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# **Design and Structural Analysis of a Connecting Rod**



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#### ABSTRACT

The main objective of this study was to explore weight and cost reduction opportunities for a production forged steel connecting rod. This has entailed performing a detailed load analysis. Therefore, this study has dealt with two subjects, first, dynamic load and quasi-dynamic stress analysis of the connecting rod, and second, optimization for weight and cost.

In the first part of the study, the loads acting on the connecting rod as a function of time were obtained. The relations for obtaining the loads and accelerations for the connecting rod at a given constant speed of the crankshaft were also determined. Quasi dynamic finite element analysis was performed at several crank angles. The stress-time history for a few locations was obtained. The difference between the static FEA, quasi dynamic FEA was studied. Based on the observations of the quasi-dynamic FEA, static FEA and the load analysis results, the load for the optimization study was selected. The results were also used to determine the variation of Stress plots, displacements and mode shapes are validated. The component was optimized for weight and cost subject to dynamic load and manufacturability. It is the conclusion of this study that the connecting rod can be designed and optimized under a load range comprising tensile load corresponding to 3600 crank angle at the maximum engine speed as one extreme load, and compressive load corresponding to the peak gas pressure as the

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other extreme load. Furthermore, the existing connecting rod can be replaced with a new connecting rod made of C-70 steel that is 10% lighter and25% less expensive due to the steel's fracture crackability. The fracture crackability feature, facilitates separation of cap from rod without additional machining of the mating surfaces. Yet, the same performance can be expected in terms of component durability.

#### **INTRODUCTION:**

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. Every vehicle that uses an internal combustion engine requires at least one connecting rod depending upon the number of cylinders in the engine. Connecting rods for automotive applications are typically manufactured by forging from either wrought steel or powdered metal. They could also be cast.

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 (Gupta, 1993). Between the forging processes, powder forged or drop forged, each process has its own pros and cons. Powder metal manufactured blank shave the advantage of being near net shape, reducing material waste. However, the cost of the blank is high due to the high

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material cost and sophisticated manufacturing techniques (Repgen, 1998). 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 (Repgen, 1998). A sizeable portion of the US market for connecting rods is currently consumed by the powder metal forging industry. A comparison of the European and North American connecting rod markets indicates that according to an unpublished market analysis for the year 2000 (Ludenbach, 2002), 78% of the connecting rods in Europe (total annual production: 80 million approximately) are steel forged as opposed to 43% in North America (total annual production: 100 million approximately), as shown in Figure 1.1.

In order to recapture the US market, the steel industry has focused on development of production technology and new steels. AISI (American Iron and Steel Institute) funded a research program that had two aspects to address. The first aspect was to investigate and compare fatigue strength of steel forged connecting rods with that of the powder forged connecting rods. The second aspect was to optimize the weight and manufacturing cost of the steel forged connecting rod. The first aspect of this research program has been dealt with in a master's thesis entitled "Fatigue Behavior and Life predictions of Forged Steel and PM Connecting Rods" (Afzal A., 2004). This current thesis deals with the second aspect of the study, the optimization part. Due to its large volume production, it is only logical that optimization of the connecting rod for its weight or volume will result in large-scale savings. It can also achieve the objective of reducing the weight of the engine component, thus reducing inertia loads, reducing engine weight and improving engine performance and fuel economy.

In a reciprocating piston engine, the connecting rod connects the piston to the crank or crankshaft. Together with the crank, they form a simple mechanism that converts reciprocating motion into rotating motion. Connecting rods may also convert rotating motion into reciprocating motion. Historically, before the development of engines, they were first used in this way.

As a connecting rod is rigid, it may transmit either a push or a pull and so the rod may rotate the crank through both halves of a revolution, i.e. piston pushing and piston pulling. Earlier mechanisms, such as chains, could only pull. In a few two-stroke engines the connecting rod is only required to push.

Today, connecting rods are best known through their use in internal combustion piston engines, such as automotive engines. These are of a distinctly different design from earlier forms of connecting rods, used in steam engines and steam locomotives.

