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Comparative Analysis of Conventional and Composite Connecting Rod

A Project Report

Submitted by

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BONAFIDE CERTIFICATE

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ABSTRACT

The main objective of this project work titled "Comparitive Analysis of Conventional and Composite Connecting Rod" is to compare the structural characteristics of conventional and composite material connecting rod. In order to improve the fuel economy, performance and weight of the engine, the moving parts of the engine should be manufactured weightless.

Connecting rod is an important component that should be made weightless to reduce the inertia forces and high reciprocating forces. Metal matrix materials are used as an alternate material for connecting rods used in diesel powered automotive engine. The material used for the connecting rod is modified so as to maintain its weight while maintaining its shape, mechanical strength, dimensional stability and structural integrity at all engine operating conditions. The structural characteristics of connecting rod is determined using finite element method and simulated using commercial coded finite element analysis software ANSYS 8.0.

The angular velocity and angular acceleration is calculated for three different engine speeds and are plotted in the graphs. The connecting rod is meshed with different element sizes and the stress results were calculated. On comparison between conventional and composite connecting rod, the results obtained through finite element analysis show that the composite material is best. This is because the composite material connecting rod weighs less and withstands the stress produced.

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LIST OF SYMBOLS AND ABBREVIATIONS

D : Diameter of piston

P : Maximum pressure of gas

A : Cross sectional area of piston

m_R : Mass of reciprocating parts

ω : Angular speed of crank

ϕ : Angle of inclination of the connecting rod with the line of stroke

θ : Angle of inclination of the crank from top dead centre

r : Radius of crank

l : Length of connecting rod

n : Ratio of length of connecting to radius of crank = l/r

F_L : Force on the piston due to pressure of gas

F_I : Inertia force of the reciprocating parts

F_P : Net force acting on the piston

F_C : Force in the connecting rod

F : Friction force

D_1 : Cylinder bore

t_R : Axial width of rings

n_R : Number of rings

p_R : Pressure of rings

μ : Coefficient of friction

v_{PO} : Velocity of the piston

a_P : Acceleration of the piston

ω_{PC} : Angular velocity of connecting rod

α_{PC} : Acceleration of connecting rod

E_1 : Longitudinal Modulus, GPa

E_2 : Transverse Modulus, Gpa

v_{12} : Major Poisson's Ratio

v_{21} : Minor Poisson's Ratio

F_{1t} : Longitudinal Tensile strength, MPa

F_{2t} : Traverse Tensile strength, MPa

F_6 : In-plane shear strength, MPa

$\in_{1t}^{u}$: Ultimate Longitudinal tensile strain

$\in_{2t}^{u}$: Ultimate Traverse Longitudinal tensile strain

F_{1e} : Longitudinal Compressive strength, MPa

F_{2c} : Traverse Compressive strength, MPa

α_1 : Longitudinal Thermal Expansion Coefficient

α_2 : Traverse Thermal Expansion Coefficient

β_1 : Longitudinal Moisture Expansion Coefficient

β_2 : Traverse Moisture Expansion Coefficient

ε_1 : Normal strain in longitudinal direction

ε_2 : Normal strain in transverse direction

ε_x : Normal strain in X- direction

ε_y : Normal strain in Y- direction

σ_1 : Normal stress acting in the longitudinal direction of the lamina, Mpa

σ_2 : Normal stress acting along the transverse direction of the lamina, Mpa

σ_x : Normal stress acting along the X- direction of the lamina, Mpa

σ_y : Normal stress acting along the Y- direction of the lamina, Mpa

τ_{12} : Shear stress acting in the 12-direction of the lamina, Mpa

τ_{xy} : Shear stress acting in the XY-direction of the lamina, Mpa

E : Modulus of elasticity or Young's modulus for the material of the column

k : Least radius of gyration of cross-section,

W_C : Ultimate crushing load for the column

W_B : Buckling load

W_E : Crippling load obtained by Euler's formula

W_{cr} : Crippling load

σ_c : Crushing stress or yield stress in compression

a : Rankine' constant

A : Area of Cross-section of the column

L : Equivalent length of the column

l : Length of connecting rod

C	:	Constant, representing the end conditions of the column
p	:	Normal pressure
p_o	:	Pressure constant
P_t	:	Tensile load acting on connecting rod
P_c	:	Compressive load acting on connecting rod
t	:	Thickness of the connecting rod at the loading surface
r_1	:	Radius of crankpin
Z	:	Section modulus
I_{xx}	:	Movement of inertia of the section about X-axis
I_{yy}	:	Movement of inertia of the section about Y-axis
k_{xx}	:	Radius of gyration of the section about X-axis
k_{yy}	:	Radius of gyration of the section about Y-axis
a_2	:	Angular acceleration of connecting rod
β	:	Connecting rod angle with positive direction of X axis
$a_{c.gX}$	:	X components of the acceleration of the C.G. of connecting rod
$a_{c.gY}$	:	Y components of the acceleration of the C.G. of connecting rod
$\sigma_{c(max)}$	:	Whipping Stress
σ_{max}	:	Maximum bending stress
M_{max}	:	Maximum bending moment
$[\underline{K}]$	:	Assembled stiffness matrix
$\underline{\vec{\Phi}}$	:	Vector of nodal displacement
$\underline{\vec{P}}$	:	Vector of nodal forces
$[K]$	:	Stiffness matrix
$\vec{P}$	:	Load vector
$\vec{\Phi}$	:	Nodal Displacement
ANSYS	:	Analysis of Engineering Systems
MMC	:	Metal Matrix Composites
CMC	:	Ceramic Matrix Composites
PMC	:	Polymer Matrix Composites
FEA	:	Finite Element Analysis
FEM	:	Finite Element Method

IGES	:	Initial Graphics Exchange Specification
CAD	:	Computer Aided Geometry
T.D.C	:	Top Dead Centre
B.D.C	:	Bottom Dead Centre
BLF	:	Buckling Load Factors

Composite Connecting Rod:

It is a light weight rod comprising a split crankshaft receiving end, a wrist pin receiving end and an elongated intermediate connecting member made up of metal matrix fibre and reinforcement materials.

Conventional Connecting Rod:

It is intermediate member between piston and crankshaft made up of steel or cast iron.

CHAPTER 1
INTRODUCTION

1.1 BACKGROUND

The automobile engine connecting rod is a high volume production, critical component. It connects reciprocating piston to rotating crankshaft, transmitting the thrust of the piston to the crankshaft. The connecting rods are manufactured by drop forging process and it should have minimum weight, adequate strength and stiffness. The material mostly used for connecting rods varies from mild carbon steels to alloy steel. Connecting rods for automotive applications are typically manufactured by forging process from either steel or powdered metal. They could also be casted. However, castings could have blow-holes which are detrimental from durability and fatigue points of view. The fact that forgings produce blow-hole-free and better rods gives them an advantage over cast rods. Between the forging processes, powder forged or drop forged, each process has its own advantages and disadvantages.

Powdered metal manufactured blanks have the advantage of being near net shape, reducing material waste. However, the cost of the blank is high due to the high material cost and sophisticated manufacturing techniques. With steel forging, the material is inexpensive and the rough part manufacturing process is cost effective. Bringing the part to final dimensions under tight tolerance results in high expenditure for machining as the blank usually contains more excess material.

1.2 PROBLEM DEFINITION

Traditionally, engines have been made of metal, usually steel or cast iron. Steel and cast iron engines are useful, except they are quite heavy and consume considerable amounts of fuel. The engines exert large compressive forces, considerable torque, and substantial secondary harmonic vibrations which have to be dampened by counter balancing pistons, flywheels, dampeners, etc. The moving metal parts of cast iron and steel engines generate high centrifugal, reciprocating, and inertial forces, momentum, and loads. Generally, the weight of the engine adversely affects its performance, efficiency, and power.

Connecting rod, the main component of the automobile engine should posses minimum weight so that the fuel consumption, increased speed of operation and high horsepower is obtained while maintaining its structural integrity. Normally connecting rods are made up of steel or powdered metal which are quiet heavy. In this work, a light weight metal matrix composite material is used as an alternate material for a diesel powered engine and its structural characteristics are analyzed using Finite element method. First the forged steel connecting rod is to be analyzed by assigning isotropic material properties. Then the composite material is to be analyzed by providing orthotropic material properties.

1.3 THE SCOPE OF PROJECT

Introducing new materials for improving the performance of engine, and to reduce the weight of the engine has a good scope in the automobile field. A slight modification by introducing an alternate material provides good improvement. In that way, using composite as a material for connecting rod is useful. This reduces the weight of the reciprocating parts and thereby reduces the weight of the engine.

1.4 LITERATURE REVIEW

Webster *et al.* (1983) performed three dimensional finite element analysis of a high-speed diesel engine connecting rod. For this analysis they used the maximum compressive load which was measured experimentally and the maximum tensile load which is essentially the inertia load of the piston assembly mass. The load distributions on the piston pin end and crank end were determined experimentally.

Holtzberg et al. (1984) developed a composite connecting rod comprising a laminated amide-imide resinous polymeric connecting rod comprising a plurality of plies, each ply having an amide-imide resinous polymeric matrix with a fabric layer comprising a fibrous reinforcing material selected from the group consisting essentially of graphite and glass. It has a laminated amide-imide connecting rod having a split crankshaft-receiving end, a wrist pin-receiving end and an elongated intermediate connecting member extending between crankshaft-receiving end and wrist pin-receiving end. It has a split crankshaft-receiving end having a detachable portion and an attached portion. The connecting rod was developed with less weight but it failed to bear the load acting on it.

Folgar *et al.* (1987) developed a fiber FP/Metal matrix composite connecting rod with the aid of FEA, and loads obtained from kinematic analysis. Fatigue was not addressed at the design stage. However, prototypes were fatigue tested. The investigators identified design loads in terms of maximum engine speed, and loads at the crank and piston pin ends. They performed static tests in which the crank ends and the piston pin end failed at different loads. Clearly, the two ends were designed to withstand different loads.

Georges Cahuzac et al. (1996) developed a connecting rod made of composite material, especially for internal-combustion engines, the composite material being formed by a reinforcement embedded in a cured material, and the connecting rod comprising, a central shank extended respectively, at its two ends, by a connecting-rod big end and a connecting-rod small end . Connecting rod is produced entirely as a single component made of composite material and the reinforcement of the composite material comprises superimposed plies of crossed straight filling yarns. The filling yarns, which extend parallel to the longitudinal extension, constitute at least twice the volume percentage of the filling yarns which extend in each of the directions. This rod failed in fatigue and the safety factor was kept high to keep the design safe.

CHAPTER 2

CONVENTIONAL CONNECTING ROD

2.1 INTRODUCTION

A connecting rod assembly is used for connecting a piston to a crankshaft in an internal combustion engine. The connecting rod assembly typically includes a connecting rod which is attached to a cap using a plurality of bolts. The connecting rod includes a first end which is attached to a piston using a piston pin. The opposite end includes a yoke with a semi-circular bearing portion which mates with a corresponding semi-circular bearing portion in the cap to define an opening for receiving the crankshaft therein. Typically, each bolt includes a head at one end and an opposite end which is threadingly engaged with a nut. To attach the cap to the yoke, it is necessary to place a wrench on both the head and the nut while tightening the nut to prevent rotation of the bolt.

It is also known to provide the connecting rod with a bolt seat against which the bolt head engages. The bolt seat must be machined into the connecting rod and therefore a small fillet with a transverse surface exists adjacent to the bolt seat. It is known to provide the bolt head with a non-circular cross section which engages the fillet or transverse surface adjacent the bolt seat to prevent rotation of

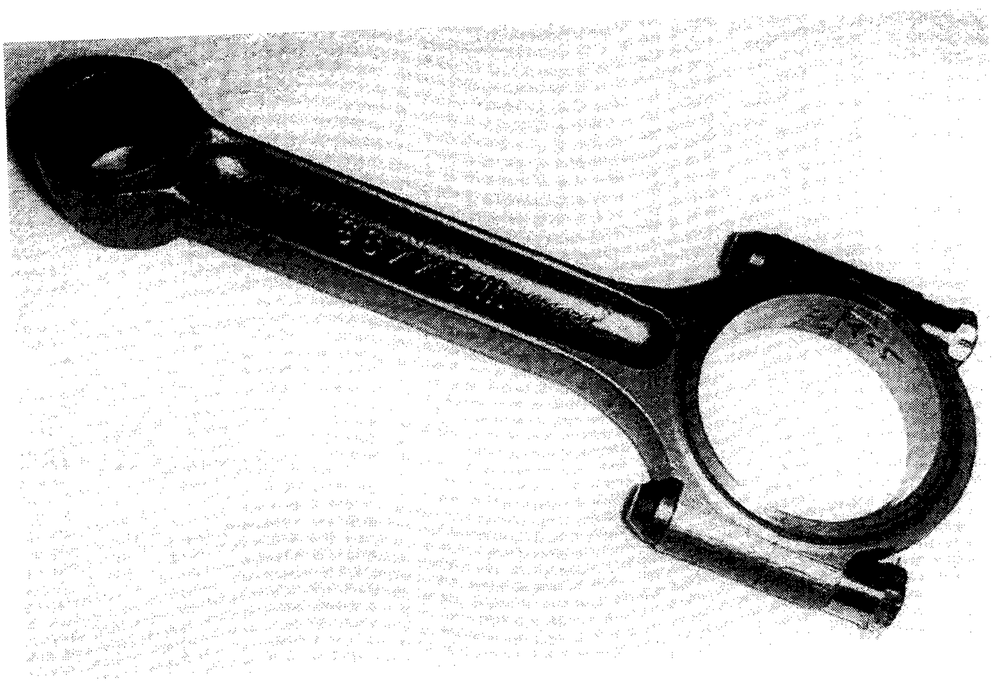


Figure 2.1 Connecting Rod

the bolt while tightening the nut. However, the peripheral edge of the bolt which engages the fillet and/or transverse surface adjacent to the bolt seat may cause undesirable stress concentrations in the connecting rod which may result in deformation or damage to the connecting rod.

2.2 TYPES OF CONNECTING ROD

Based on cross sections employed, the connecting rods are classified as

i) I-section
ii) H-section
iii) Circular section
iv) Tubular section
v) Rectangular section

Based on materials used, the connecting rods are classified as

i) Powdered metal connecting rods
ii) Forged steel connecting rods
iii) Aluminum connecting rods
iv) Titanium connecting rods

The cross section of the shank may be rectangular, circular, tubular, I-section or H-section. Generally circular section is used for low speed engines while I-section is preferred for high speed engine.

2.2.1 I-Beam Connecting Rod

The I-Beam rod is the most popular type of connecting rod. They are cheap

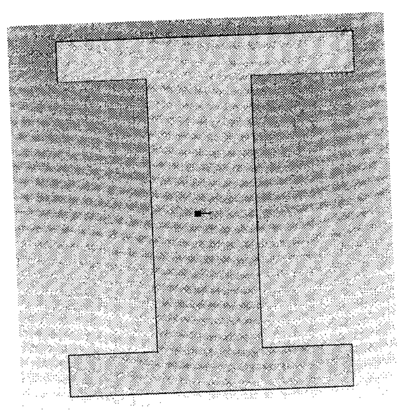


Figure 2.2 I-Section

to make and very reliable. They can be made as strong as is needed for the application. I-Beam rods can handle much more power than they were originally designed for with some careful prepping. There are inexpensive rods for most popular engines that can handle more power than prepped stock rods.

2.2.2 H-Beam Connecting Rod

H-beam rods are becoming more popular. They have some benefits over an I-Beam for certain applications. For high horsepower, low rpm engines, the rods need more strength in compression. For high rpm engines, rods need tensile strength. The tensile strength of a rod is most dependent on the material and the cross section of the rod. For compressive loads, it is a little different. When a rod fails from compressive loads, it bends. The compressive loads are not straight; this makes the shape of the rod critical to strength. The design of an H-Beam rod helps them resist bending more than an I-Beam design.

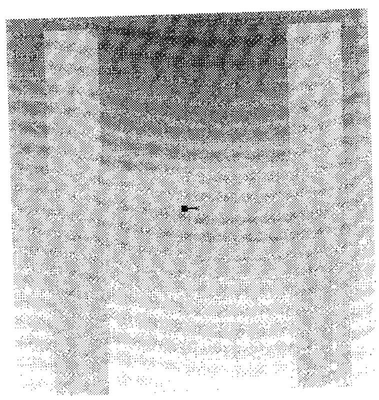


Figure 2.3 H-Section

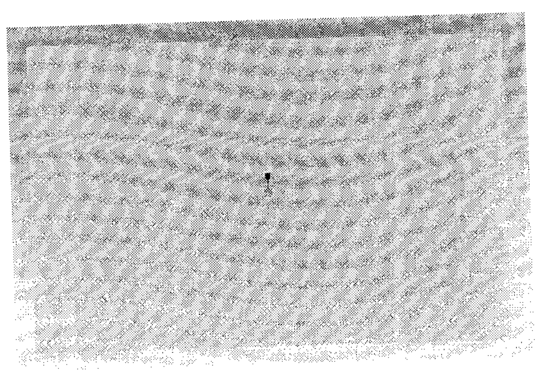


Figure 2.4 Rectangular section

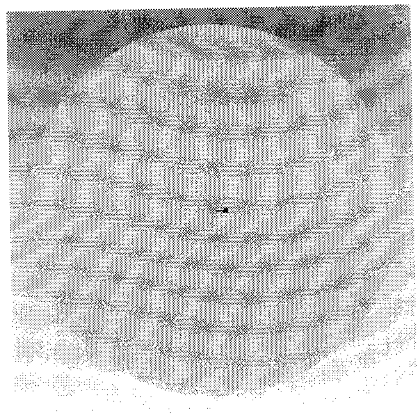


Figure 2.5 Circular section

2.2.3 Powdered Metal Connecting Rod

This is a technique that has gained quite popularity lately. In this technique steel is melted and poured it into a mold (cast) or heated up just below the melting point and stamp it into shape with several tons of force (forge), the rod is made from powered metal. The powdered material is compressed into a mold; it is then heated to the melting point to fuse it into one piece. This technique is to make a stronger rod by reducing internal stresses. Since the rods are formed from powdered metal before being heated, the shape can be very precise, reducing

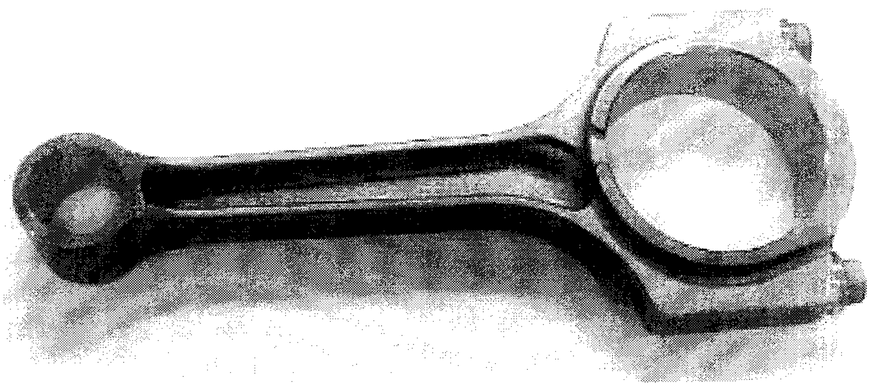


Figure 2.6 Powdered Metal Connecting Rod

waste and therefore weight. It also minimizes the amount of machine work. The caps are not cut of the rod, but precisely broken in a unique process known as "cracking". This makes for an irregular surface rather than the usual smooth machined surface of a conventional rod. The uneven surface provides a perfect fit for the cap to the rod and so no further machining of the split face is required. The powdered metal connecting rod is shown in figure 2.6.

2.2.4 Forged Steel Connecting Rod

The original-equipment forged steel rods are the one that step up the strength and reliability ladder. The forged rods begin life as bars of carbon steel that are passed through a rolling die. The rolling process compact the molecular structure and establishes a uniform, longitudinal grain flow. The bars are then heated to a plasticized state, inserted into a female die, and pressed into the near-final shape while a punch locates and knocks out the big end bore. In doing this, the grain flow at the big end is redirected in a circular pattern, like wood fibers surrounding a knot, and excellent compressive/tensile strength results. Finally the rod is put through a trimmer, the big end is severed and machined to create the cap, and bolt surfaces are spot-faced, then final machining and sizing take place.

The forged steel rods are an economical choice that should be able to handle one horsepower per cubic inch with quality fasteners, and as much as twice the factory-rated output if the beams are polished.

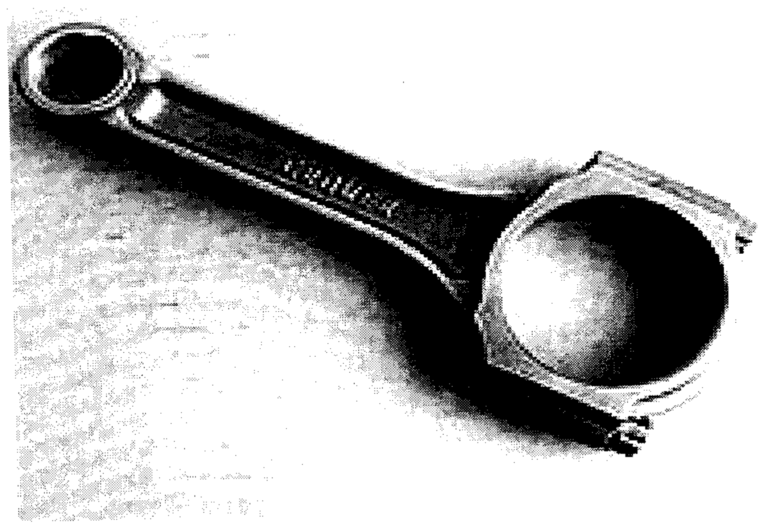


Figure 2.7 Forged Steel Connecting Rod

2.2.5 Aluminum Rod

Aluminum rods are manufactured by the forging process, or they can be cut from a sheet of aluminum plate, billet-style. Aluminum rods are generally 25-percent lighter than steel rods, and for this reason they're very popular with racers looking to shed mass from the reciprocating assembly. Lighter reciprocating parts demand less energy to set into motion, allowing more of the force of combustion to be applied to the wheels. Lower reciprocating mass also allows the engine to gain crank speed faster for quicker rpm rise after each up shift, to keep the engine near the peak of the power curve.

Aluminum rods are popular among high rpm race engines. They are very light and strong, but they a short fatigue lift. High rpm is where aluminum rods offer advantages and so if the rev is not high, there is no sense in using aluminum. They also need more pistons to head clearance due to more rod stretch; a typical aluminum rod can stretch up to 0.008" more than a steel rod in a high rpm application. Since aluminum stretches more than steel, bearing retention is also a problem; the usual tangs are not enough to be reliable. Aluminum rods must use a dowel pin to keep the bearings from spinning.

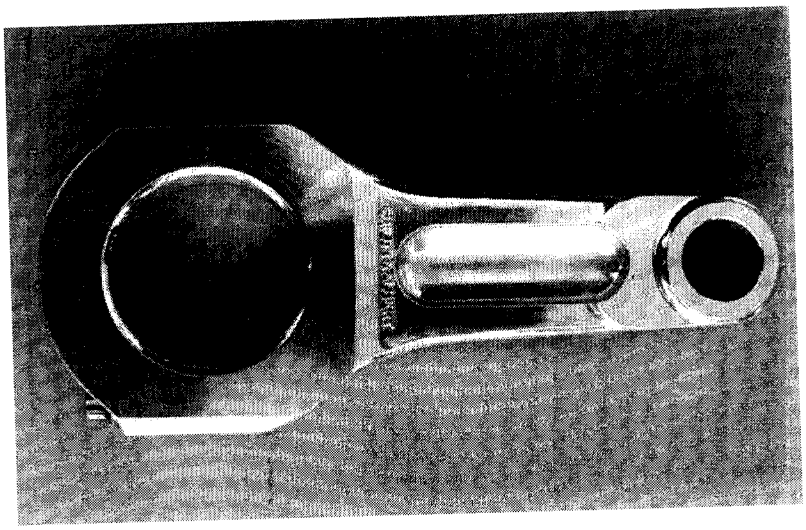


Figure 2.8 Aluminum connecting rod

2.2.6 Titanium Rod

Like steel and aluminum rods, titanium rods can be forged or cut from a billet. The titanium rods are most durable when manufactured by the forging process. This is because the grain size of even the best aerospace grade titanium is less than steel.

