simpler Design helped in understanding dynamic scatter in these type of machines. Any misalignments produce in the system will overcome by operator easily. Machine generates acceptable results.

The steel leaf spring replaced by composite glass fibre reinforced material. The Finite Element Analysis has been successfully performed to find out the deformation and Von-mises stresses for both Mild steel and Composite reinforced Glass Fibre material.

The comparison is listed in Table- 3. Fatigue life has been obtained using FEA and experimentally shown in Table- 5. Weight has been reduced approximately by 65.84% without compromising strength of leaf spring.



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