#### **DESIGN OF CONNECTING ROD**

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross- section of the connecting rod is designed as a strut and the rankine formula is used. A connecting rod subjected to an axial load W may buckle with x-axis as neutral axis in the plane of motion of the connecting rod, {or} y-axis is a neutral axis. The connecting rod is considered like both ends hinged for buckling about x-axis and both ends fixed for buckling about y-axis. A connecting rod should be equally strong in buckling about either axis. According to rankine formulae

$$\begin{split} w_{cr} \text{ About } x - axis &= [\sigma_c \times A]/1 + [L/K_{xx}]^2 = \\ [\sigma_c \times A]/1 + [l/K_{xx}]^2 \quad [\therefore \text{ for both ends hinged } L = l] \\ w_{cr} \text{ About } y - axis &= [\sigma_c \times A]/1 + [L/K_{yy}]^2 \\ &= [\sigma_c \times A]/1 + a[l/2K_{yy}]^2 \\ [\therefore \text{ for both ends fixed } L = l/2] \end{split}$$

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

$$\begin{split} [\sigma_c \times A]/1 + [l/K_{xx}]^2 &= [\sigma_c \times A]/1 + a[l/K_{yy}]^2 \quad \text{or} \\ [l/K_{xx}]^2 &= [l/2K_{yy}]^2 \end{split}$$

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$$\begin{split} K_{xx}^2 &= 4K_{yy}^2 \quad \text{or} \quad I_{xx} = 4I_{yy} \\ [ \therefore I = A \times K^2 ] \end{split}$$

This shows that the connecting rod is four times strong in buckling about y-axis than about axis. If  $I_{xx} > 4I_{yy}$ Then buckling will occur about y-axis and if  $I_{xx} < 4I_{yy}$ , then buckling will occur about x-axis .In Actual practice  $I_{xx}$  is kept slightly less than  $4I_{yy}$ . It is usually taken between 3 and 3.5 and the Connecting rod is designed for buckling about x-axis. The design will always be satisfactory for buckling about y-axis.

Area of the cross section =  $2[4t \times t] + 3t \times t=11t2$ Moment of inertia about x-axis =  $2[4t\times t] + 3t\times t=11t2$ 

#### NOMENCLATURE OF CONNECTING ROD



It interconnects the piston and the crank shaft and transmits the gas forces from the piston to the crankshaft. Its primary function is to transmit the push and pull from the piston pin to the crank pin and thus convert the reciprocating motion of the piston into rotary motion of the crank the usual form of the connecting rod in internal combustion engines. It consists of a long shank a small end and big end. The small end of connecting rod is usually made in the form of an eye and is provided with a bush. It is connected to the piston by means of piston pin. The big end of connecting rod is usually made into two halves so that it can be mounted easily on the crank pin bearing shells.



Drawing of the connecting rod (bolt holes not included).

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The split is fastened to big end with two cap bolts. Big end bearing is allowed for by inserting thin metallic strip known as shims. The big end bearing is usually splash lubricated while the small end bearing is pressure lubricated.

Forces acting on the Connecting Rod The various forces acting on the connecting rod are as follows: Forces on the piston due to gas pressure and inertia of the reciprocating parts.

1. Forces on the piston due to gas pressure and inertia of the reciprocating parts.

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

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

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



In a reciprocating piston engine, the connecting rod connects the piston to the crank or crankshaft. In modern automotive internal combustion engines, the connecting rods are most usually made of steel for production engines, but can be made of aluminium (for lightness and the ability to absorb high impact at the expense of durability) or titanium (for a combination of strength and lightness at the expense of affordability) for high performance engines, or of cast iron for applications such as motor scooters. The small end attaches to the piston pin, gudgeon pin (the usual British term) or wrist pin, which is currently most often press fit into the con rod but can swivel in the piston, a "floating wrist pin" design. The connecting rod is under tremendous stress from the reciprocating load represented by the piston, actually stretching and being compressed with every rotation, and the load increases to the third power with increasing engine speed. Failure of a connecting rod,