Titanium rods have some real advantages over steel and aluminum. Titanium has the highest strength to mass ratio of all materials used to make connecting rods. On average, a titanium rod is about 20% lighter than a comparable strength steel rod and some can be as much as 30% lighter. The downfall is cost. Titanium is a very abundant metal, but also a very difficult metal to work with, adding to the cost. The most popular titanium rods are made from titanium 6-4 alloy, which contains 6% aluminum and 4% vanadium, which are added to improve machineability. Titanium is also a very poor load bearing material. It has a tendency to gall and weld itself to anything it rubs against. This is not an issue for the big ends, since a bearing insert is used, and it is easy enough to bush the small-ends.

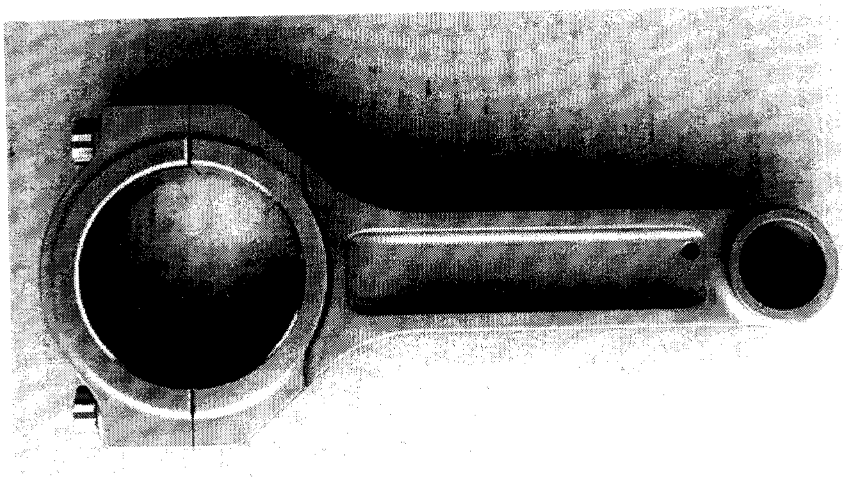


Figure 2.9 Titanium connecting rod

The connecting rods have properties based on the length (long or short) and their advantages and disadvantages based on length are listed below:

The advantages of longer connecting rod are

- Less rod angularity
- Higher wrist pin location
- Helps resist detonation
- A lighter reciprocating assembly
- Reduced piston rock
- Better leverage on the crank for a longer time
- Less ignition timing is required
- Allow slightly more compression to be used
- Before detonation is a problem
- Less average and peak piston velocity
- Peak piston velocity is later in the down stroke
- Less intake runner volume is needed

The disadvantages of longer connecting rod are

- Closer Piston-to-valve clearances
- Makes the engine run a little more cammie at low rpm
- Reduces scavenging at low rpm

The advantages of shorter connecting rod are

- Increased scavenging effect at low rpm
- Helps flow at low valve lifts (a benefit if the heads are ported with this in mind)
- Slower piston speeds near B.D.C
- Allows the intake valve to be open longer with less reversion
- More piston-to-valve clearance
- Can allow for a shorter deck height

The disadvantages of shorter connecting rod are

- More rod angularity
- Lower piston pin height (if the deck is not shorter)
- Taller and heavier pistons are required (again, if the deck height is not reduced)
- More ignition timing is required for peak power
- In general, the pistons are heavier
- More intake runner volume is needed

CHAPTER 3
COMPOSITES

3.1 COMPOSITE MATERIALS

Composites are materials that consist of two or more constituents combined in such a way that they keep their individual physical phases and do not combine to form a new chemical compound. One constituent is called reinforcing phase and the one in which the reinforcing phase is embedded is called matrix.

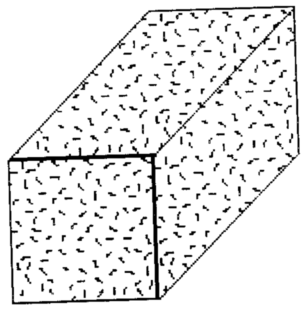


Figure 3.1 Short fiber reinforced composites

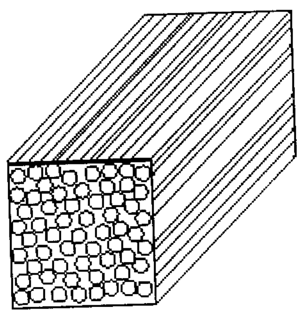


Figure 3.2 Long fiber reinforced composites

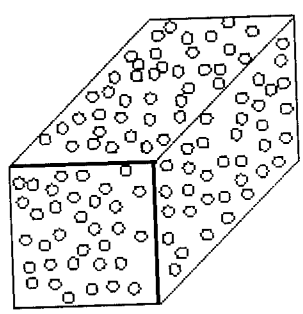


Figure 3.3 Particulate composites

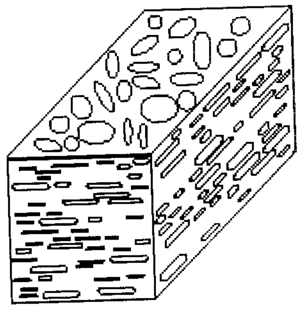


Figure 3.4 Flake composites

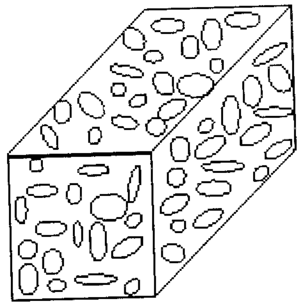


Figure 3.5 Filler composites

There are two phase in composites. They are matrix phase and reinforcement phase. Matrix is the continuous phase and surrounds the reinforcements. Reinforcement is the dispersed phase, which normally bears the majority of stress.

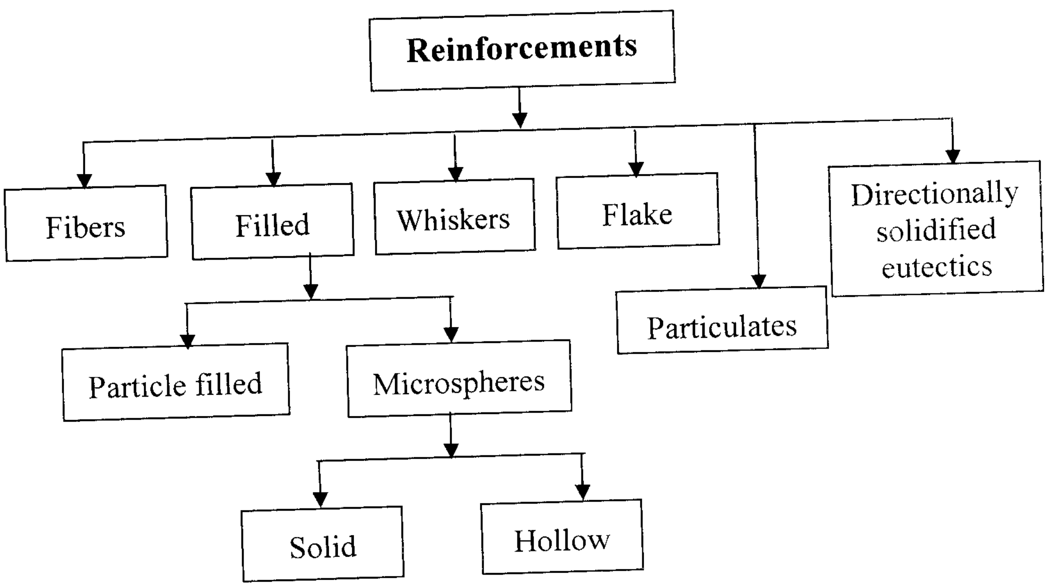


Figure 3.6 Types of Reinforcement

Reinforcement is the strong, stiff integral component which is incorporated into the matrix to achieve desired properties. Reinforcement constituent in composites provide the strength to the composites. But they also serve certain additional purposes of heat resistance or conduction, resistant to corrosion and provide rigidity. The stiffer reinforcement will usually be laid in a particular direction, within the matrix, so that the resulting material will have different properties in different directions.

Matrix is defined as a homogeneous material in which the fiber system of a composite is embedded. The functions of matrix phase are

- Binds the reinforcements (fibers/particulates) together.
- Mechanically supporting the reinforcements.
- Load transfer to the reinforcements.
- Protect the reinforcements from surface damage due to
- Abrasion or chemical attacks
- High bonding strength between fiber and matrix is important

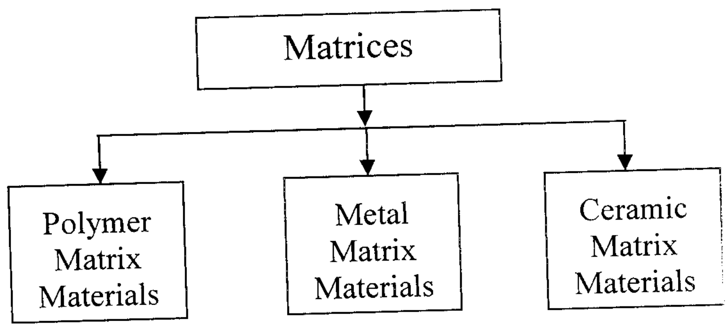


Figure 3.7 Classification of Matrices

3.1.1 Properties of Composites

Most engineering materials are essentially isotropic. That is, they have the same properties such as strength and modulus, in any direction. Most composites will have very different properties in different directions. This is because, although the matrix material is isotropic, the reinforcement is not. However, a different design procedure is required for composites compared to that required for metals. Under the static tensile stresses, the criterion of failure refers to the ultimate tensile stress based on original area that the material specimen can support.

3.1.2 Characteristics and Design Considerations

An important characteristic of MMCs, is that by appropriate selection of matrix materials, reinforcements, and layer orientations, it is possible to meet the needs of a specific design.

The wide variety of MMCs has properties that differ dramatically. Factors influencing their characteristics include:

- Reinforcement properties, form, and geometric arrangement
- Reinforcement volume fraction
- Matrix properties, including effects of porosity
- Reinforcement-matrix interface properties
- Residual stresses arising from the thermal and mechanical history of the composite
- Possible degradation of the reinforcement resulting from chemical reactions at high temperatures, and mechanical damage from processing, impact, etc.

The properties of materials reinforced with whiskers depend strongly on their orientation. Randomly oriented whiskers produce an isotropic material. Processes such as extrusion can orient whiskers, however, resulting in anisotropic properties. Whiskers also reduce ductility and fracture toughness.

MMCs reinforced with aligned fibers have anisotropic properties. They are stronger and stiffer in the direction of the fibers than perpendicular to them. The transverse strength and stiffness of unidirectional MMCs (materials having all fibers oriented parallel to one axis), however, are frequently great enough for use in components such as stiffeners and struts. This is one of the major advantages of MMCs over PMCs, which can rarely be used without transverse reinforcement. Because the modulus and strength of metal matrices are significant with respect to those of most reinforcing fibers, their contribution to composite behavior is important. The stress-strain curves of MMCs often show significant nonlinearity resulting from yielding of the matrix.

Another factor that has a significant effect on the behavior of fiber-reinforced metals is the frequently large difference in coefficient of expansion between the two constituents. This can cause large residual stresses in composites when they are subjected to significant temperature changes. In fact, during cool down from processing temperatures, matrix thermal stresses are often severe enough to cause yielding. Large residual stresses can also be produced by mechanical loading.

3.1.3 Mechanics of Composites

The relation between ply unaxial strengths and constituent properties in a structure is obtained through the composite micromechanics. Properties of the matrix, reinforcement and details of the manufacturing process are required to obtain information in impact resistance, fracture toughness, uniaxial strength etc. The results usually consist of strength and fracture properties.

Composite micromechanics in terms of mathematical models, equation and concepts are based on assumptions and principles of solid mechanics. The assumptions include: (a) Ply strength are associated with their respective fracture modes and there is a intense bond at the interface between the various constituents; (b) The ply and its constituents resist load and behave linearly elastic to fracture.

Material mechanics is used to derive the equations as it yields explicit equations of simple form for each property. By convention properties along the fiber directions are called longitudinal properties, the perpendicular ones are called transverse properties, in-plan shear is called intra-laminar shear and those through the thickness are called interlaminar properties.

Properties of ply are defined with respect to ply axis for purpose of description or analysis. Load from matrix are taken by the fiber in shear length. The shear length increases in the direction of fiber as the load gets transmitted to the fiber. The maximum load is reached when the fiber length is equal to critical length. The possibility of the composite strength reaching that of rule-of-mixtures value occurs when the length of the fiber is greater in relation to the critical length.

The constituents of sheet material could be group of fiber layers bounded together. The composite strength can be fully utilized if tension axis is in the direction of the fibers. The shear strength is weaker when the force in which the sheet is angled at 90 degrees and the tension applied perpendicular to the fiber orientation that pulls the matrix apart is weak in tension. The angle between the stress application and fiber direction is inversely proportional to composite strength and least strength is at 45°.

3.2 TYPES OF COMPOSITES

3.2.1 Polymer Matrix Composites (PMC)

Polymer matrix composites (PMCs) are materials that use a polymer based resin as a matrix material with some form of fibres embedded in the matrix, as reinforcement. Both thermosetting and thermoplastic polymers can be used for the matrix material. Common polymer composite thermosetting matrix materials include polyester, vinyl ester and epoxy. The reinforcements include glass, carbon and aramid fibres. Polymer composites are chosen for the manufacture of particular articles or components, because of weight saving for their relative stiffness and strength. It plays a role in carrying transverse and shear loading. Transfer load between fibers. And also play a vital role in longitudinal tensile strength, as indicated by the experimental observation that the strength of matrix-impregnated fiber bundles could be double of that of a dry fiber bundle.

The modern development of polymeric materials and high modulus fibres (carbon, aramidic) introduced a new generation of composites. The most relevant benefit has been the possibility of energetically convenient manufacturing associated with the low weight features. Polymeric composites were mainly developed for aerospace applications where the reduction of the weight was the principal objective, irrespective of the cost. The effort to produce economic attractive composite components has resulted in several innovative manufacturing techniques currently being used in the composite industry.

3.2.2 Metal Matrix Composites (MMC)

Metal matrix composites offer higher strength, fracture toughness and stiffness than those offered by their polymer counterparts. They can withstand elevated temperatures in corrosive environment than composites. Most metals and alloys could be used as matrices and they require reinforcement materials which need to be stable over a range of temperature and non reactive too. The reinforcement may improve specific strength/stiffness, abrasion resistance, creep resistance, thermal conductivity and dimensional stability of the overall composites. The advantages over PMC are higher operating temperatures, non-flammability and greater resistance to degradation by organic fluids.

3.2.3 Ceramic Matrix Composites (CMC)

Ceramic matrix composites have been developed to overcome the intrinsic brittleness and lack of reliability of monolithic ceramics, with a view to introduce ceramics in structural parts used in severe environments, such as rocket and jet engines, gas turbines for power plants, heat shields for space vehicles, fusion reactor first wall, aircraft brakes, heat treatment furnaces, etc.

A given ceramic matrix can be reinforced with either discontinuous reinforcements, such as particles, whiskers or chopped fibres, or with continuous fibres. There is a wide spectrum of CMCs depending on the chemical composition of the matrix and reinforcement.

Ceramics can be described as solid materials which exhibit very strong ionic bonding in general and in a few cases covalent bonding. High melting points, good corrosion resistance, stability at elevated temperatures and high compressive strength, make ceramic based matrix materials a favorite for applications requiring a structural material that does not give way at temperatures above 1500°C. The advantages are high strength and modulus, very high service temperature and reduced weight (lower fuel consumption). Its use in high and severe stress applications, e.g. automobile and aircraft gas turbine engines.

3.3 METAL MATRIX COMPOSITES (MMC)

A metal matrix composite system is defined as composite system which contains two or more discrete insoluble phases.

3.3.1 MMC Reinforcements

MMC reinforcements can be divided into five major categories: continuous fibers, discontinuous fibers, whiskers, particulates, and wires. MMCs can be reinforced either by the three dimensional shapes (particulate), two dimensional (lamina) or one dimensional shapes (fibrous).

The role of the reinforcement depends upon its type in structural MMCs. In particulate and whisker reinforced MMCs, the matrix is the major load bearing constituent. The role of the reinforcement is to strengthen and stiffen the composite through prevention of matrix deformation by mechanical restraint. This restraint is generally a function of the ratio of interparticle spacing to particle diameter. In continuous fiber reinforced MMCs, the reinforcement is the principal load-bearing constituent. The metallic matrix serves to hold the reinforcing fibers together and transfer as well as distribute the load. Discontinuous fiber reinforced MMCs display characteristics between those of continuous fiber and particulate reinforced composites.

Typically, the addition of reinforcement increases the strength, stiffness and temperature capability while reducing the thermal expansion coefficient of the resulting MMC. When combined with a metallic matrix of higher density, the reinforcement also serves to reduce the density of the composite, thus enhancing properties such as specific strength.

The points to be noted in selecting the reinforcements include compatibility with matrix material, thermal stability, density, melting temperature etc.

3.3.2 MMC Matrices

Light metals form the matrix for low temperature application. Only light metals are responsive, with their low density proving an advantage. Titanium aluminium and magnesium are the popular matrix metals currently in usage, which are particularly useful for aircraft applications. Aluminium and Magnesium MMCs are used more in automotive industry. Though the costs are very high, these materials are amenable. Connecting rods has been successfully manufactured on a mass scale, from fiber reinforced metals. They improve fuel efficiency because of weight reduction.

The choice of a matrix alloy for an MMC is decided by several considerations. The particular importance is of whether the composite is to be continuously or discontinuously reinforced. The use of continuous fibers as reinforcements may result in transfer of most of the load to the reinforcing filaments and hence composite strength will be governed primarily by the fiber strength. The primary roles of the matrix alloys are to provide efficient transfer of load to the fibers and to blunt cracks in the event that fiber failure occurs and so the matrix alloy for a continuously reinforced MMC may be chosen more for toughness than for strength. On this basis, lower strength, more ductile, and tougher matrix alloys may be utilized in continuously reinforced MMCs.

For discontinuously reinforced MMCs, the matrix may govern composite strength. Then, the choice of matrix will be influenced by consideration of the required composite strength and higher strength matrix alloys may be required. Additional considerations in the choice of the matrix include potential reinforcement/matrix reactions, either during processing or in service that might result in degraded composite performance; thermal stresses due to thermal expansion mismatch between the reinforcements and the matrix; and the influence of matrix fatigue behavior on the cyclic response of the composite.

Aluminium and magnesium alloys are regarded as widely used matrices due low density and high thermal conductivity. MMCs with low matrix alloying additions result in attractive combinations of ductility, toughness and strength.

The interface between the matrix and the reinforcement is important as the properties of this region dictate load transfer and crack resistance of MMCs during deformation. The interactions between the matrix and reinforcement may be in the form of mechanical locking or chemical bonding.

The types of matrix materials used in metal matrix composites are Aluminium, Titanium, Magnesium, and Copper alloys and Superalloys. The most important MMC systems are:

- Aluminium matrix
    - Continuous fibers: boron, silicon carbide, alumina, graphite
    - Discontinuous fibers: alumina, alumina-silica
    - Whiskers: silicon carbide
    - Particulates: silicon carbide, boron carbide
- Magnesium matrix
    - Continuous fibers: graphite, alumina
    - Whiskers: silicon carbide
    - Particulates: silicon carbide, boron carbide
- Titanium matrix
    - Continuous fibers: silicon carbide, coated boron
    - Particulates: titanium carbide

- Copper matrix
  - Continuous fibers: graphite, silicon carbide
  - Wires: niobium-titanium, niobium-tin
  - Particulates: silicon carbide, boron carbide, titanium carbide
- Superalloy matrices
  - Wires: tungsten

3.3.3 Characteristics of MMC

The significant characteristic of metal matrices is their ability to retain the wrought metal properties which can be altered to mechanical and physical specifications. This characteristic is important in high modulus composites where loading and interfacial properties should be consistent and reproducible in addition to matrix alloy properties.

The reinforcement mostly being linear elastic solid does not exhibit good impact resistance. Hence, the high toughness and impact properties of metal alloys are very much important. The ductile matrix also allows stress concentrations by plastic deformations, which improves the material's fracture toughness property.

The composite materials consist of two or more unlike phases. These two phases have been in coordination with one another in order to promote an efficient bonding of the constituents. In metal matrix composites, the physical compatibility problems of reinforcing a metal matrix with a filament are related to the material constants associating dilatation to pressure (stress) or thermal changes. The chemical compatibility problems relate to are interfacial chemical reactions, environmental chemical reactions and interfacial bonding during processing of the composite.

The need for physical compatibility demands that the matrix should have required ductility and strength so as to transmit the structural loads laid on it to reinforcing member uniformly. Mechanical properties of the matrix should comprise high ductility and compliance to enable its use various applications. The matrix being ductile is required to have higher coefficient of thermal expansion.

3.3.4 Advantages of MMC

1. MMCs have higher strength-to-density ratios, higher stiffness-to-density ratios, better fatigue resistance, better elevated temperature properties,

lower creep rate, lower coefficients of thermal expansion, better wear resistance.

2. MMCs have high strength, fracture toughness, high modulus, thermal stability, and ductility and enhanced elevated temperature performance.
3. MMCs are lightweight, and exhibit good stiffness and low specific weight. It is generally considered that these materials offer savings in weight, at the same time maintaining their properties.
4. They do not yield to changes in temperature or thermal shock. They can be easily fabricated, shaped, joined and given a smooth finish.
5. The advantages of MMCs over polymer matrix composites are higher temperature capability, fire resistance, higher transverse stiffness and strength, no moisture absorption, higher electrical and thermal conductivities, better radiation resistance, no outgassing.

MMCs are also used in automotive industry, aerospace industry, biomedical, consumer products, electronics industry sport equipment industry etc.

3.4 COMPOSITE CONNECTING ROD

A composite connecting rod is comprised of aluminium metal matrix with silicon carbide (reinforcing material). The connecting rod has a split crankshaft-receiving end, a wrist pin-receiving end and an elongated intermediate connecting member extending between crankshaft-receiving end and wrist pin-receiving end.

The composite connecting rod has a greater strength-to-weight ratio than metal, is flame resistant, and is stable to heat. The composite engine part is capable of effectively functioning at engine operating temperatures and start-up conditions during hot and cold weather. The composite connecting rod has high mechanical strength, thermal stability, fatigue strength, and excellent tensile, compressive, and flexural strength. The composite connecting rod is resistant to wear, corrosion, impact, rupture, and creep, and reliably operates in the presence of engine fuels, oils, and exhaust gases.

Composite connecting rods decrease secondary harmonic vibrations, fluttering, and engine shaking, and enhance more efficient combustion temperatures. Advantageously, the composite connecting rod maintains its shape and structural integrity at engine operating conditions. The lightweight composite engine part decreases fuel consumption, attenuates noise for quieter performance, and permits increased speed of operation. The lightweight composite engine part produces higher horsepower for its weight than conventional engine parts, while maintaining its shape, dimensional stability, and structural integrity at engine operating conditions. The lightweight composite connecting rod decreases centrifugal, reciprocating, and inertial forces on the engine.

3.5 MATERIAL SELECTION

The connecting rod used in automobiles requires high strength in both tension and compression. It also requires high fatigue strength. For the engine to perform efficiently the weight also plays an important role. Most slender rod designs must have high stiffness, good strength and consistent weight to facilitate engine balancing.

In this work Aluminium is chosen as a matrix material due to its light weight and reinforced with silicon carbide.

TABLE 3.1 MATERIAL PROPERTIES OF ALUMINIUM/ SILICON CARBIDE COMPOSITES MATERIALS

S.No.	Property	Silicon Carbide/ Aluminium SCS2/6061 A1
1.	Fiber volume Ratio, V_f	0.43
2.	Density (p, g/cm^3)	2.85
3.	Longitudinal Modulus (E_1,GPa)	204
4.	Transverse Modulus (E_2,GPa)	118
5.	In-plane shear modulus (G_{12},GPa)	41
6.	Major Poisson's Ratio (v_{12})	0.27
7.	Minor Poisson's Ratio (v_{21})	0.12
8.	Longitudinal Tensile strength (F_{1t}, MPa)	1462
9.	Traverse Tensile strength (F_{2t}, MPa)	86
10.	In-plane shear strength (F_6, MPa)	113

11.	Ultimate Longitudinal tensile strain ($\in_{1t}^{u}$)	0.009
12.	Ultimate Traverse Longitudinal tensile strain ($\in_{2t}^{u}$)	0.001
13.	Longitudinal Compressive strength (F_{1e}, MPa)	2,990
14.	Traverse Compressive strength (F_{2c}, MPa)	285
15.	Longitudinal Thermal Expansion Coefficient, (α_1, 10^{-6} /°C [10^{-6}/°F])	9.1 (5.0)
16.	Traverse Thermal Expansion Coefficient, (α_2, 10^{-6} /°C [10^{-6}/°F])	17.8 (9.9)
17.	Longitudinal Moisture Expansion Coefficient (β_1)	0
18.	Traverse Moisture Expansion Coefficient (β_2)	0

3.6 MANUFACTURING METHODS

Fabrication is an important part of the design process for all structural materials, including MMCs.

Current methods can be divided into two major categories, primary and secondary. Primary fabrication methods are used to create the MMC from its constituents. The resulting material may be in a form that is close to the desired final configuration, or it may require considerable additional processing, called secondary fabrication, such as forming, rolling, metallurgical bonding, and machining. The processes used depend on the type of reinforcement and matrix.

A critical consideration is reactions that can occur between reinforcements and matrices during primary and secondary processing at the high temperatures required to melt and form metals. These impose limitations on the kinds of constituents that can be combined by the various processes. Sometimes, barrier coatings can be successfully applied to reinforcements, allowing them to be combined with matrices that otherwise would be too reactive.

Cast MMCs now consistently offer net or net-net shape, improved stiffness and strength, and compatibility with conventional manufacturing techniques. They are also consistently lower in cost than those produced by other methods, are available from a wide range of fabricators, and offer dimensional stability in both large and small parts.

At the current time, the most common method used to make graphite/aluminium and graphite/magnesium composites are by infiltration.

Graphite yarn is first passed through a furnace to burn off any sizing that may have been applied. Next it goes through a CVD process that applies a coating of titanium and boron which promotes wetting by the matrix. Then it immediately passes through a bath or fountain of molten metal, producing an infiltrated bundle of fibers known as a "wire." Plates and other structural shapes are produced in a secondary operation by placing the wires between foils and pressing them, as is done with monofilaments. Recent development of "air stable" coatings permits use of other infiltration processes, such as casting, eliminating the need for "wires" as an intermediate step.

3.6.1 Stir Casting

In stir casting, molten metal is stirred with the help of either a mechanical stirrer or high intensity ultrasonic treatment. This action disperses the reinforcing phase, which is added to the surface of the melt in the molten metal, and solidifies the composite melt which contains reinforcements suspended in the melt. The thermodynamics and kinetics of transfer of particle or fibrous reinforcements from the gas phase, to the liquid phase through the oxide film, and finally from the liquid to the solid phase has been worked on by the author and his colleagues to design the processes to make different Cast Metal Matrix Composites. Stir mixing and Casting is now used for large-scale production of Metal Matrix Particulate Composites. Various metals such as Al, Mg, Ni, and Cu have been used as the matrix and a wide variety of reinforcements like SiC, graphite, SiO_2, Al_2O_3 Si_3N_4, and $ZrSiO_4$, have been used as reinforcements.

The study of cast composites has added considerable understanding to the solidification of conventional monolithic castings, especially issues related to the effects of inclusions on the fluidity, viscosity, nucleation, growth, particle settling and particle pushing. Processing of metal matrix composites by stir mixing and casting requires special precautions including temperature control and design of pouring and gating systems.

3.6.2 Infiltration Process

In the infiltration technique, liquid metal is infiltrated through the narrow crevices between the fibers or particulate reinforcements which are arranged in a preform, fixed in space unlike the stir mixing and casting process where the reinforcements are free to float or settle in the melt due to density differences. As the liquid metal enters between the fibers or particles during infiltration, it cools and then solidifies, producing a composite. In general, the infiltration technique is divided into three distinct operations: these are the perform preparation (during this operation reinforcement elements are assembled together into a porous body), the infiltration process (during which the liquid metal enters within the preform), and the solidification of liquid metal (in this operation, liquid to solid transformation takes place).

The formation of the composite through solidification during non-isothermal infiltration of a fiber preform where the fiber temperature is initially lower than the temperature of the liquid. In this situation, a transient layer of solidified metal can form as soon as the liquid metal comes in contact with the cold fiber. Melt infiltration can be achieved with the help of mechanical pressure, inert gas pressure or vacuum. Recently techniques of pressure less infiltration have also been developed.

CHAPTER 4

DYNAMIC LOAD ANALYSIS OF CONNECTING ROD

4.1 ENGINE CONFIGURATION

1.	Piston diameter	= 88mm
2.	Compression ratio	= 12 : 1
3.	Stroke	= 125 mm
4.	R.P.M	= 5700 rev/min (maximum)
5.	Radius of Crank	= 62.5 mm
6.	Factor of Safety	= 3 to 6
7.	l/r ratio	= 4.5 (Range of l/r = 4 to 5)
8.	Maximum gas pressure	= 37.3 bar
9.	Mass of connecting rod	= 0.439 kg
10.	Mass of reciprocating parts	= 1.6 kg
11.	Mass of piston assembly	= 0.434 kg
12.	Length of the connecting rod (centre to centre)	= 300mm
13.	Rankine's Formula	
	(i) Numerator constant	= 320 N/mm^2
	(ii) Denominator	= 1/ 7500

Engine make : Kirloskar diesel engine

4.2 KINEMATIC ANALYSIS OF SLIDER CRANK MECHANISM

For a 2D mechanism such as a slider crank mechanism, the forces are only in the plane of motion. Forces in Z direction will be zero. There will also be no moments since there are pin joints at both the ends of the connecting rod.

To understand the kinematics of connecting rod, consider a slider crank mechanism as shown in the figure 4.1. Let OC be the crank and PC be the connecting rod. The crank rotates with angular velocity of ω radians/sec and crank turns through an angle θ from the inner dead centre. Let x be the displacement of a reciprocating body from the inner dead centre after time t seconds during which the crank had turned through an angle θ.

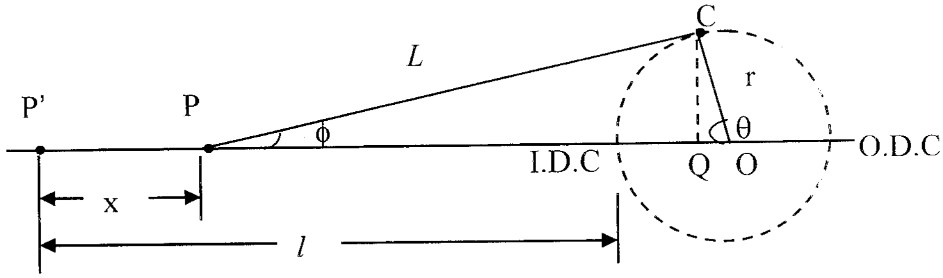


Figure 4.1 Vector representation of slider crank mechanism.

The velocity and acceleration of internal combustion engine are determined by analytical method.

Velocity of the piston

$$v_{PO} = v_P = \omega.r\left(\sin\theta + \frac{\sin 2\theta}{2n}\right)$$

Acceleration of the piston

$$a_P = \omega^2.r\left(\cos\theta + \frac{\cos 2\theta}{2n}\right)$$

Angular velocity of connecting rod

$$\omega_{PC} = \frac{\omega \mathrm{Cos}\theta}{(n^2 - \sin^2\theta)^{1/2}}$$

Acceleration of connecting rod

$$\alpha_{PC} = \frac{-\omega^2 \sin\theta(n^2 - 1)}{(n^2 - \sin^2\theta)^{3/2}}$$

The negative sign shows that the sense of the acceleration of the connecting rod is such, that it tends to reduce the angle ϕ.

The angular velocity and angular acceleration is calculated for three different engine speeds. Tables 4.1, 4.2 and 4.3 show the angular velocity and angular acceleration at 2000, 4000 and 5700 rev/min respectively. The graphs are plotted for crank angle Vs angular velocity and crank angle Vs angular acceleration at three different speeds.

TABLE 4.1 ANGULAR VELOCITY AND ANGULAR ACCELERATION AT 2000 REV/MIN

Crank Angle	Angular velocity	Angular acceleration
0	46.54	0
10	45.93	-1612.57
20	43.86	-3196.89
30	40.55	-4720.05
40	36.02	-6142.84
50	30.36	-7418.15
60	23.718	-8492.5
70	16.27	-9308.97
80	8.282	-9822.74
90	0	-9997.9
100	-8.282	-9822.74
110	-16.27	-9308.97
120	-23.718	-8492.5
130	-30.36	-7418.15
140	-36.02	-6142.84
150	-40.55	-4720.05
160	-43.86	-3196.89
170	-45.93	-1612.57
180	-46.54	0
190	-45.93	1612.57
200	-43.86	3196.89
210	-40.55	4720.05
220	-36.02	6142.84
230	-30.36	7418.15
240	-23.718	8492.5
250	-16.27	9308.97
260	-8.282	9822.74
270	0	9997.9
280	8.282	9822.74
290	16.27	9308.97
300	23.718	8492.5
310	30.36	7418.15
320	36.02	6142.84
330	40.55	4720.05
340	43.86	3196.89
350	45.93	1612.57
360	46.54	0

TABLE 4.2 ANGULAR VELOCITY AND ANGULAR ACCELERATION AT 4000 REV/MIN.

Crank Angle	Angular velocity	Angular acceleration
0	93.08	0
10	91.73	-6450.59
20	87.72	-12788.18
30	81.11	-18881.1
40	72.04	-24572.31
50	60.71	-29673.03
60	47.42	-33971.65
70	32.47	-37237.58
80	16.56	-39289.21
90	0	-39993.51
100	-16.56	-39289.21
110	-32.47	-37237.58
120	-47.42	-33968.1
130	-60.71	-29673.03
140	-72.04	-24572.31
150	-81.11	-18881.1
160	-87.72	-12788.18
170	-91.73	-6450.59
180	-93.08	0
190	-91.73	6450.59
200	-87.72	12788.18
210	-81.11	18881.1
220	-72.04	24572.31
230	-60.71	29673.03
240	-47.42	33968.1
250	-32.47	37237.58
260	-16.56	39289.21
270	0	39993.51
280	16.56	39289.21
290	32.47	37237.58
300	47.42	33971.65
310	60.71	29673.03
320	72.04	24572.31
330	81.11	18881.1
340	87.72	12788.18
350	91.73	6450.59
360	93.08	0

TABLE 4.3 ANGULAR VELOCITY AND ANGULAR ACCELERATION AT 5700 REV/MIN.

Crank Angle	Angular velocity	Angular acceleration
0	132.44	0
10	130.52	-13059.5
20	124.81	-25890.64
30	115.41	-38226.22
40	102.5	-49748.5
50	86.39	-60075.29
60	67.48	-68770.46
70	46.31	-75390.31
80	23.56	-79544.8
90	0	-80961.24
100	-23.56	-79543.99
110	-46.31	-75390.31
120	-67.48	-68770.46
130	-86.39	-60075.29
140	-102.5	-49748.5
150	-115.41	-38226.22
160	-124.81	-25890.64
170	-130.52	-13059.5
180	-132.44	0
190	-130.52	13059.5
200	-124.81	25890.64
210	-115.41	38226.22
220	-102.5	49748.5
230	-86.39	60075.29
240	-67.48	68770.46
250	-46.31	75390.31
260	-23.56	79543.99
270	0	80961.24
280	23.56	79544.8
290	46.31	75390.31
300	67.48	68770.46
310	86.39	60075.29
320	102.5	49748.5
330	115.41	38226.22
340	124.81	25890.64
350	130.52	13059.5
360	132.44	0

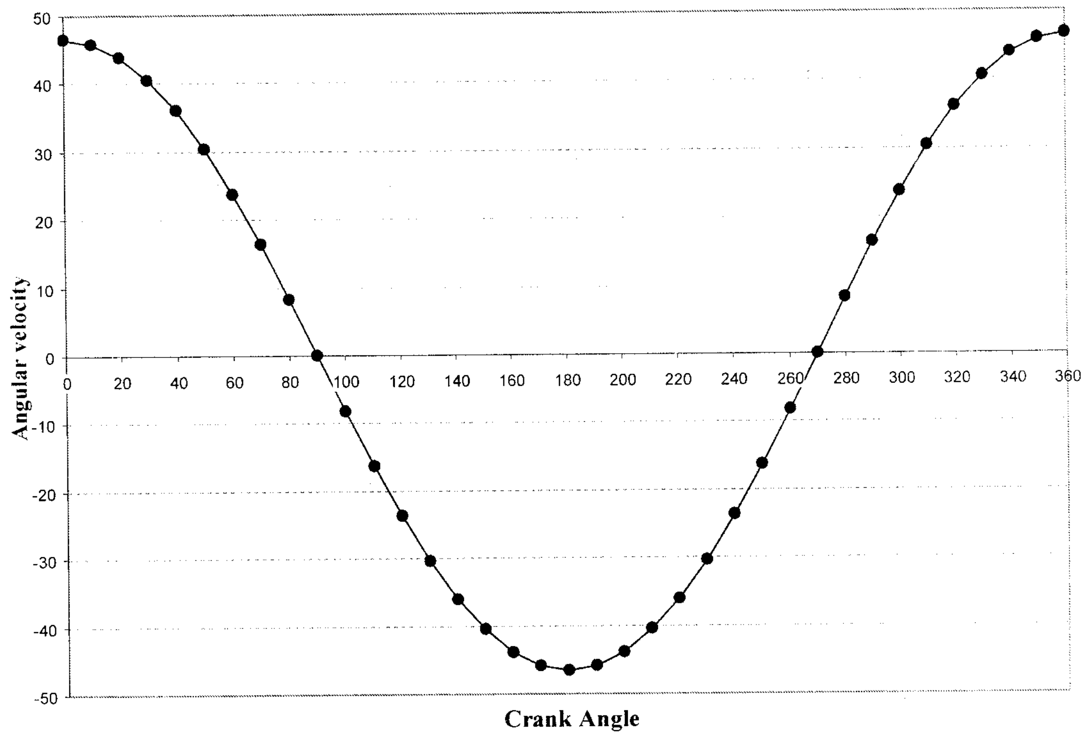


Figure 4.2 Variation of angular velocity with crank angle at speed 2000 rev/min

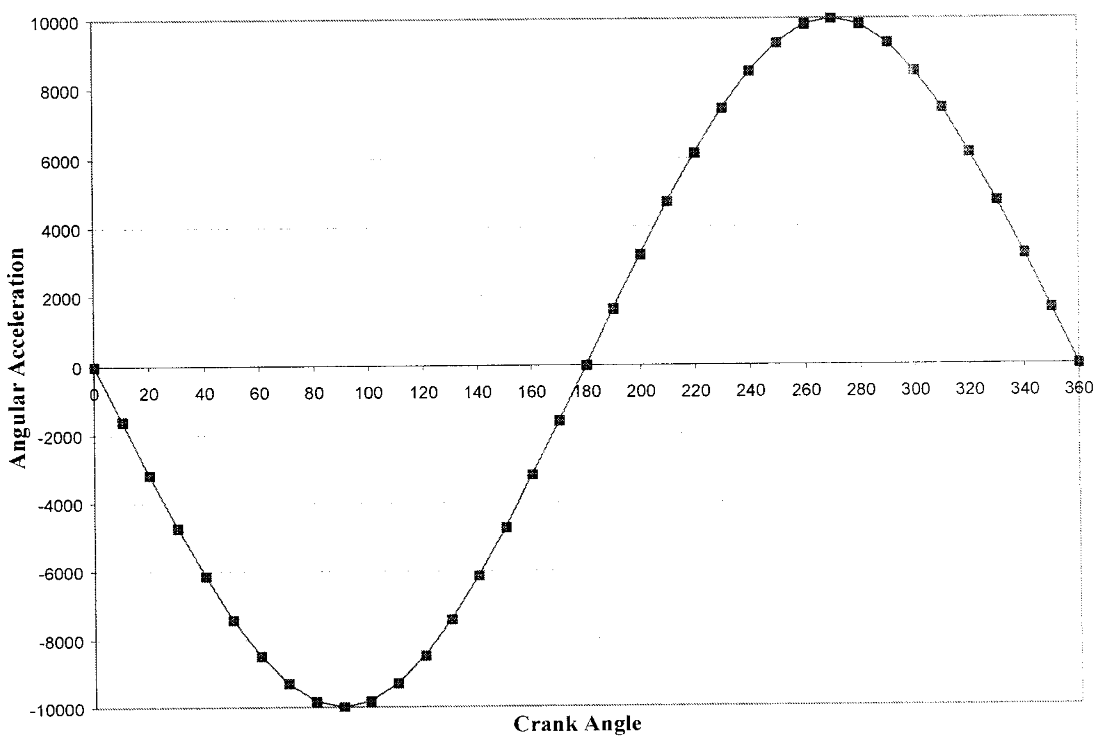


Figure 4.3 Variation of angular acceleration with crank angle at speed 2000 rev/min

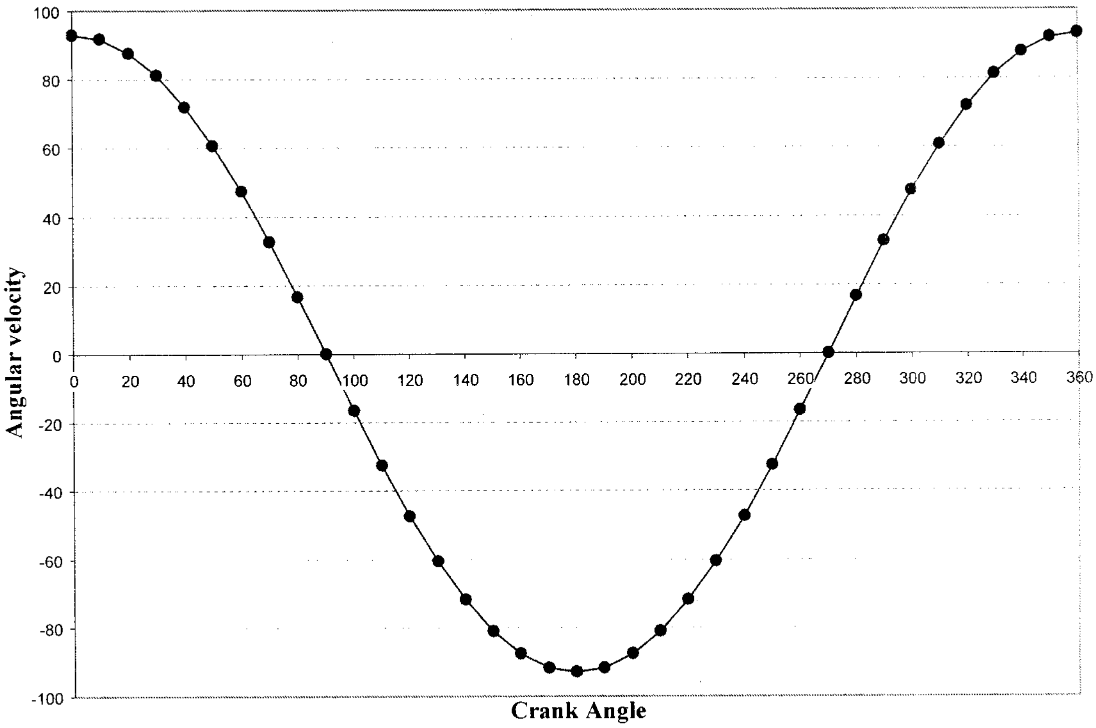


Figure 4.4 Variation of angular velocity with crank angle at speed 4000 rev/min

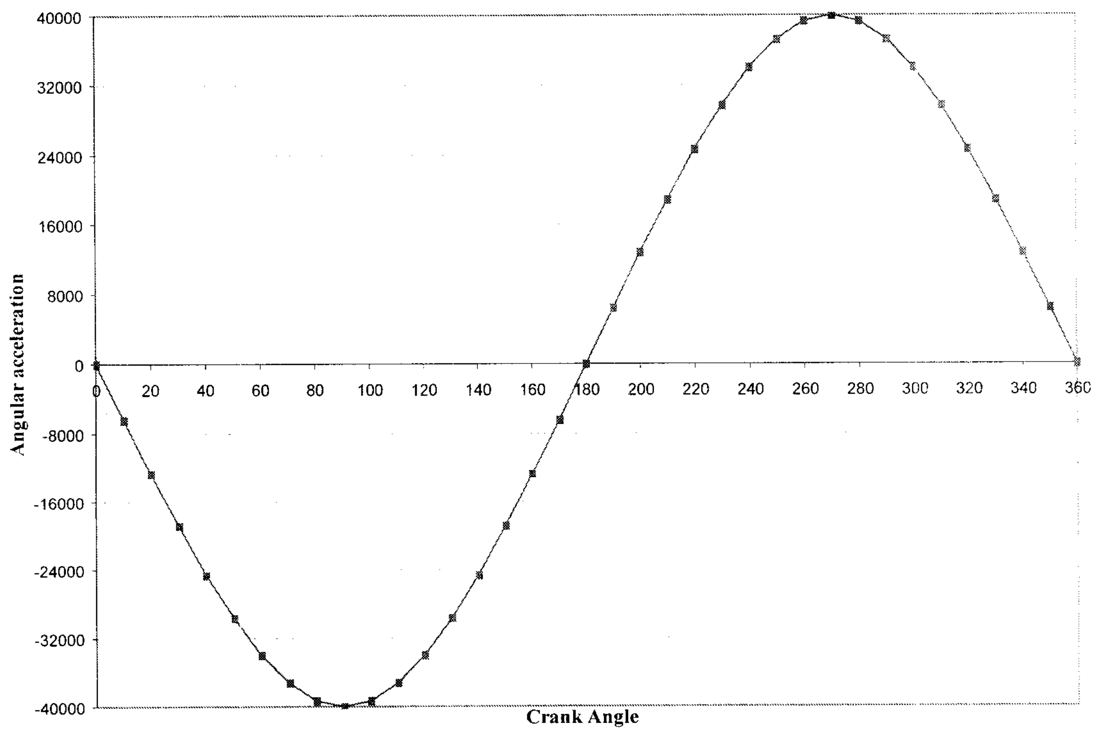


Figure 4.5 Variation of angular acceleration with crank angle at speed 4000 rev/min

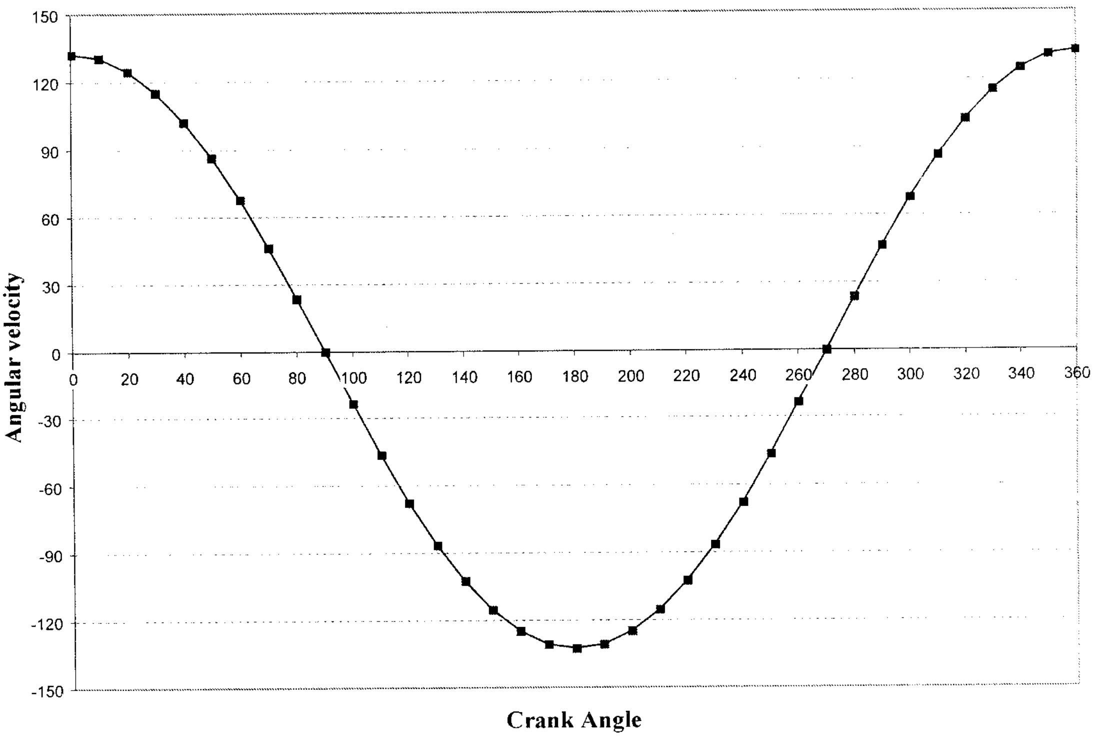


Figure 4.6 Variation of angular velocity with crank angle at speed 5700 rev/min

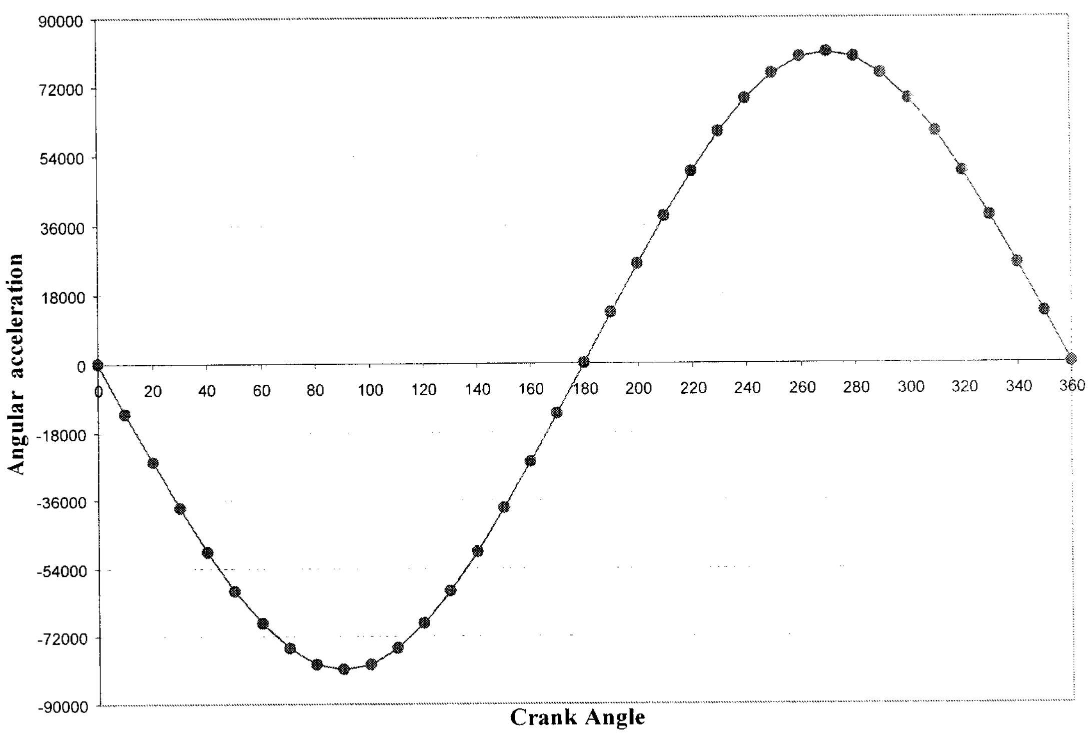


Figure 4.7 Variation of angular acceleration with crank angle at speed 5700 rev/min

4.3 DESIGN OF CONNECTING ROD

The connecting rod is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile forces, the cross-section of the connecting rod is designed as a strut.

The connecting rod is considered like both ends hinged for buckling about X-axis and both ends fixed for buckling about Y-axis. The connecting rod should be equally strong in buckling about both the axes.

According to Rankine's formula,

$$\text{WB about X-axis} = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} \quad [\because \text{For both ends hinged, } L = l]$$

and

$$\text{WB about X-axis} = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{yy}}\right)^2} = \frac{\sigma_c . A}{1 + a\left(\frac{L}{k_{yy}}\right)^2} \quad [\because \text{For both ends hinged, } L = \frac{l}{2}]$$

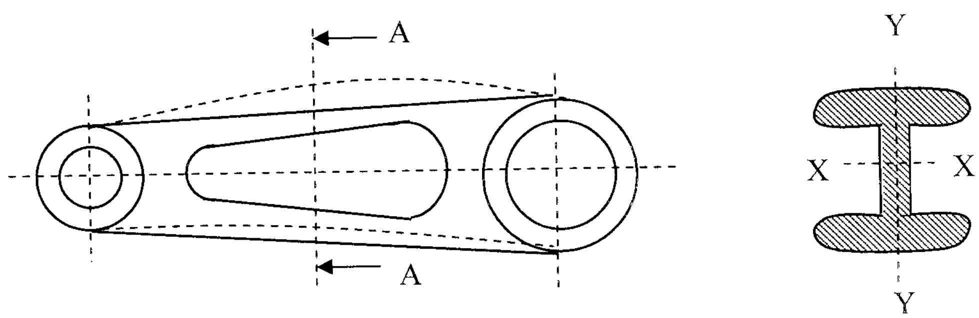


Figure 4.8 Buckling of connecting rod

In order to have a connecting rod equally strong in buckling about the axes, the buckling loads must be equal, i.e.,

$$\frac{\sigma_c . A}{1 + a\left(\frac{l}{k_{xx}}\right)^2} = \frac{\sigma_c . A}{1 + a\left(\frac{l}{2k_{yy}}\right)^2} \quad \text{or} \quad \left(\frac{l}{k_{xx}}\right)^2 = \left(\frac{l}{2k_{yy}}\right)^2$$

$$\therefore \quad k_{xx}^2 = 4k_{yy}^2 \quad \text{or} \quad I_{xx} = I_{yy} \qquad (\because I = A . k^2)$$

This shows that the connecting rod is four times strong in buckling about Y- axis than about X-axis.

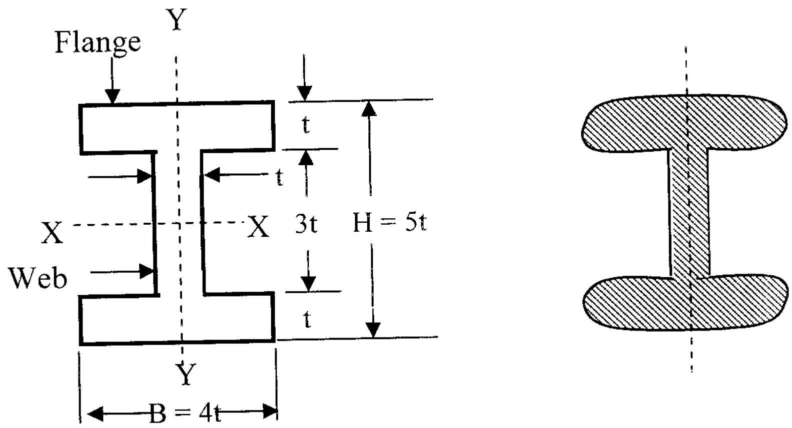


Figure 4.9 I-section of connecting rod

The most suitable section for the connecting rod is I-section with the proportion as shown in Figure 4.9 because of its lightness and to keep the inertia forces as low as possible especially in case of high speed engines. It can also withstand high gas pressure.

Let Thickness of flange and web of section $= t$

Width of the section, $B = 4t$ and

Depth or Height of the section, $H = 5t$

From figure 4.9,

Area of the section, $A = 2\,(4\,t \times t) + 3\,t \times t = 11\,t^2$

Moment of inertia of the section about X-axis,

$$I_{xx} = \frac{1}{12}\left[4t(5t)^3 - 3t(3t)^3\right] = \frac{419}{12}t^4$$

and moment of inertia of the section about Y-axis,

$$I_{yy} = \left[2 \times \frac{1}{12}t \times (4t)^3 + \frac{1}{12}(3t)t^3\right] = \frac{131}{12}t^4$$

$$\because \quad \frac{I_{xx}}{I_{yy}} = \frac{419}{12} \times \frac{12}{131} = 3.2$$

Since the value of $\frac{I_{xx}}{I_{yy}}$ lies between 3 and 3.5, therefore I-section chosen is quite satisfactory. The connecting rod is designed by taking the force on the connecting rod (F_C) equal to the maximum force on the piston (F_L) due to gas pressure.

$$\therefore F_C = F_L = \frac{\pi D^2}{4} \times p$$

$$= \frac{3.14(88)^2 \times 37.3}{4}$$

$$= 226863.20 \text{ N}$$

The dimensions are determined by considering the buckling of the rod about x-axis (assuming both ends hinged) and applying the Rankine's formula.

The buckling load (W_B) may be calculated by using the following relation, i.e.,

$$W_B = \text{Max. gas force} \times \text{Factor of safety}$$

The recommended factor of safety for the connecting rod is 3 to 6.

$$\text{Buckling load} = \text{Force on connecting rod} \times \text{Factor of safety}$$

$$= 226863.20 \times 4.3$$

$$= 975511.76 \text{ N}$$

The radius of gyration of the section about x-axis,

$$K_{xx} = \sqrt{I_{xx}/A} = 1.78t$$

Length of the connecting rod = 300mm

Equivalent length of the connecting rod for both ends hinged, L= l = 300mm.

According to Rankine's formula, buckling load

$$W_B = \frac{\sigma_c.A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} \qquad \text{......(A)}$$

$$975511.76 = \frac{\sigma_c.A}{1 + a\left(\frac{L}{k_{xx}}\right)^2} = \frac{320 \times 11t^2}{1 + \frac{1}{7500}\left(\frac{300}{1.78t}\right)^2}$$

By solving the above equation A,

Thickness of I-section = 6mm

Width of the section = 24mm

Depth of the section = 30mm

The dimension B = 4t and H = 5t, as obtained above by applying the Rankine's formula, at the middle of the connecting rod. The width of the section (B) is kept constant throughout the length of the connecting rod, but the depth or height varies. The depth near the small end (or piston end) is taken as H_1 = 0.75H to 0.9H and the depth near the big end (or crank end) is taken H_2 = 1.1H to 1.25H.

Depth near the small end = 0.8 x30 =24mm.

Depth near the big end = 1.2 x30 =36mm.

4.3.1 Validation of I-Section

The I-section of the connecting rod is validated using finite element analysis. The 2D beam element is used to model the shank portion of the connecting rod and the properties of the I-section are given. The results are shown in the figure 4.10 and 4.11.

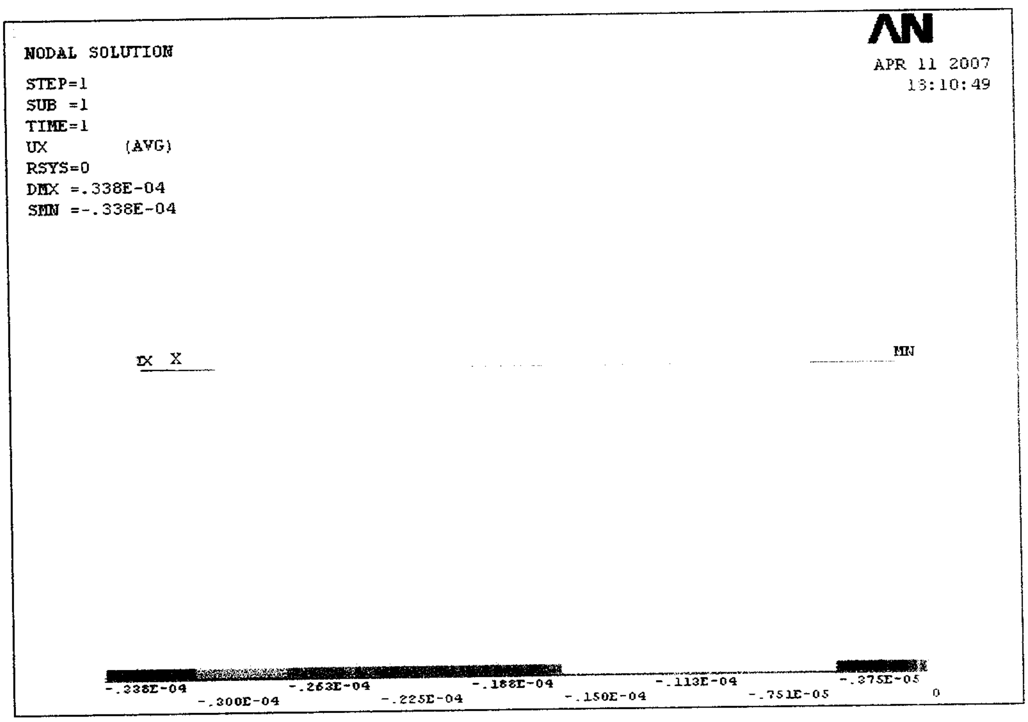


Figure 4.10 Displacement in X-direction

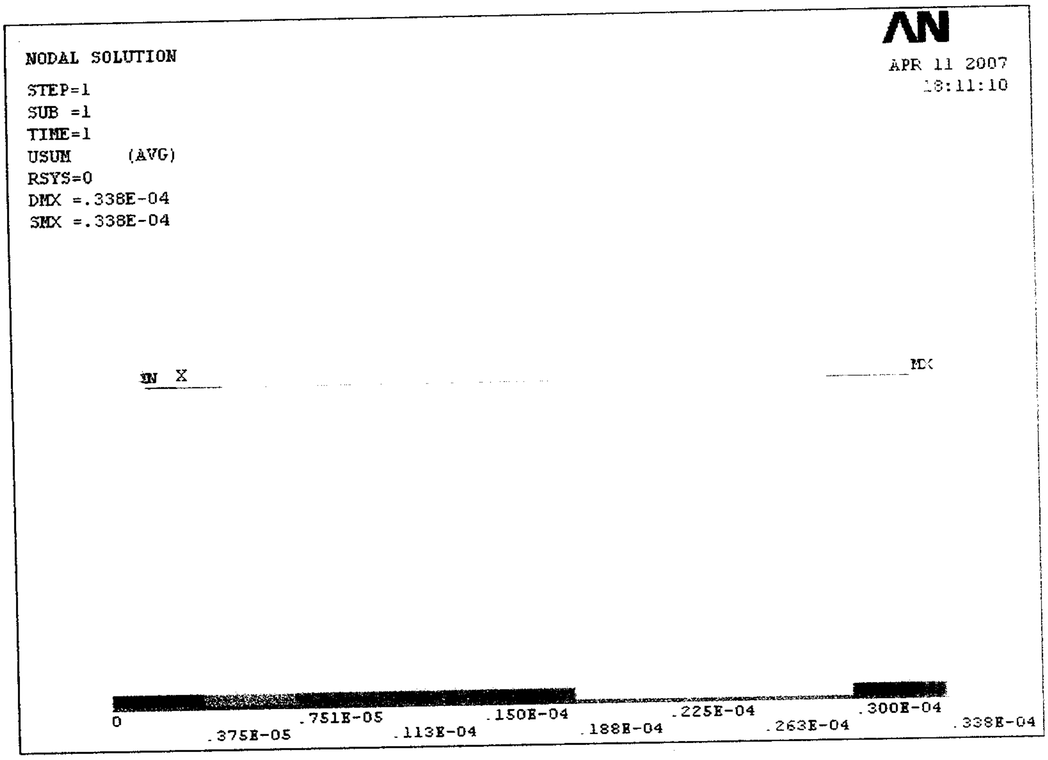


Figure 4.11 Total displacement

4.4 FORCES ACTING ON THE CONNECTING ROD

The connecting rod is subjected to a complex state of loading. It undergoes high cyclic loads of the order of 10^8 to 10^9 cycles, which range from high compressive loads due to combustion, to high tensile loads due to inertia. Therefore it is essential to determine the magnitude of the loads acting on the connecting rod. A proper picture of the stress variation during a loading cycle is essential and this will require FEA over the entire engine cycle.

Once the components of forces at the connecting rod ends in the X and Y directions are obtained, they can be resolved into components along the connecting rod length and normal to it. The components of the inertia load acting at the center of gravity can also be resolved into similar components. It is neither efficient nor necessary to perform FEA of the connecting rod over the entire cycle and for each and every crank angle. Therefore, a few positions of the crank were selected depending upon the magnitudes of the forces acting on the connecting rod, at which FEA was performed.

The stress at a point on the connecting rod as it undergoes a cycle consists of two components, the bending stress component and the axial stress component. The bending stress depends on the bending moment, which is a function of the load at the C.G. normal to the connecting rod axis, as well as angular acceleration and linear acceleration component normal to the connecting rod axis.

The axial loads at the crank end and at the piston pin end are not generally identical at any point in time. They differ due to the inertia load acting on the connecting rod. The load at either end could be used as a basis for deciding points at which to perform FEA. The load at the crank end was used in this work.

The various forces acting on the connecting rod are as follows:

1. Force on the piston due to gas pressure and inertia of the reciprocating parts,
2. Force due to inertia of the connecting rod or inertia bending forces,
3. Force due to friction of the piston rings and of the piston, and
4. Force due to friction of the piston pin bearing and the crankpin bearing.

The expressions for the forces acting on the vertical engine are discussed below.

1. Force on the piston due to gas pressure or inertia of reciprocating parts

   Consider a connecting rod PC as shown in the figure 4.12.

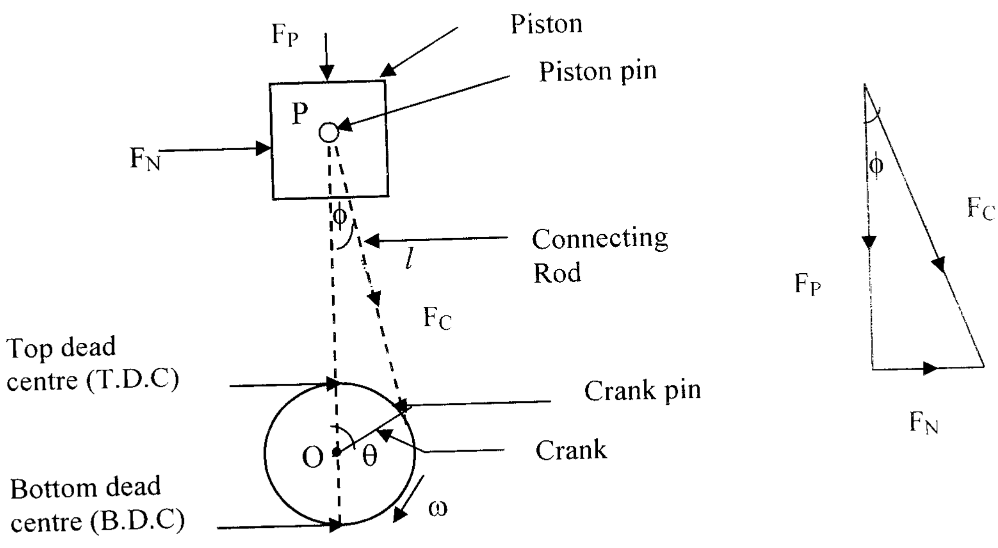


Figure 4.12 Forces on the Connecting Rod

The force on the piston due to pressure of gas,

$$F_L = \text{Pressure} \times \text{Area} = p.\ A = p \times \pi\ D^2/4$$

Inertia force of the reciprocating parts,

$$F_I = \text{Mass} \times \text{Acceleration} = m_R\ .\ \omega^2.\ r\left(\text{Cos}\theta + \frac{\cos 2\theta}{n}\right)$$

The inertia force of reciprocating parts opposes the force on the piston when it moves during its downward stroke (i.e., when the piston moves from the top dead centre to bottom dead centre). On the other hand, the inertia force of the reciprocating parts helps the force on the piston when it moves from the bottom dead centre to top dead centre.

Therefore, net force acting on piston or piston pin (or gudgeon pin or wrist pin),

$$F_P = \text{Force due to gas pressure} \mp \text{Inertia force}$$

$$= F_L \mp F_I$$

When weight of the reciprocating parts is to be taken into consideration, then

$$F_P = F_L \mp F_I \mp W_R$$

The force F_P gives rise to a force F_C in the connecting rod and a thrust F_N on the sides of the cylinder walls.

$$F_C = \frac{F_P}{\cos\varphi} = \frac{F_P}{\sqrt{1 - \frac{\sin^2\theta}{n^2}}}$$

The force in the connecting rod will be maximum when the crank and the connecting rod are perpendicular to each other (i.e., $\theta = 90°$). But at this position, the gas pressure would be decreased considerably. Therefore in designing connecting rods the force in the connecting rod (F_C) is taken equal to the maximum force on the piston due to pressure of gas (F_L), neglecting piston inertia effects.

2. Force due to inertia of the connecting rod or inertia bending forces.

Considering a connecting rod PC and a crank OC rotating with uniform angular velocity ω rad/s. The inertia forces will be opposite to the direction of acceleration are centrifugal forces. The inertia forces can be resolved into two components, one parallel to the connecting rod and the other perpendicular to the rod. The parallel (or longitudinal) components add up algebraically to the force acting on the connecting rod (F_C) and produce thrust on the pins. The perpendicular components produces bending action also called whipping action and the stress induced in the connecting rod is called whipping stress.

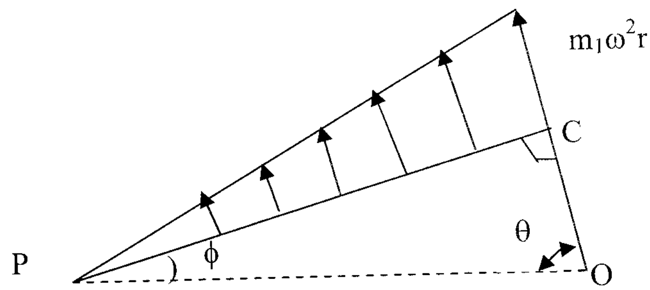


Figure 4.13 Inertia bending force

It may be noted that the perpendicular components will be the maximum, when the crank and the connecting rod are at right angle to each other.

The maximum bending stress, due to inertia of connecting rod is

$$\sigma_{max} = \frac{M_{max}}{Z}$$

where Z = Section modulus.

From the above it can be seen that the maximum bending moment varies as the square of speed, therefore, the bending stress due to high speed will be dangerous. It may be noted that the maximum axial force and the maximum bending stress do not occur simultaneously. In an I.C engine, the maximum gas load occurs close to top dead centre where as the bending stress occurs when the crank angle θ = 65° to 70° from top dead centre. The pressure of gas falls suddenly as the piston moves from dead centre. Thus the general practice is to design a connecting rod by assuming the force in the connecting rod (F_C) equal to the maximum force due to pressure (F_L), neglecting pressure inertia effects and then checked for bending stress due to inertia force (i.e., whipping stress).

3. Force due to friction of piston rings and of the piston.

The friction force (F) of the piston rings may be determined by using the following expression:

$$F = \pi D_1 \cdot t_R \cdot n_R \cdot p_R \cdot \mu$$

Since the frictional force of the piston rings usually very small, therefore, it may be neglected.

4. Force due to friction of the piston pin bearing and crank pin bearing.

The force due to friction of the piston pin bearing and crankpin bearing is to bend the connecting rod and to increase the compressive stress on the connecting rod due to the direct load. Thus, the maximum compressive stress in the connecting rod will be

$\sigma_{c(max)}$ = Direct compressive stress + maximum bending or Whipping stress due to inertia bending stress

4.5 CALCULATION OF FORCES

Forces acting on the connecting rod and the piston are shown in Figure 4.14 and 4.15. The force is calculated at crank angles where their magnitudes are high. The crank angles are 180 and 360 for maximum engine speeds. Neglecting the effect of friction and of gravity, equations to obtain these forces are listed below. Note that mp is the mass of the piston assembly and m is the mass of the connecting rod. Forces at the piston pin and crank ends in X and Y directions are given by:

$$F_{BX} = -(m_p a_P + \text{Gas Load})$$

$$F_{AX} = m_c a_{c.gX} - F_{BX}$$

$$F_{BY} = [m_c a_{c.gY} u \cos\beta - m_c a_{c.gX} u \sin\beta + I_{zz} a_2 + F_{BX} l \sin\beta] / (l \cos\beta)$$

$$F_{AY} = m_c a_{c.gY} - F_{BY}$$

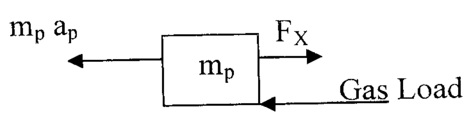


Figure 4.14 Free body diagram of Piston

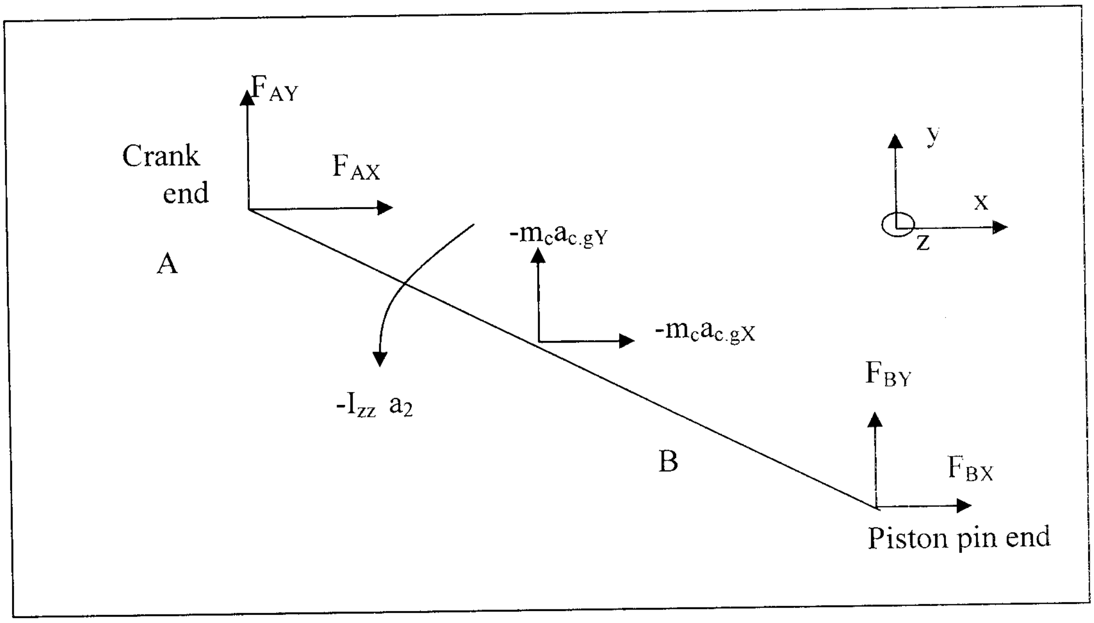


Figure 4.15 Free body diagram of connecting rod

CHAPTER 5

FINITE ELEMENT ANALYSIS

5.1 DESCRIPTION OF FINITE ELEMENT METHOD

In the finite element method (FEM), the actual continuum or body of matter like solid, liquid or gas is represented as an assemblage of sub divisions called finite elements. These elements are considered to be interconnected at specified joints called nodes or nodal point. The nodes usually lie on the element boundaries where adjacent elements are considered to be connected. Since the actual variation of the field variable inside a finite element can be approximated by a simple function, these approximating functions (also called interpolation models) are defined in terms of the values of the field variables at the nodes. When field equations (like equilibrium equations) for the whole continuum are written, the new unknowns will be the nodal values of the field variable. By solving the field equations, which are generally in the form of matrix equation, the nodal value of the field variable will be known. Once these are known the approximating functions define the field variable throughout the assemblage of elements.

The solution of a general continuum problem by the finite element method always follows an orderly step-by-step process. With reference to static structural problems, the step-by-step procedure can be stated as follows:

Step (i) Discretization of the structure

The first step in the finite element method is to divide a structure or solution region into subdivisions or elements. Hence the structure is to be modeled with suitable finite elements. The number, type, size and arrangement of the elements are to be decided.

Step (ii) Selection of a proper interpolation or displacement model

Since the displacement solution of a complex structure under any specified load conditions can be predicted exactly, assume some suitable solution within an element to approximate the unknown solution. The assumed solution must be simple from a computational point of view, but it should satisfy certain convergence requirements. In general, the solution or the interpolation model is taken in the form of a polynomial.

Step (iii) Derivation of element Stiffness matrices and load vectors

From the assumed displacement model, the stiffness matrix $[K^{(e)}]$ and the load vector $\vec{P}^{(e)}$, of element "e" are to be derived by using either equilibrium conditions or a suitable variational principle

Step (iv) Assemblage of element equations to obtain the overall equilibrium equations. Since the structure is composed of several finite elements, the individual element stiffness matrices and load vectors are to be assembled in a suitable manner and the overall equilibrium equations have to be formulated as

$$[\underline{K}]\vec{\underline{\Phi}} = \vec{\underline{P}}$$

where $[\underline{K}]$ is called the assembled stiffness matrix, $\vec{\underline{\Phi}}$ is the vector of nodal displacement and $\vec{\underline{P}}$ is the vector of nodal forces for the complete structure.

Step (v) Solution for the unknown nodal displacements

The overall equilibrium equations have to be modified to account for the boundary conditions of the problem. After the incorporation of the boundary conditions, the equilibrium equations can be expressed as

$$[K]\vec{\Phi} = \vec{P}$$

For linear problems, the vector $\vec{\Phi}$ can be solved very easily for the non-linear problems the solution has to be obtained in a sequence of steps, each step involving the modification of the stiffness matrix [K] and/or the load vector $\vec{P}$.

Step (vi): Computation of element strains and stresses

From the known nodal displacement $\vec{\Phi}$ if required, element strains and stresses can be computed by using the necessary equations of solid or structural mechanics.

5.2 ELEMENT SELECTION AND DESCRIPTION

The shapes, sizes, number and configurations of the elements have to be chosen carefully such that the original body or domain is simulated as closely as possible without increasing the computational effort needed for the solution.

Mostly the choice of the element is decided by the geometry of the body and the number of properties and the field variable of the problem.

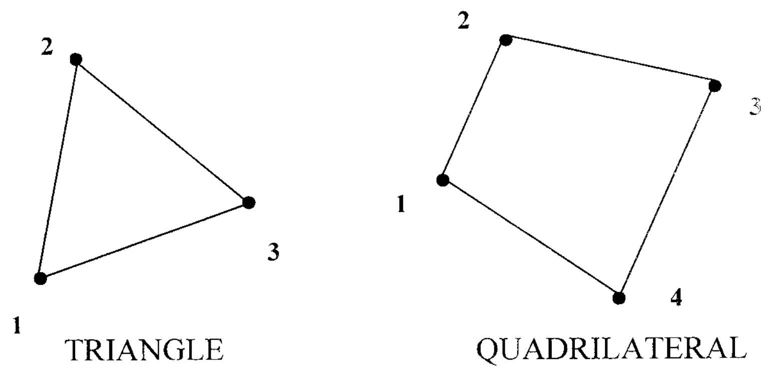


Figure 5.1 Two-dimensional Elements

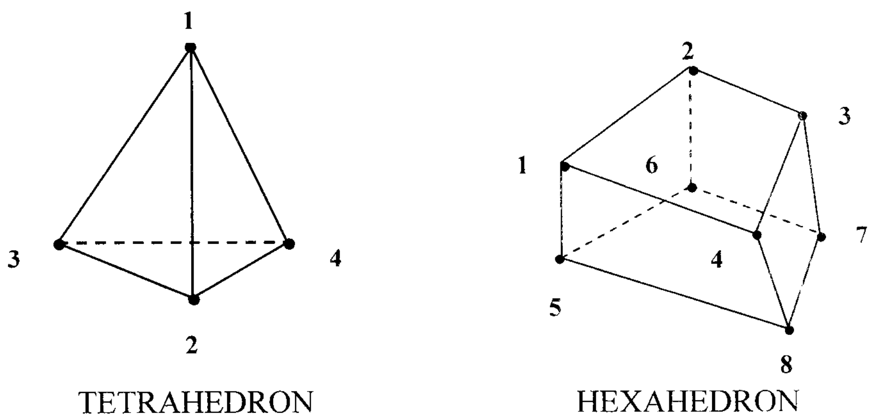


Figure 5.2 Three-dimensional Elements

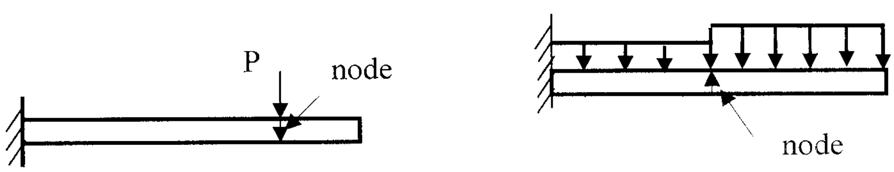
The two-dimensional elements are used for two independent spatial coordinates. The basic element useful for two dimensional analysis is the triangular element. Although a quadrilateral element can be obtained by assembling two or four triangular elements, in some cases the use of quadrilateral proves to be advantageous.

If the geometry, material properties and the other parameters of the body can be described by three independent spatial coordinates and idealize the body by using the three dimensional elements shown in figure 5.2. The basic three dimensional elements analogous to the triangular element in the case of two-dimensional problems, is the tetrahedron elements. In some cases the hexahedron element, which can be obtained by assembling five tetrahedrons can be used advantageously. Some problems, which are actually three-dimensional, can be described by only one or two independent coordinates.

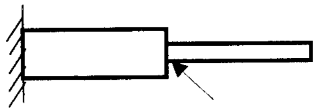
For the discretization of problems involving curved geometries, finite elements with curved sides are useful. The typical elements are having curved boundaries. The ability to model curved boundaries has been made possible by the addition of midside nodes. Finite elements with straight sides are known as linear elements, while those with curved sides are called higher order elements.

The size of element influences the convergence of solution directly and hence it has to be chosen with care. If the size of the element is small, the final solution is expected to be more accurate. However the use of elements of small size will increase the computational time. In general, whenever steep gradients of field variable are expected, finer mesh is used. Another characteristics related to the size of elements which affects the finite element solution is the aspect ratio of the elements. The aspect ratio describes the shape of the element in the assemblage of elements. For two-dimensional elements, the aspect ratio is the ratio of the largest dimension of the element to the smallest dimension. Elements with an aspect ratio of nearly unity generally yield best results.

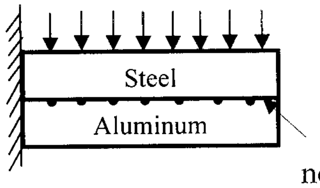
If the body has no abrupt change in geometry, material properties and external conditions (like load, temperature, etc), the body can be divided into equal subdivisions and hence spacing of nodes can be uniform. If there are any discontinuities in the problem, nodes have to be introduced at these discontinuities. The load is concentrated on a beam and abrupt change in the distributed load is shown in figure 5.3. The discontinuity in geometry and in material properties are shown in figure 5.4 and 5.5 respectively.



5.3 Discontinuity in loading



5.4 Discontinuity in geometry



5.5 Discontinuity in material properties

The number of elements to be chosen for idealization is related to the accuracy desired, size of elements, and the degrees of freedom involved. An increase in the number of elements generally means more accurate results, for any given problem, there will be a certain number of elements beyond which the accuracy can be improved by any significant amount.

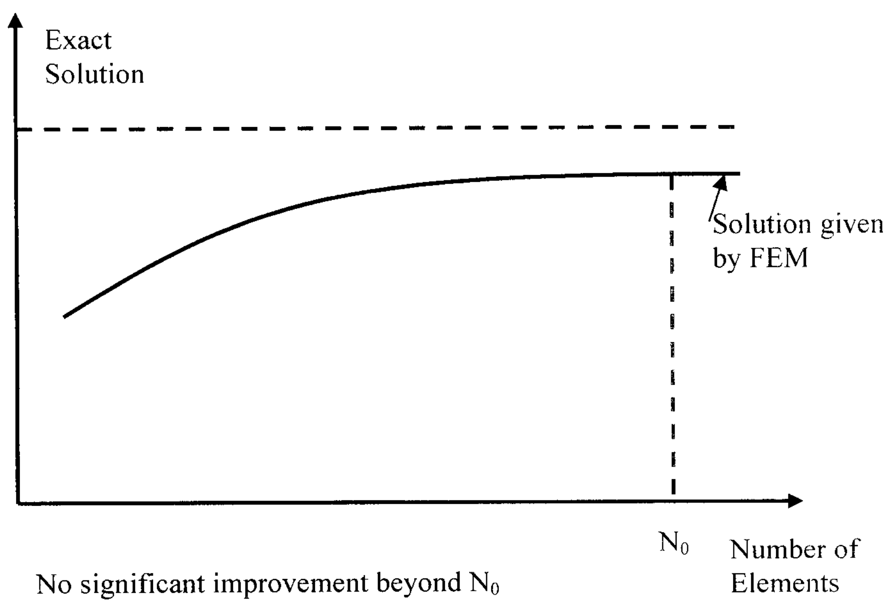


Figure 5.6 Effect of varying the number of elements

5.3 FINITE ELEMENT ANALYSIS (FEA)

FEA consists of a computer model of a material or design that is loaded and analyzed for specific results. It is used in new product design, and existing product refinement. Modifying an existing product or structure is utilized to qualify the product or structure for a new service condition. In case of structural failure, FEA may be used to help determine the design modifications to meet the new condition.

Mathematically, the structure to be analyzed is subdivided into a mesh of finite sized elements of simple shape. Within each element, the variation of displacement is assumed to be determined by simple polynomial shape functions and nodal displacements. Equations for the strains and stresses are developed in terms of the unknown nodal displacements. From this, the equations of equilibrium are assembled in a matrix form which can be easily be programmed and solved on a computer. After applying the appropriate boundary conditions, the nodal displacements are found by solving the matrix stiffness equation. Once the nodal displacements are known, element stresses and strains can be calculated.

Within each of these modeling schemes, the programmer can insert numerous algorithms (functions) which may make the system behave linearly or non-linearly. Linear systems are far less complex and generally ignore many subtleties of model loading and behaviour. Non-linear systems can account for more realistic behaviour such as plastic deformation, changing loads etc. and is capable of testing a component all the way to failure.

5.3.1 Difference between FEM and FEA

The finite element method is a mathematical method for solving ordinary and elliptic partial differential equations via a piecewise polynomial interpolation scheme. In simple words, FEM evaluates a differential equation curve by using a number of polynomial curves to follow the shape of the underlying and more complex differential equation curve. Each polynomial in the solution can be represented by a number of points and so FEM evaluates the solution at the points only. A linear polynomial requires 2 points, while a quadratic requires 3. The points are known as node points or nodes.

There are essentially three mathematical ways that FEM can evaluate the values at the nodes, there is the non-variational method, the residual method and the variational method.

FEA is an implementation of FEM to solve a certain type of problem.

5.3.2 Applications of FEA

Finite element analysis was first developed for use in the aerospace and nuclear industries where the safety of structures is critical. Today, the growth in usage of the method is directly attributable to the rapid advances in computer technology in recent years. As a result, commercial finite element packages exist that are capable of solving the most sophisticated problems such as in structural analysis.

- The finite element is a mathematical method for solving ordinary and partial differential equations. Because it is a numerical method, it has the ability to solve complex problems that can be represented in differential equation form. As these types of equations occur naturally in virtually all fields of the physical sciences, the applications of the finite element method are limitless as regards the solution of practical design problems.
- FEA has been used almost universally to solve structural engineering problems. One discipline that has relied heavily on the technology is the aerospace industry. Due to the extreme demands for faster, stronger, lighter and more efficient aircrafts, manufacturers have to rely on the technique to stay competitive.
- FEA is capable of solving the most sophisticated problems in steady state and dynamic temperature distributions, fluid flow and manufacturing processes such as injection molding and metal forming.
- It is essential to have an understanding of the technique and physical processes involved in the analysis. Only then can an appropriate and accurate analysis model be selected, correctly defined and subsequently interpreted.
- FEA has been used routinely in high volume production and manufacturing industries for many years.

- The finite element method is a very important tool for those involved in engineering design, it is now used routinely to solve problems in the following areas:
  - Structural strength design
  - Structural interaction with fluid flows
  - Analysis of Shock (underwater & in materials)
  - Acoustics
  - Thermal analysis
  - Vibrations
  - Fluid flows
  - Electrical analysis
  - Mass diffusion
  - Buckling problems
  - Dynamic analysis
  - Electromagnetic evaluations
  - Metal forming

5.3.3 Fluid Analysis

FEA facilitates the prediction of fluid flow, heat & mass transfer, and chemical reactions (explosions) and related phenomena. By solving the fundamental equations governing fluid flow processes, FE analysis provide information on important flow characteristics such as pressure loss, flow distribution, and mixing rates. This results in better designs, lower risk, and faster time to the market place for product or processes. Models can be developed for physical phenomena such as turbulence, multiphase flow, chemical reactions, and radiative heat transfer.

Due to the complex nature of the physical processes being modelled, it is not unusual to conduct coupled analyses as part of a design program. Fluid-structural, fluid-thermal & fluid-acoustic analyses are not uncommon.

5.3.4 Stress and Displacement Analysis

The most common application of FEA is the solution of stress related design problems. As a result, all commercial packages have an extensive range of stress analysis capabilities. Stress can be described as a measurement of intensity of force. If this intensity increases beyond a limit known as yield, the component's material will undergo a permanent change in shape or may even be subjected to dramatic failure. From a formal point of view, three conditions have to be met in any stress analysis, equilibrium of forces (or stresses), compatibility of displacements and satisfaction of the state of stress at continuum boundaries. These conditions, which are usually described mathematically in good undergraduate strength of material texts, are also applicable to non-linear analysis.

5.3.5 Vibration Analysis

Vibration usually becomes a concern when it's amplitudes grow large enough to cause either excessive stress, or if it disturbs the people in, on or near the vibrating object(s). There are many items of equipment (balances, microscopes, cameras, transmission equipment etc.) that are very sensitive to vibration.

Modal analysis is important in machines where there is likely to be cyclic out of balance forces, such as in rotating machinery (engines, electric & pneumatic motors, generators, industrial equipment, etc.) and fluid flow applications (due to alternating vortex shedding). The chief aim of any vibration analysis is to ensure that the system is not subject to a dangerous resonant condition during the range of operation. A point to note is that although the response of the system is time dependant, any excitation will be harmonic and the solution may be obtained using the Eigenvalue approach. It is important to note that many applications fall in a category beyond this range, and full dynamic analysis are required.

5.3.6 Acoustics Analysis

Noise in the acoustic sense is usually produced by vibration of a body or is caused by instabilities in fluid flow (such as turbulence). An acoustic analysis enables the prediction of the way sound radiates inside and outside of a product, simultaneously or separately. The problem is oftentimes viewed from three different viewpoints, sound generation, sound propagation and structural interaction. It is usual to perform the analysis in the frequency domain.

In an analysis, boundary conditions include velocities, pressures, impedances (relationship between pressure & velocity), and acoustic monopole and plane wave sources. Velocity and impedance can usually be combined into a simultaneous velocity and impedance type boundary conditions. The acoustic investigations usually determine pressure, intensity, acoustic velocity, and radiated power for the domain of interest.

5.3.7 Creep and Fatigue Analysis

Stress-life fatigue procedures may be applied to the analysis of structures or products to determine cycles to failure. Strain-life fatigue procedures may be applied to the analysis of structures or products to determine cycles to crack initiation. Linear elastic fracture mechanics may be applied to the analysis of structures or products to determine the life expectancy from crack initiation to final fracture.

The interaction of creep & stress rupture with cyclic stressing and the fatigue process is not yet understood, but is of great importance in many high performance engineering applications. Creep is effectively a pre-failure of the material. Creep may occur under the following conditions. Cyclic or repeated loading can cause failure at lower stresses than static loading. This aspect is central to fatigue performance. Fatigue can be described as a progressive failure phenomenon that proceeds by the initiation and propagation of cracks to an unstable size.

5.3.8 Electromagnetic Analysis

Many kinds of electromagnetic phenomenon can be modeled from the propagation of microwaves to the torque in an electric motor. Analysis of electrostatic and magnetic fields passing through and around a structure provides insight into the response, and hence a means for regulating these fields to attain specific responses. FEA can be used to analyze the linear electric or magnetic behaviour of devices. Analysis typically involves the evaluation of magnetic, electric and thermal fields. Further applications include the analysis of shape-memory materials & piezoelectric effects. An analysis can be static. harmonic or transient state in nature. Due to the complexity of the practical applications of the technique, it is not unusual to have magnetic, dielectric and thermal couplings in a single model.

5.3.9 Temperature Analysis

Thermal analysis is used to determine the temperature distribution, heat accumulation or dissipation, and other related thermal quantities in an object. The nodal degrees of freedom (primary unknown data) are the temperatures. The primary heat transfer mechanisms are conduction, convection and radiation. In addition, less dominant phenomena such as change of phase (melting or freezing) & internal heat generation can occur.

5.3.10 Transient Analysis

Transient dynamic analysis determines the time-response history of a structure subjected to a forced displacement function. The structure may behave linearly, or in some cases, friction, plasticity, large deflections or gaps may produce nonlinear behavior. Once the time response history is known, complete deflection and stress information can be obtained for specific times.

The first step in any dynamic analysis should be the determination of the frequencies and shapes of the natural vibration modes. In a 3-D structure there are three dynamic degrees of freedom for every unrestrained node with non-zero mass and there is potentially a natural vibration mode for each degree of freedom. Thus, there are usually many potential vibration modes in a typical structure, but usually only a small number of vibration modes with the lowest frequencies that are of interest.

5.4 ANSYS SOFTWARE

ANSYS (Analysis of Engineering Systems) is a complete FEA simulation software package developed by ANSYS Inc USA. ANSYS is a finite-element analysis package used widely in industry to simulate the response of a physical system to structural loading, and thermal and electromagnetic effects. ANSYS uses the finite-element method to solve the underlying governing equations and the associated problem-specific boundary conditions.

ANSYS is the original (and commonly used) name for ANSYS Mechanical or ANSYS Multiphysics, general-purpose finite element analysis software. ANSYS, Inc. actually develops a complete range of CAE products but is perhaps best known for ANSYS Mechanical and ANSYS Multiphysics.

ANSYS Mechanical and ANSYS Multiphysics are self contained analysis tools incorporating pre-processing (geometry creation, meshing), solver and post processing modules in a unified graphical user interface. ANSYS is a general purpose finite element modeling package for numerically solving a wide variety of mechanical problems. These problems include: static/dynamic structural analysis, heat transfer and fluid problems, as well as acoustic and electro-magnetic problems.

The software is used to analyze a broad range of applications. It includes solvers for thermal, structural, CFD, electromagnetics, acoustics and can couple these separate physics together in order to address multi-disciplinary applications. The program has a comprehensive graphical user interface (GUI) that gives users easy, interactive access to program functions, commands, documentation, and reference material. An intuitive menu system helps users navigate through the program. Users can input data using a mouse, a keyboard, or a combination of both.

There are three stages in Finite element analysis

i) Preprocessing
ii) Analysis
iii) Postprocessing

i) The major steps in preprocessing are

- Define keypoints/lines/areas/volumes
- Define element type and material/geometric properties
- Mesh lines/areas/volumes are required

ii) The major steps in analysis are

- Assigning loads (point or pressure)
- Constraints (Translational and rotational)
- Solving the resulting set of equation.

iii) The major steps in postprocessing

- List of nodal displacements
- Element forces and moments
- Deflection plots

ANSYS is used by engineers worldwide in virtually all fields of engineering such as Structural, Thermal, fluid, low and high frequency electromagnetics. There are some industries where ANSYS are used:

- Aerospace
- Automotive
- Biomedical
- Bridges and buildings
- Electronics and appliances
- Heavy equipment and machinery
- Micro Electro Mechanical Systems (MEMS)
- Sporting goods

ANSYS finite element analysis software enables engineers to perform the following tasks:

- Build computer models or transfer CAD models of structures, products, components, or systems.
- Apply operating loads or other design performance conditions.
- Study physical responses, such as stress levels, temperature distributions, or electromagnetic fields.
- Optimize a design early in the development process to reduce production costs.
- Do prototype testing in environments where it otherwise would be undesirable or impossible (for example, biomedical applications).

The following analysis can be performed in ANSYS

- Structural analysis
- Thermal analysis
- Electromagnetic analysis
- Computational fluid dynamics

5.5 BUCKLING ANALYSIS

Buckling is a critical state of stress and deformation, at which a slight disturbance causes a gross additional deformation, or perhaps a total structural failure of the part. Structural behaviour of the part near or beyond 'buckling' is not evident from the normal arguments of statics. Buckling failures do not depend on the strength of the material, but are a function of the component dimensions and modulus of elasticity. Therefore, materials with a high strength will buckle just as quickly as low strength ones. If a structure has one or more dimensions that are small relative to the others (slender or thin-walled), and is subject to compressive loads, then a buckling analysis may be necessary.

From an FE analysis point of view, a buckling analysis is used to find the lowest multiplication factor for the load that will make a structure buckle. The result of such an analysis is a number of buckling load factors (BLF). The first BLF (the lowest factor) is always the one of interest. If it is less than unity, then buckling will occur due to the load being applied to the structure. The analysis is also used to find the shape of the buckled structure.

There are two types of analysis to predict the buckling load and buckling mode shape of the connecting rod: nonlinear buckling analysis and Eigenvalue (or linear) buckling analysis.

Nonlinear buckling analysis (Figure 5.7) is usually the more accurate approach and is therefore recommended for design or evaluation of actual structures. This technique employs a nonlinear static analysis with gradually increasing loads to seek the load level at which your structure becomes unstable.

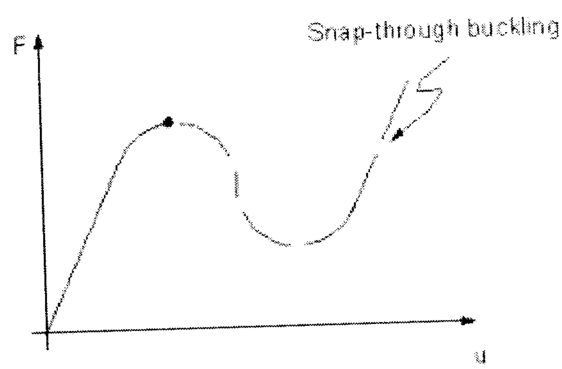


Figure 5.7 Non linear buckling curve

Eigenvalue (or linear) buckling analysis (Figure 5.8) predicts the theoretical buckling strength (the bifurcation point) of an ideal linear elastic structure. This method corresponds to the textbook approach to elastic buckling analysis: for instance, an eigenvalue buckling analysis of a column will match the classical Euler solution.

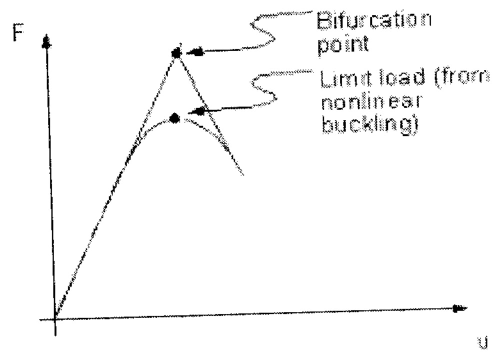


Figure 5.8 Linear bucking curve

A machine part subjected to an axial compressive force is called a strut. A strut may be horizontal, inclined or even vertical. But a vertical strut is known as a column. When a column or a strut is subjected to a compressive load and the load is gradually increased, a stage will reach and the column will be subjected to ultimate load. Beyond this the column will fail by crushing and the load will be known as crushing load or buckling load.

Sometimes a compression member does not fail entirely by crushing, but also by bending i.e., buckling. It has also been observed that all the short columns fail due to their crushing. But, if a long column is subjected to a compressive load, it is subjected to a compressive stress. If the load is gradually increased, the column will reach a stage, when it will start buckling. The load, at which the column tends to have lateral displacement or tense to buckle is called buckling load, critical load, or gripping load and the column is said to have developed an elastic instability. The buckling takes place about the axis having minimum radius of gyration or least moment of inertia. Moreover, the value of buckling load is low for long columns, and relatively high for short columns.

5.5.1 Types of End Conditions of Columns

There are a number of end conditions for columns. Euler's column theory provides three types of end conditions (as in figure 5.9) which are important for the analysis:

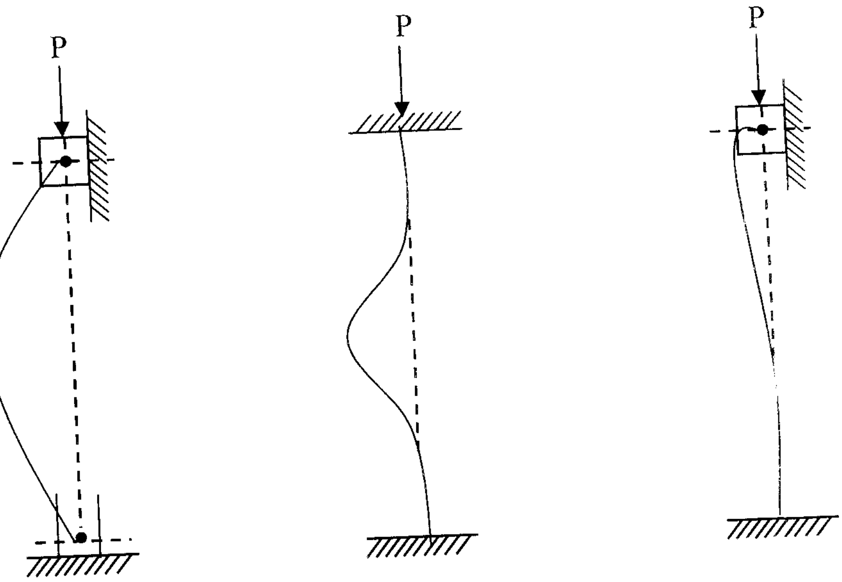


Figure 5.9 Types of End conditions for columns

1. Both the ends hinged or pin jointed.
2. Both the ends fixed.
3. One end is fixed and the other hinged.

According to the Euler's theory, the crippling or buckling load (W_{CR}) under various end conditions is represented by a general equation,

$$W_{cr} = \frac{C\pi^2 EI}{l^2} = \frac{C\pi^2 EAk^2}{l^2} \qquad \ldots(\because\ I = A.k^2)$$

$$= \frac{C\pi^2 EA}{(l/k)^2}$$

where E = Modulus of elasticity or Young's modulus for the material of the column,

A = Area of Cross-section,

k = Least radius of gyration of cross-section,

l = Length of the column, and

C = Constant, representing the end conditions of the column or end fixity coefficient.

Sometimes, the crippling load according to Euler's formula may be written as

$$W_{cr} = \frac{\pi^2 EI}{L^2}$$

where L is the equivalent length or effective length of the column. The equivalent length of a given column with given end conditions is the length of an equivalent column of the same material and the cross section with hinged ends to that of the given column.

The Euler's formula gives correct results only for very long columns. Though this formula is applicable for columns, ranging from very large to short ones, yet it does not give reliable results. Prof. Rankine, gave the following empirical formula for columns.

$$\frac{1}{W_{cr}} = \frac{1}{W_C} + \frac{1}{W_E} \qquad \ldots (1)$$

where W_{cr} = Crippling load by Rankine's formula,

W_C = Ultimate crushing load for the column $= \sigma c \times A$,

W_E = Crippling load, obtained by Euler's formula $\frac{\pi^2 EI}{L^2}$

A little consideration will show that the value of W_C will remain constant irrespective of the fact whether the column is a long one or short one. Moreover, in the case of short columns the value of W_E will be very high, therefore the value of $1/W_E$ will be quite negligible as compared to $1/W_C$. It is thus obvious, that the Rankine's formula will give the value of its crippling load (i.e., W_{cr}) approximately equal to the ultimate crushing load (i.e., W_C). It is thus obvious, that the Rankine's formula will give the value of its crippling load (i.e., W_{cr}) approximately equal to the crippling load by Euler's formula (i.e.. W_E). Thus, Rankine's formula gives a fairly correct result for all case of columns, ranging from short to long columns.

From equation (1),

$$\frac{1}{W_{cr}} = \frac{1}{W_C} + \frac{1}{W_E} = \frac{W_E + W_C}{W_C \times W_E}$$

$$W_{cr} = \frac{W_C \times W_E}{W_C + W_E} = \frac{W_C}{1 + \frac{W_C}{W_E}}$$

Now substituting the value of W_C and W_E in the above equation, we have

$$W_{cr} = \frac{\sigma_c \times A}{1 + \frac{\sigma_c \times A \times L^2}{\pi^2 EI}} = \frac{\sigma_c \times A}{1 + \frac{\sigma_c}{\pi^2 E} \times \frac{AL^2}{Ak^2}} \qquad \dots. (I = A \,.\, k^2)$$

$$= \frac{\sigma_c \times A}{1 + a\left(\frac{L}{k}\right)^2} = \frac{\text{Crushing Load}}{1 + a\left(\frac{L}{k}\right)^2}$$

where
- σ_c = Crushing stress or yield stress in compression,
- A = Cross-sectional area of the column,
- a = Rankine' constant = $\frac{\sigma_c}{\pi^2 E}$,
- L = Equivalent length of the column, and
- k = Least radius of gyration.

CHAPTER 6

FEA MODELLING OF THE CONNECTING ROD

6.1 GEOMETRIC MODEL

The connecting rod is designed as per the dimensions in PRO ENGINEER software. The geometric model consists of many features like fillets, oil holes, chamfers etc. A solid model of the connecting rod is shown in the figure. For FEA small CAD features that affect the analysis are removed to minimize the solution time and to increase the accuracy. The file is converted into IGES format and transferred to ANSYS software for analysis.

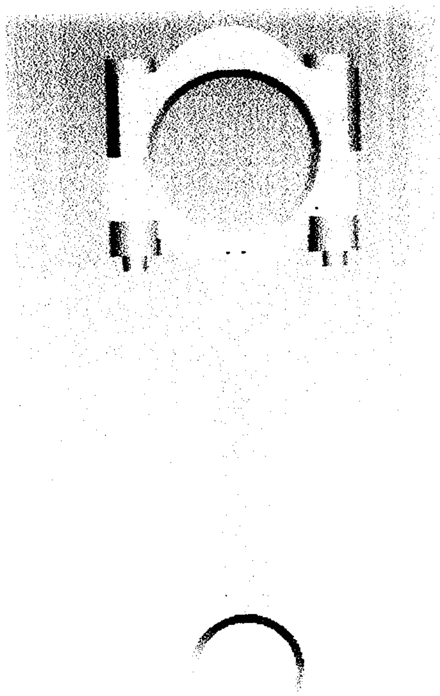
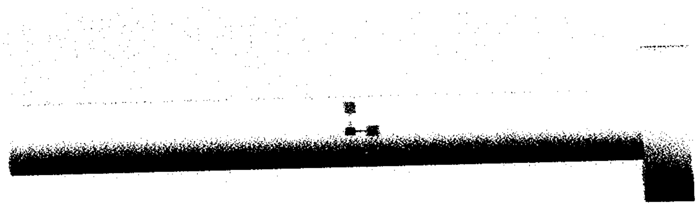
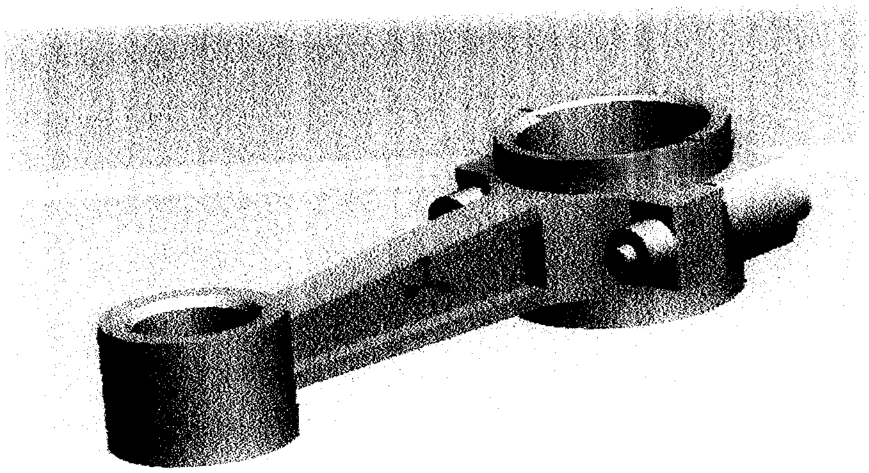


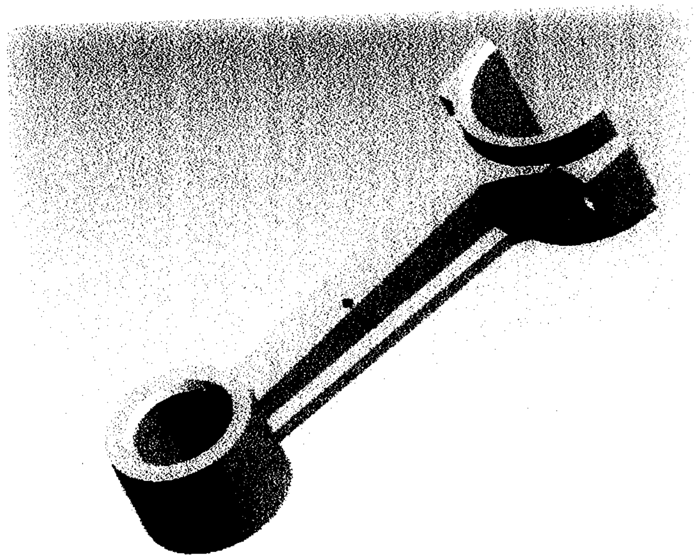
Figure 6.1 Top View



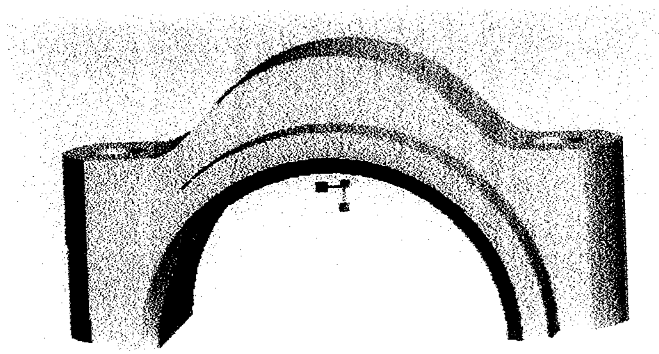
6.2 Geometric model of Bolt



6.3 Default view



6.4 Geometric model of Connecting Rod



6.5 Geometric model of Split Cap

6.1.1 Material Properties of Forged Steel

The connecting rod material is forged steel and it is isotropic material having same material properties in all directions.

TABLE 6.1 MATERIAL PROPERTIES OF FORGED STEEL

Young's Modulus E	207GPa
Yield Strength	700MPa
Ultimate Strength	938MPa
Poisson's Ratio	0.30
Mass Density	7820 kg/m3
Percent reduction in area, %RA	42%
Strength coefficient, K , MPa (ksi)	1400 (203.0)

6.2 MESH GENERATION

Meshing is one of the important steps in the finite element analysis of any component. The geometry is meshed with tetrahedral elements with various element lengths of 3 mm, 3.5 mm, 4 mm, and 4.5 mm elements.

SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has a quadratic displacement behavior and is well suited to modeling irregular meshes. The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. The element has plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities.

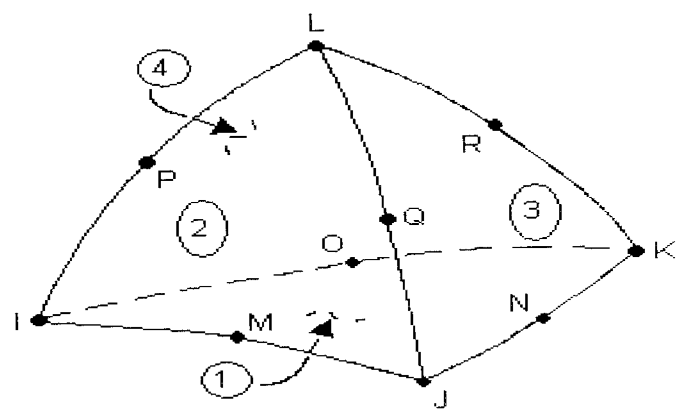


Figure 6.6 SOLID187 Geometry

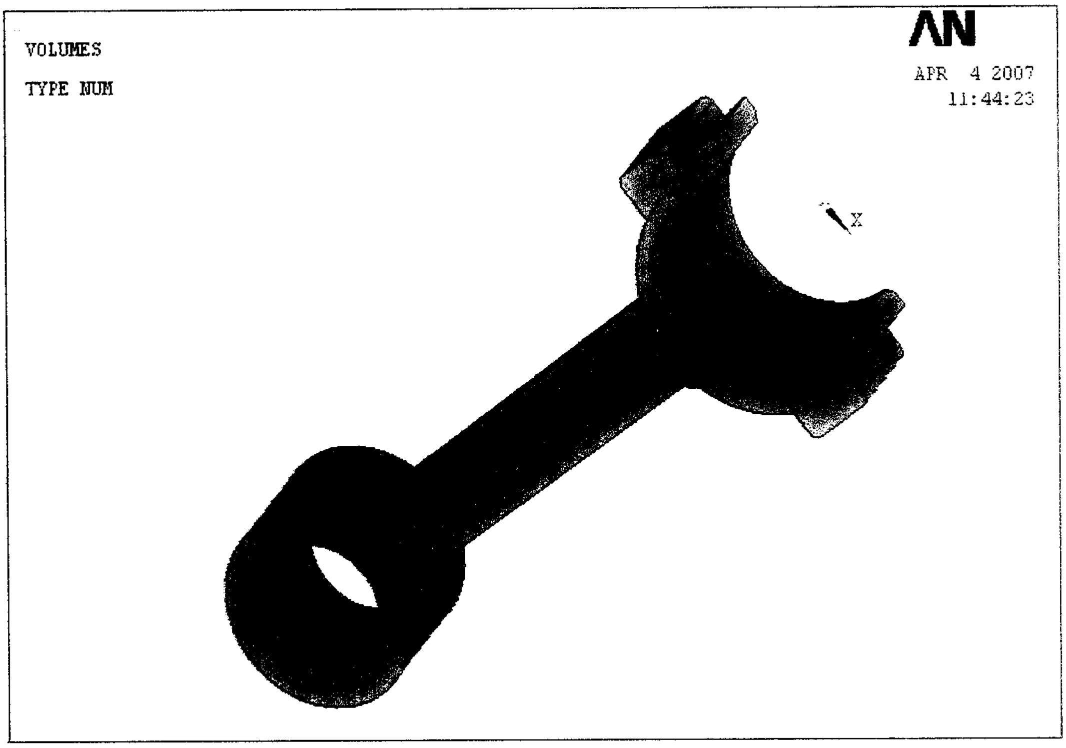


Figure 6.7 Imported CAD Geometry from PRO/Engineer

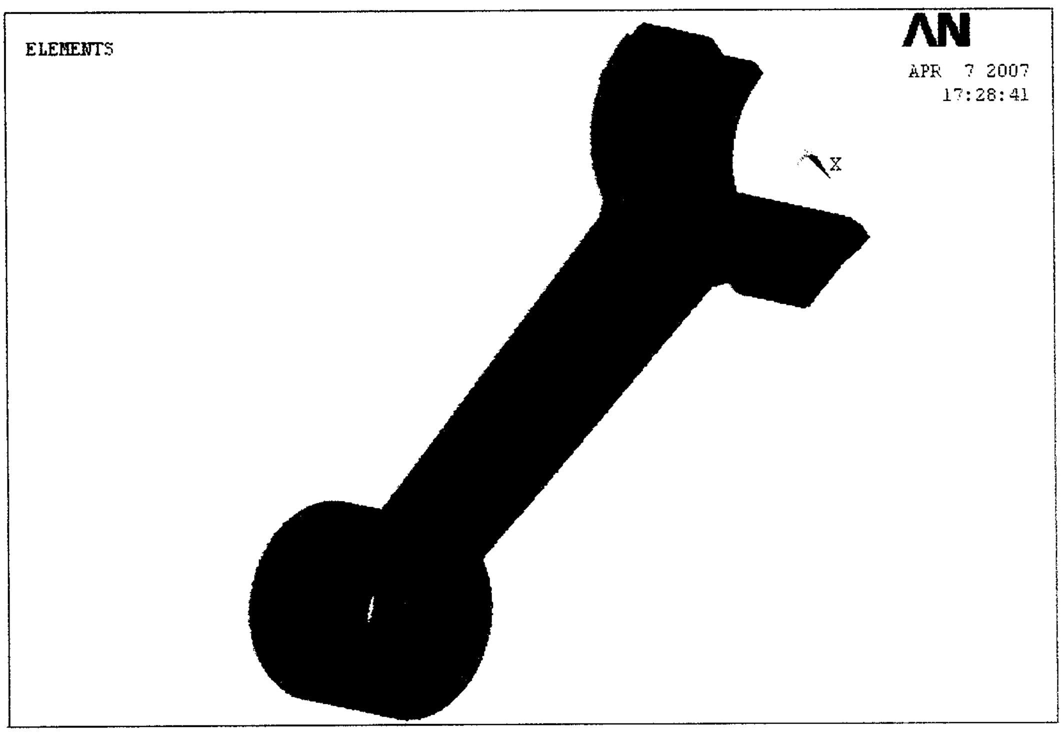


Figure 6.8 Finite element meshed model with element size 3

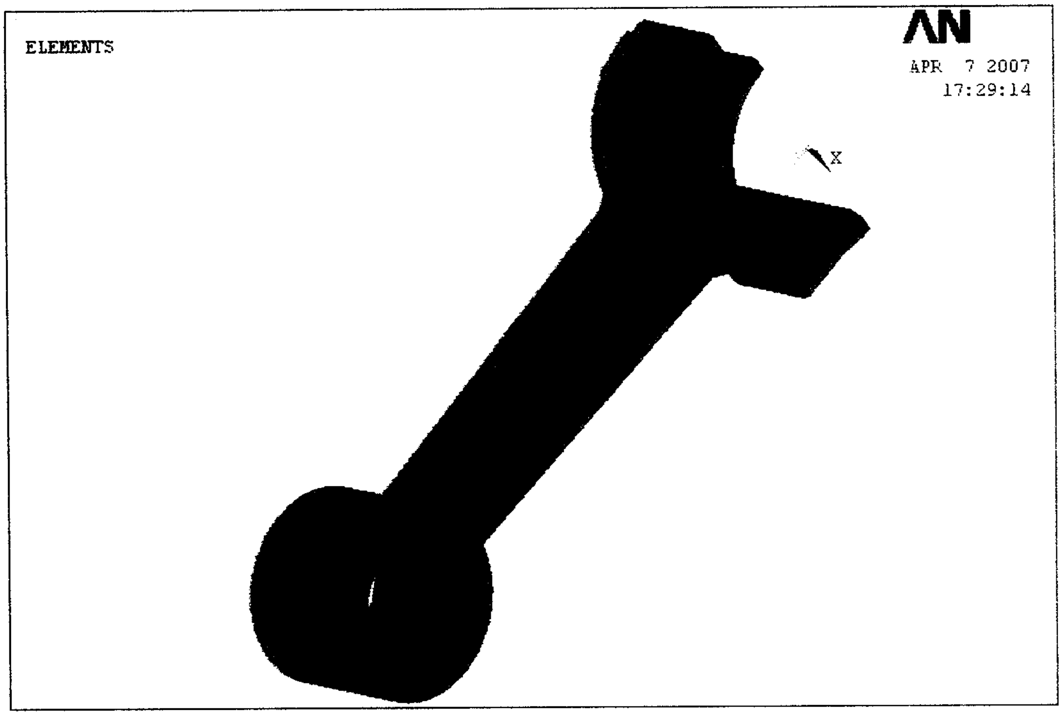


Figure 6.9 Finite element meshed model with element size 3.5

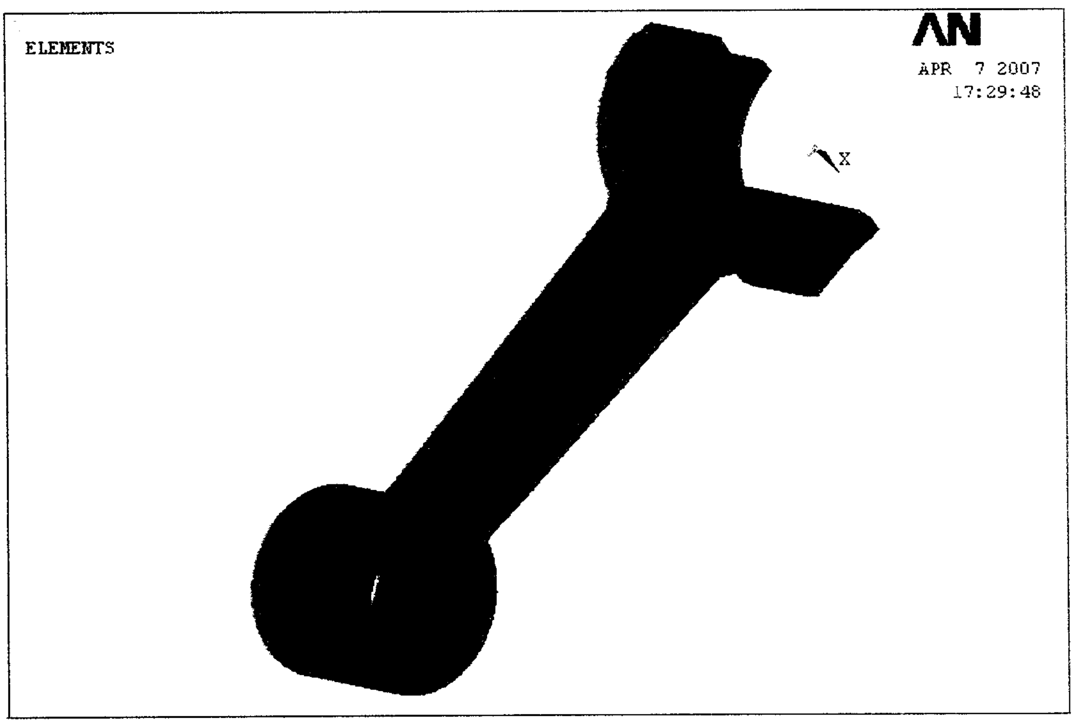


Figure 6.10 Finite element meshed model with element size 4

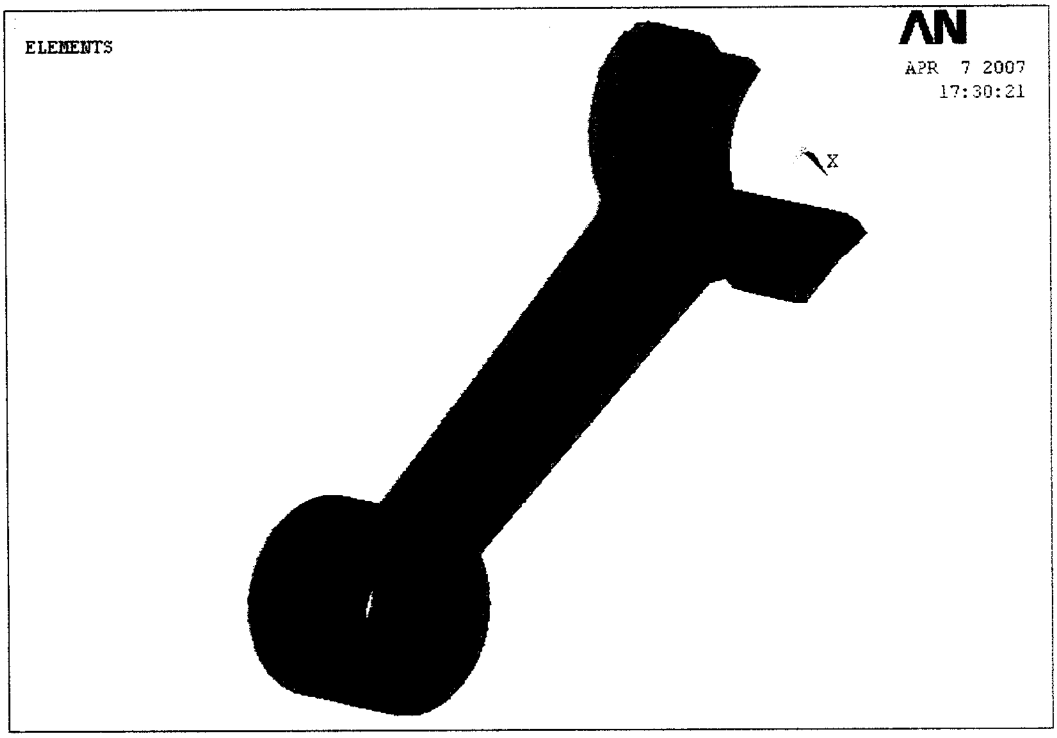


Figure 6.11 Finite element meshed model with element size 4.5

6.3 BOUNDARY CONDITIONS

6.3.1 Loadings

The crank and piston pin ends are assumed to have a sinusoidal distributed loading over the contact surface area, under compressive loading. The normal pressure on the contact surface is given by:

$$p = p_o \cos\theta$$

The total resultant load is given by:

$$P_t = \int_{-\pi/2}^{\pi/2} p_o(\cos^2\theta) r\, t\, d\theta = p_o\, r_1\, t\, \pi/2$$

$$p_o = P_t / (r\, t\, \pi/2)$$

For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 120° contact surface. For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 120° contact surface. The normal pressure is given by:

$$p = p_o$$

The total resultant load is given by:

$$P_c = \int_{-\pi/3}^{\pi/3} p_o (\cos^2 \theta) r_1 \, t \, d\theta = p_o \, r \, t \sqrt{3}$$

$$p_o = P_c / (r \, t \sqrt{3})$$

In this work two finite element models were analyzed. FEA for both conventional and composite connecting rod were analyzed. The connecting rod with load applied at the piston pin end and restrained at the crank end. The connecting rod was analyzed with different element sizes. In the analysis carried out, the axial load was 17.6 KN. Then the pressure constants are calculated.

The calculation of pressure constants done by:

Compressive Loading

Crank End: $p_o = 17600/ (24 \times 17.056 \times \sqrt{3}) = 24.82$ MPa

Piston pin End: $p_o = 17600/ (11.97 \times 18.402 \times \sqrt{3}) = 46.13$ MPa

Tensile Loading

Crank End: $p_o = 17600/ [24 \times 17.056 \times (\pi / 2)] = 27.37$ MPa

Piston pin End: $p_o = 17600/ [11.97 \times 18.402 \times (\pi / 2)] = 50.866$ MPa

6.3.2 Restraints

The FEA models were solved. The figures shows a FEA model in which compressive load is applied at the piston pin end and crank end is restrained. The half of the piston pin inner surface (120°) is completely restrained. The finite element model was analyzed for four different element sizes 3, 3.5, 4 and 4.5mm as in figure 6.12, 6.13, 6.14 and 6.15 respectively.

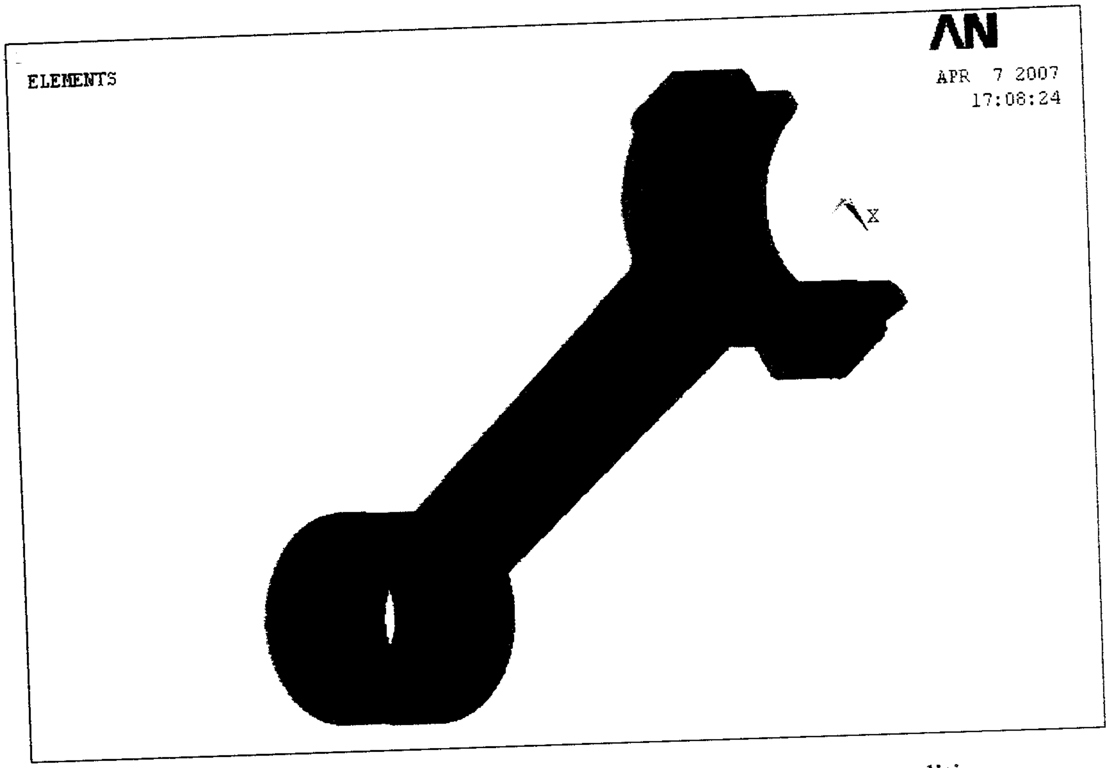


Figure 6.12 FEA model of connecting rod with boundary conditions
(element size 3)

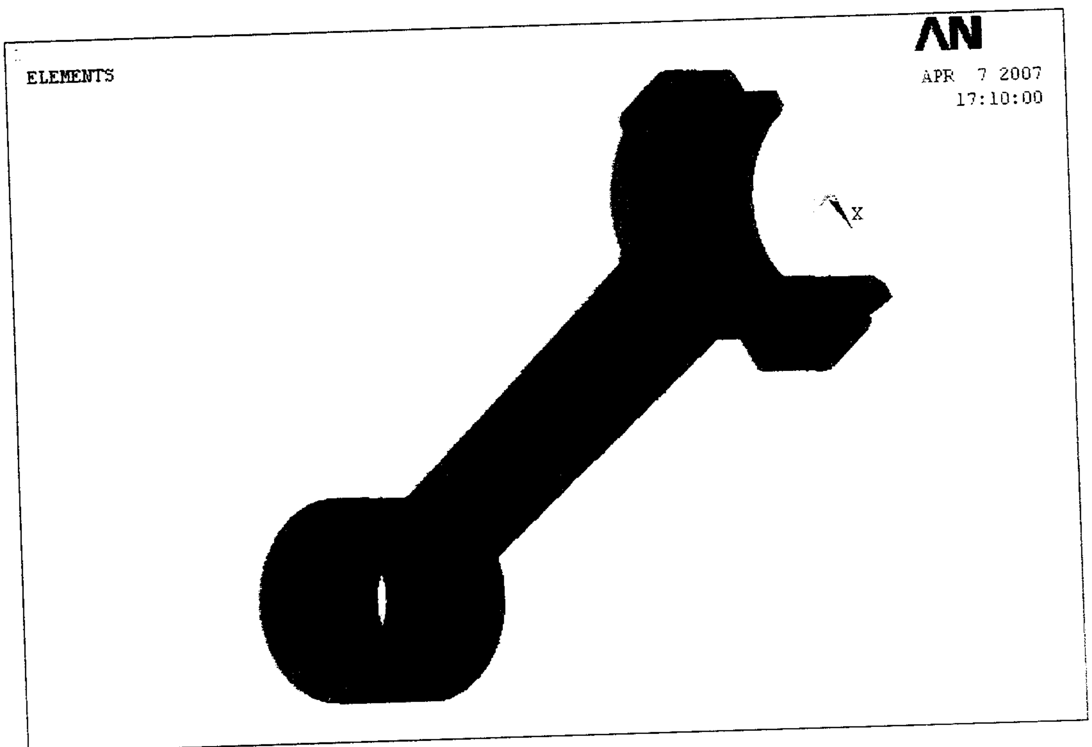


Figure 6.13 FEA model of connecting rod with boundary conditions
(element size 3.5)

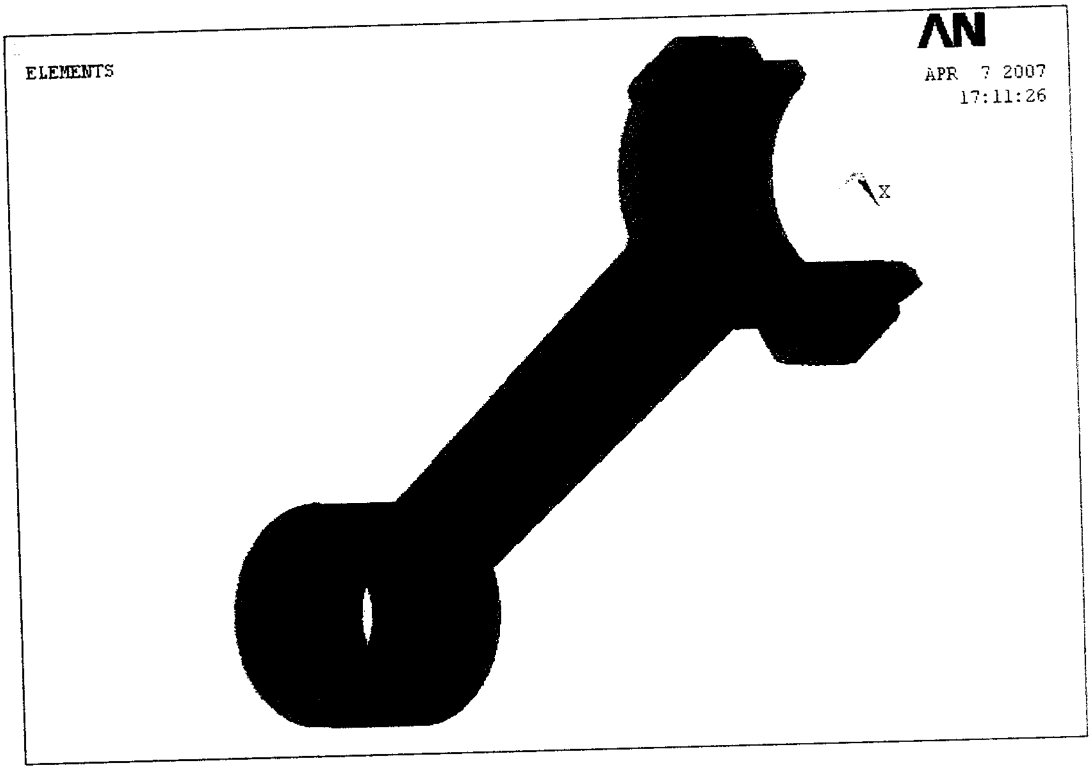


Figure 6.14 FEA model of connecting rod with boundary conditions
(element size 4)

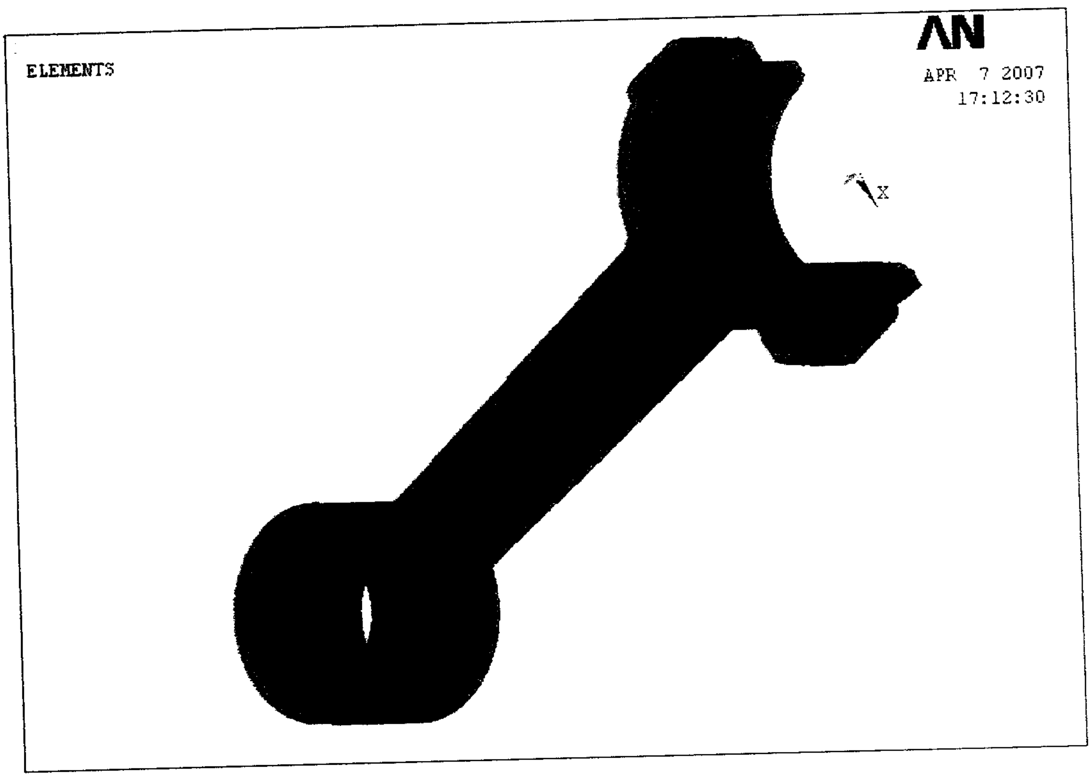


Figure 6.15 FEA model of connecting rod with boundary conditions
(element size 4.5)

CHAPTER 7

RESULTS AND DISCUSSIONS

The finite element model was analyzed in commercial coded finite element analysis software ANSYS. The stress, deflection patterns are determined using ANSYS. The connecting rod is meshed with different element sizes.

7.1 RESULTS OF BUCKLING ANALYSIS

The Von Misses stress variation, the stress intensity and the total displacement for the composite connecting rod with the load applied at the piston pin end with element size 3 is shown in the figure 7.1, 7.2 and 7.3 respectively.

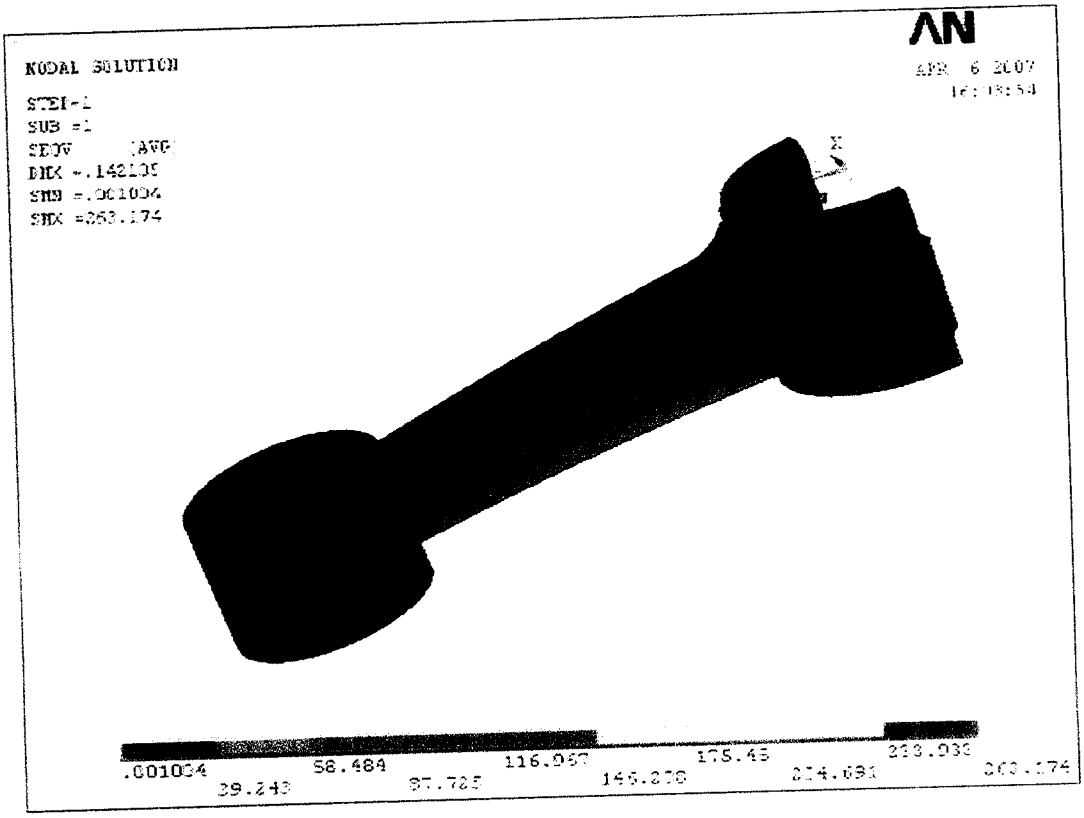


Figure 7.1 Von Misses stress variation for composite connecting rod with element size 3

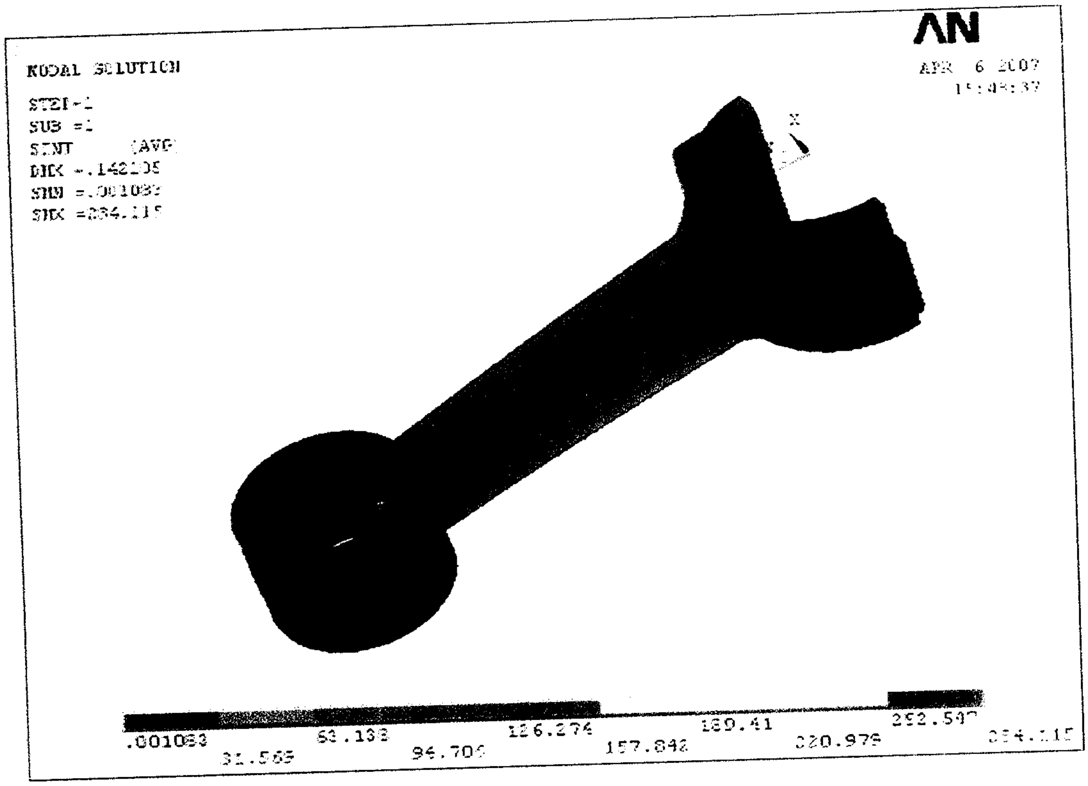


Figure 7.2 Stress intensity variation for composite connecting rod with element size 3

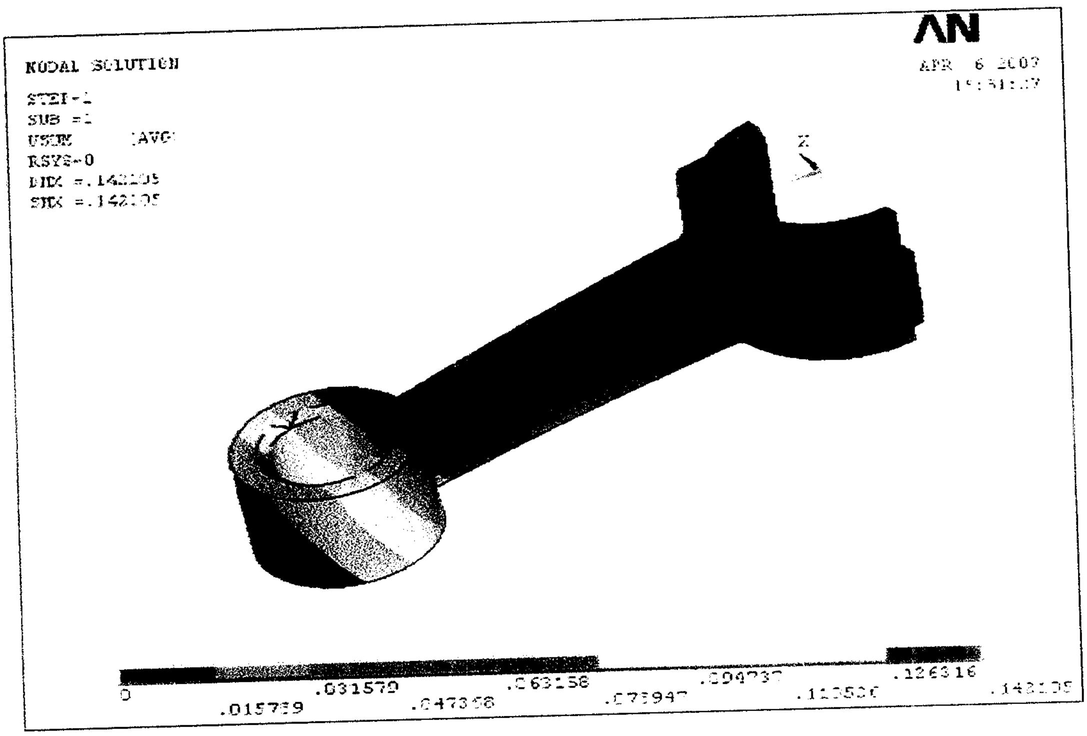


Figure 7.3 Total displacement for composite connecting rod with element size 3

The Von Misses stress variation and the total displacement for the conventional connecting rod with the load applied at the piston pin end with element size 3 is shown in the figure 7.4 and 7.5 respectively.

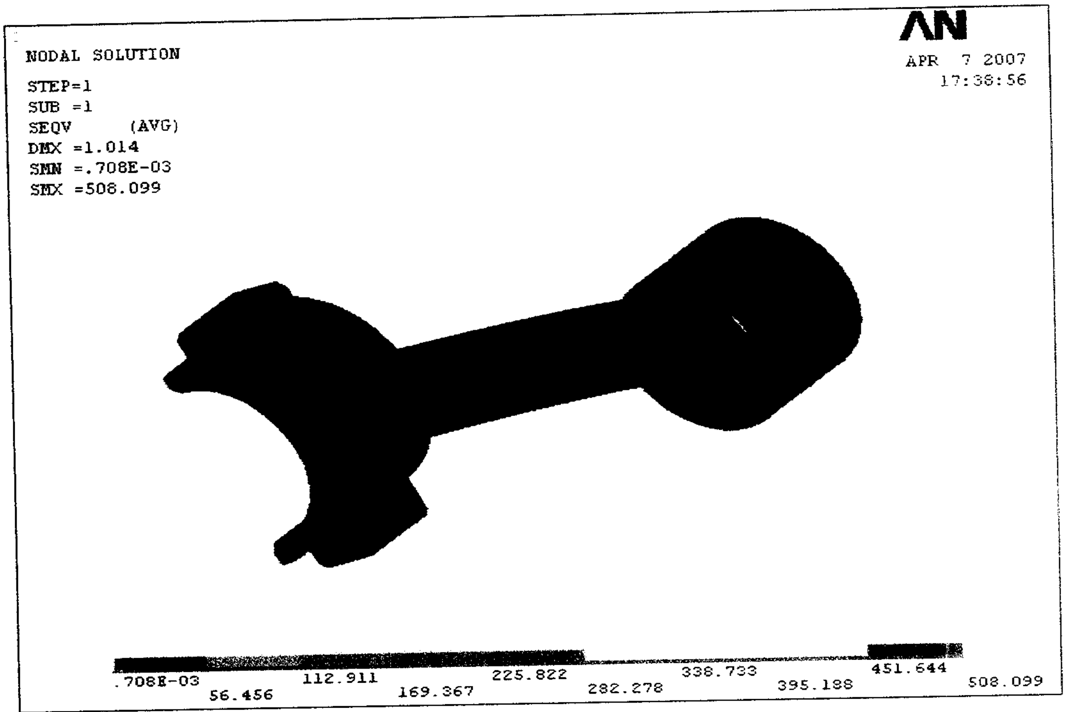


Figure 7.4 Von Misses stress variation for conventional connecting rod with element size 3

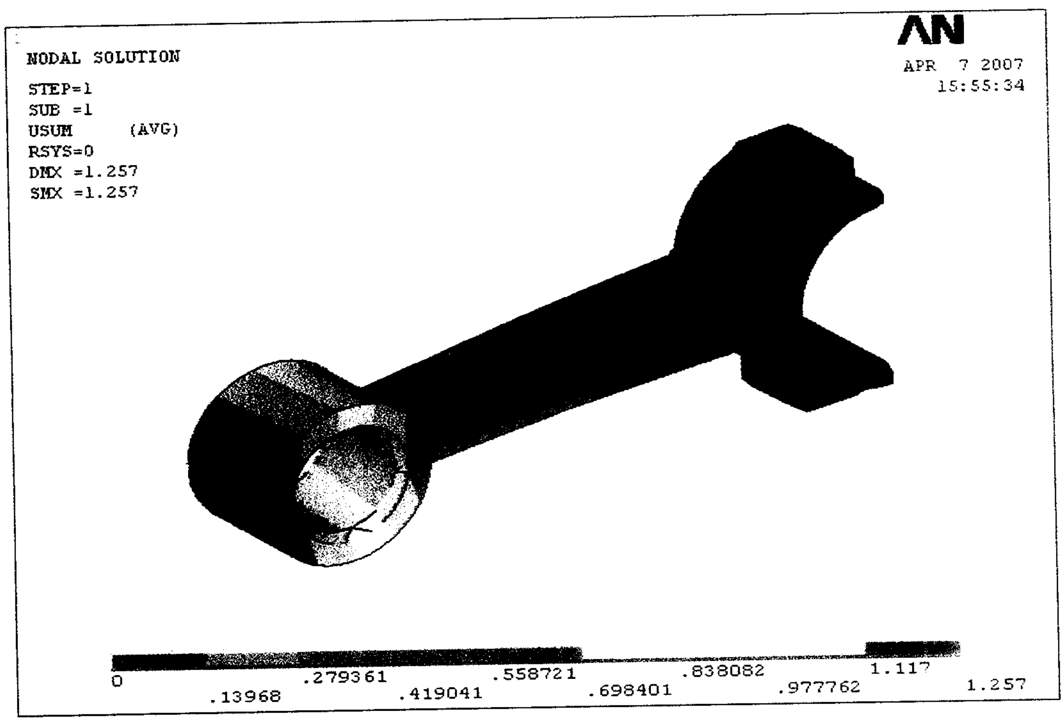


Figure 7.5 Total displacement for conventional connecting rod with element size 3

The Von Misses stress variation and the total displacement for the composite connecting rod with the load applied at the piston pin end with element size 3.5 is shown in the figure 7.6 and 7.7.

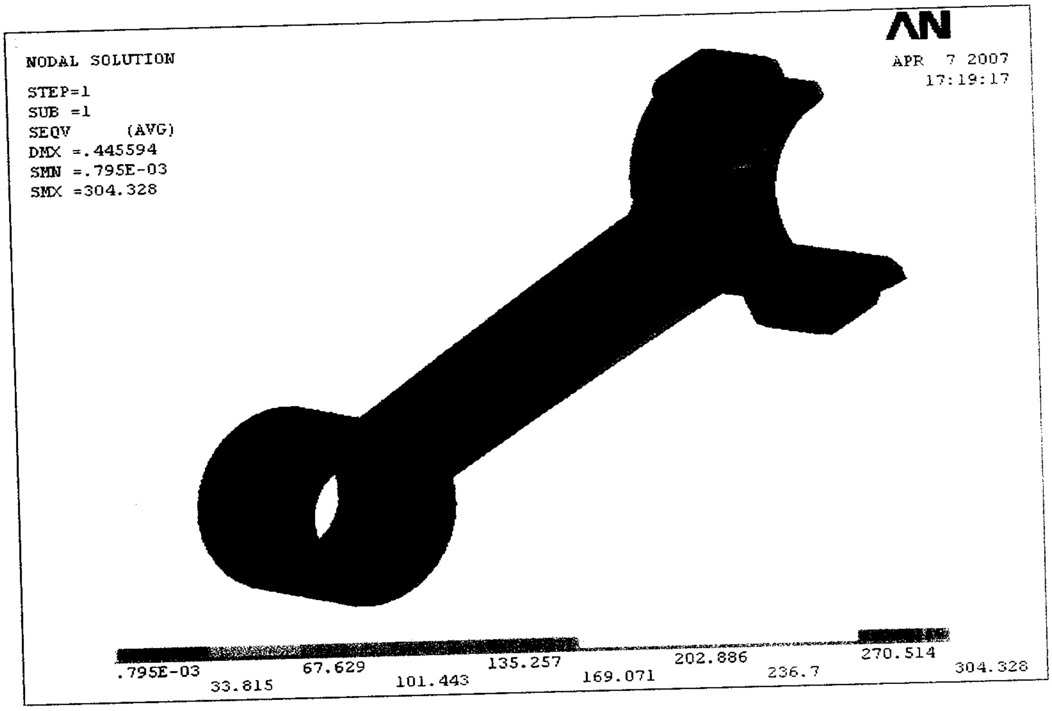


Figure 7.6 Von Misses stress variation for composite connecting rod with element size 3.5

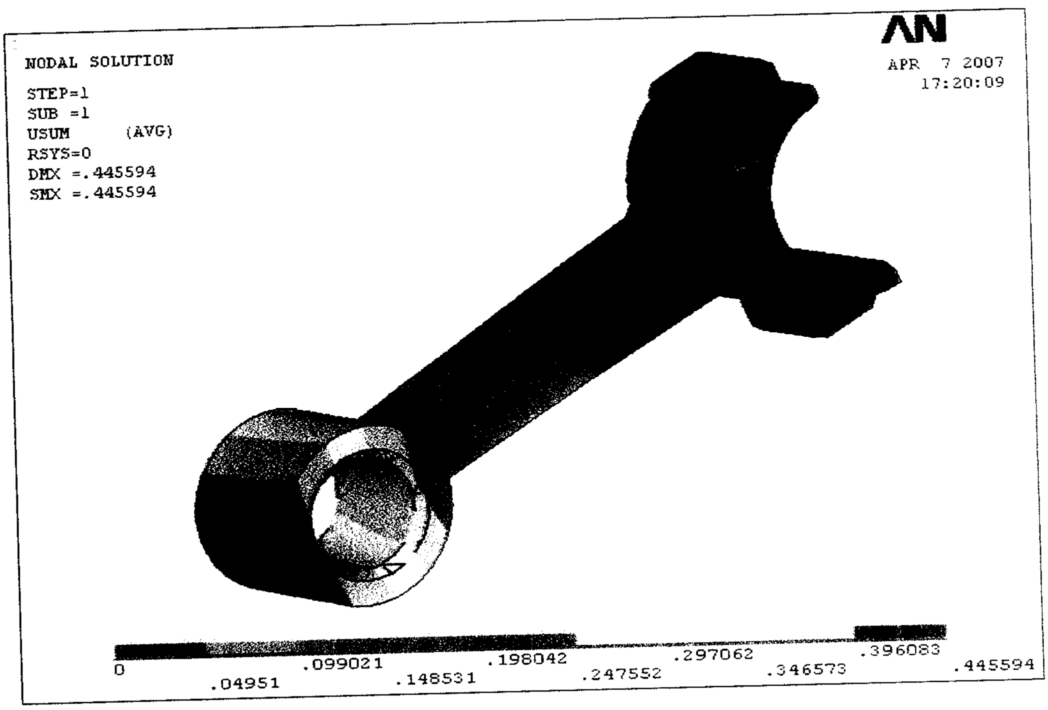


Figure 7.7 Total displacement for composite connecting rod with element size 3.5

The Von Misses stress variation for the conventional connecting rod with the load applied at the piston pin end with element size 3.5 is shown in the figure 7.8.

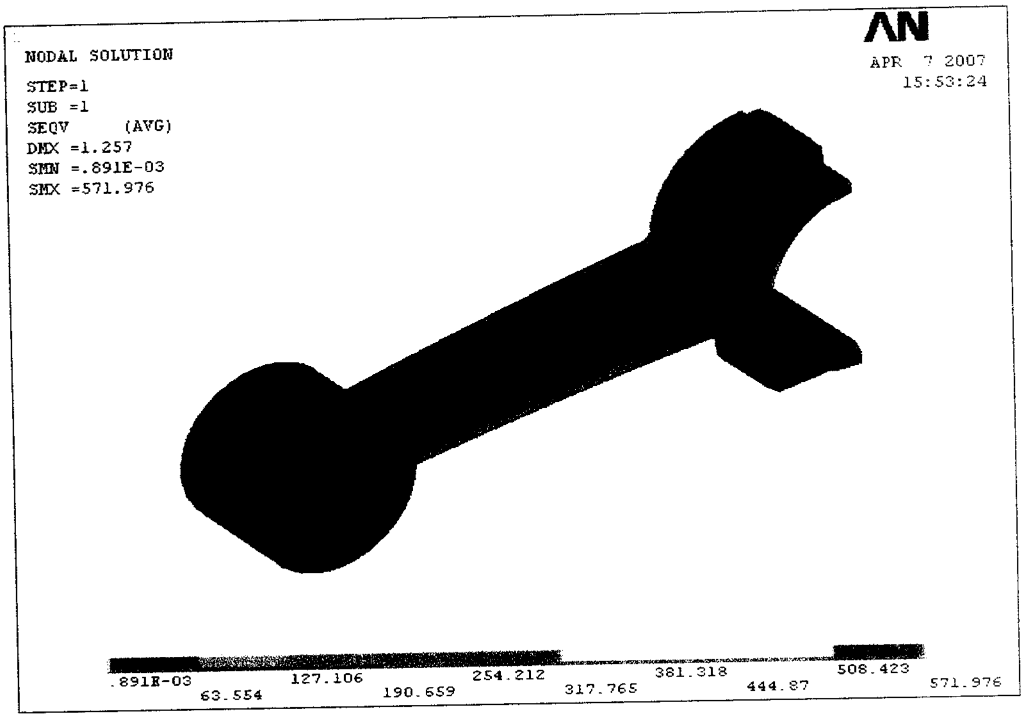


Figure 7.8 Von Misses stress variation for conventional connecting rod with element size 3.5

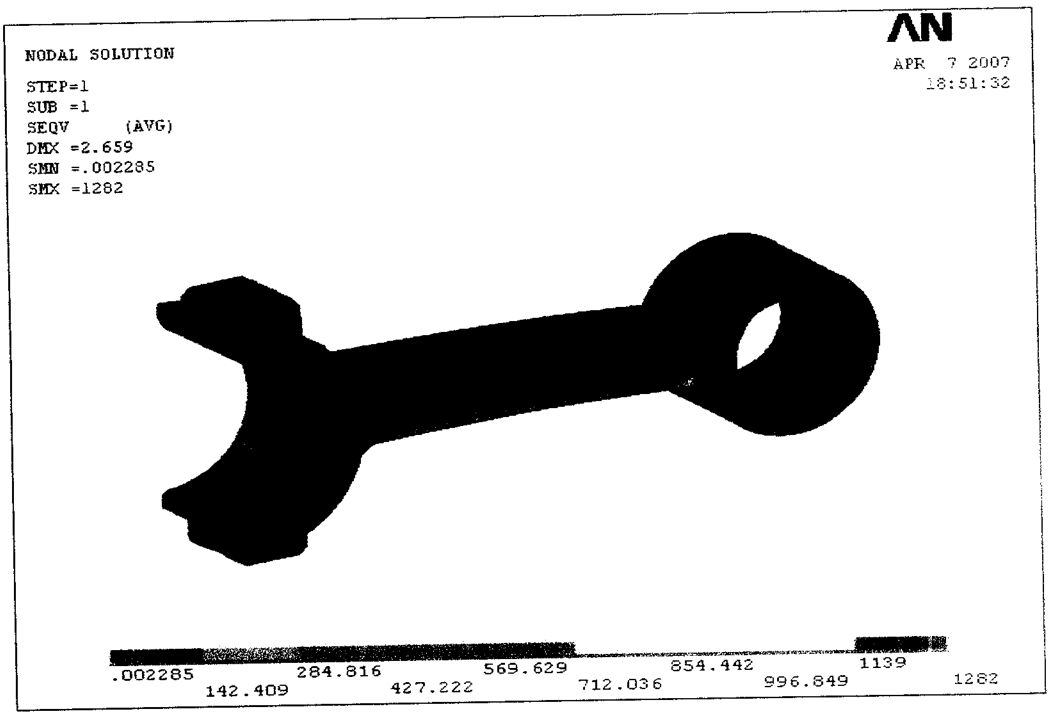


Figure 7.9 Von Misses stress variation for conventional connecting rod with element size 4.5

The Von Misses stress variation and total displacement for the conventional connecting rod with the load applied at the piston pin end with element size 4.5 is shown in the figure 7.9 and 7.10 respectively.

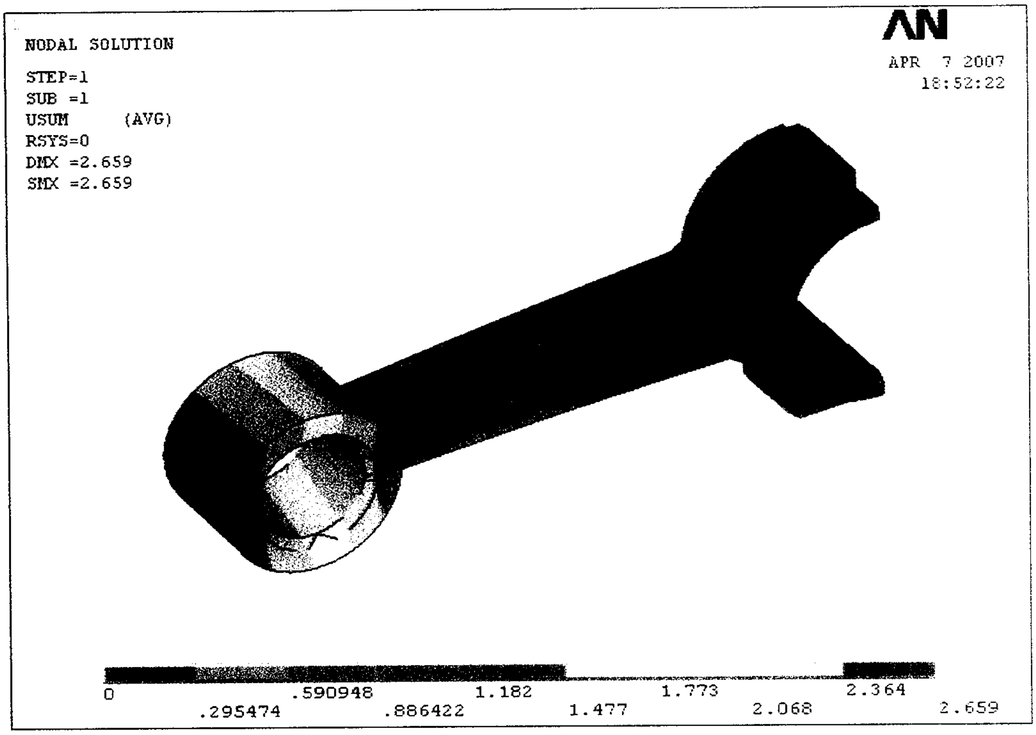


Figure 7.10 Total displacement for conventional connecting rod with element size 4.5

7.2 RESULTS COMPARISON

The Von Misses stress results for the conventional connecting rod is 571.976N/mm^2 which is less than the yield strength of the material 700Mpa. The predicted deflection value is 1.257mm. The Von Misses stress results for the composite connecting rod is 263.174N/mm^2 which is less than the longitudinal tensile strength of 1462MPa and longitudinal compressive strength of 2990MPa. The predicted deflection value is 0.142105mm.

The above results are for element size of 3mm. When the element size was varied to 3.5mm the predicted stress values for composite connecting rod was 304.328N/mm^2 and the deflection value is 0.445594mm. For the conventional connecting rod with element size of 4mm the stress value is 508.099N/mm^2 and the deflection value is 1.014mm. From the above results it is proved that composite connecting rod has minimum deflection and stress values less than the yield stress and it posses enough stiffness to bear the load while maintaining its weight.

TABLE 7.1 COMPARISON OF CONVENTIONAL ROD AND COMPOSITE ROD

Element Sizes	**Conventional Connecting Rod**		**Composite Connecting Rod**	
	Von Misses Stress (N/mm^2)	Total displacement (mm)	Von Misses Stress (N/mm^2)	Total displacement (mm)
Element Size 3	508.099	1.014	263.174	0.142
Element Size 3.5	571.976	1.257	304.328	0.4455

CHAPTER 9
CONCLUSION AND FUTURE WORK

A comparative analysis of the conventional and composite material connecting rod has been carried out. The angular velocity and the angular acceleration for the connecting rod are determined at each crank angles and the maximum angular velocity and angular acceleration was determined. Then the crank angles were selected based on the magnitude of forces and the FEA analysis was performed with different element sizes. From the analysis stage the stress and deflection values are analyzed for different element sizes (stress and strain analysis of both connecting rods). First the stress and deflection values are predicted for the conventional connecting rod and then the analysis was performed for composite connecting rod and the stress and deflection values are calculated.

The limitations of the project are:

The analysis of connecting rod is performed in ANSYS 8.0. Analysis of composite materials can be handled by ANSYS 8.0 up to certain extent. The results obtained from ANSYS 8.0 are less accurate. Composite materials analysis can be handled easily by ABAQUS. The same analysis performed in ABAQUS gives much more closer and accurate solution. Due to availability of the software the analysis is performed only in ANSYS 8.0.

This project work can be further extended by

1. The element sizes can be varied to different sizes which will help to determine the mesh convergence effectively.
2. The analysis can be done for the whole connecting rod assembly by modeling the crankshaft, crankshaft bearings etc. Spring elements can be used to model the above components.
3. Adequate analysis procedure for composite connecting rod can be used to model the composite materials.

4. Fatigue strength, an important design parameter can be included which helps to predict the life of the connecting rod. Experiments can be conducted for determining the alternating stress and no of cycles failed.
5. Aluminium silicon carbide is used as MMC's for this analysis. Different MMC's can be used for the analysis by giving their material properties.

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