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usually called "throwing a rod" is one of the most common causes of catastrophic engine failure in cars, frequently putting the broken rod through the side of the crankcase and thereby rendering the engine irreparable; it can result from fatigue near a physical defect in the rod, lubrication failure in a bearing due to faulty maintenance or from failure of the rod bolts from a defect, improper tightening, or re-use of already used (stressed) bolts where not recommended. Despite their frequent occurrence on televised competitive automobile events, such failures are quite rare on production cars during normal daily driving.

This is because production auto parts have a much larger factor of safety, and often more systematic quality control. When building a high performance engine, great attention is paid to the connecting rods, eliminating stress risers by such techniques as grinding the edges of the rod to a smooth radius, shot penning to induce compressive surface stresses (to prevent crack initiation), balancing all connecting rod/piston assemblies to the same weight and Magna fluxings to reveal otherwise invisible small cracks which would cause the rod to fail under stress. In addition, great care is taken to torque the con rod bolts to the exact value specified; often these bolts must be replaced rather than reused. The big end of the rod is fabricated as a unit and cut or cracked in two to establish precision fit around the big end bearing shell. Recent engines such as the Ford 4.6 litre engine and the Chrysler 2.0 litre engine have connecting rods made using powder metallurgy, which allows more precise control of size and weight with less machining and less excess mass to be machined off for balancing.

The cap is then separated from the rod by a fracturing process, which results in an uneven mating surface due to the grain of the powdered metal. This ensures that upon reassembly, the cap will be perfectly positioned with respect to the rod, compared to the minor misalignments, which can occur if the mating surfaces are both flat. A major source of engine wear is the sideways force exerted on the piston through the con rod by the crankshaft, which typically wears the cylinder into an oval cross-section rather than circular, making it impossible for piston rings to correctly seal against the cylinder walls. Geometrically, it can be seen that longer connecting rods will reduce the amount of this sideways force, and therefore lead to longer engine life.

However, for a given engine block, the sum of the length of the con rod plus the piston stroke is a fixed number, determined by the fixed distance between the crankshaft axis and the top of the cylinder block where the cylinder head fastens; thus, for a given cylinder block longer stroke, giving greater engine displacement and power, requires a shorter connecting rod (or a piston with smaller compression height), resulting in accelerated cylinder wear.

#### **2 LITERATURE REVIEW**

The connecting rod is subjected to a complex state of loading. It undergoes high cyclic loads of the order of 108 to 109 cycles, which range from high compressive loads 3 due to combustion, to high tensile loads due to inertia. Therefore, durability of this component is of critical importance. Due to these factors, the connecting rod has been the topic of research for different aspects such as production technology, materials, performance simulation, fatigue, etc. For the current study, it was necessary to investigate finite element modelling techniques, optimization techniques, developments in production technology, new materials, fatigue modelling, and manufacturing cost analysis. This brief literature survey reviews some of these aspects.

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. They modelled the connecting rod cap separately, and also modelled the bolt pretension using beam elements and multi point constraint equations.



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In a study reported by Repgen (1998), based on fatigue tests carried out on identical components made of powder metal and C-70 steel (fracture splitting steel), he notes that the fatigue strength of the forged steel part is 21% higher than that of the powder metal component. He also notes that using the fracture splitting technology results in a 25% cost reduction over the conventional steel forging process. These factors suggest that a fracture splitting material would be the material of choice for steel forged connecting rods. He also mentions two other steels that are being tested, a modified micro-alloyed steel and a modified carbon steel. Other issues discussed by Repgen are the necessity to avoid jig spots along the parting line of the rod and the cap, need of 4 consistency in the chemical composition and manufacturing process to reduce variance in microstructure and production of near net shape rough part.

Park et al. (2003) investigated micro structural behaviour at various forging conditions and recommend fast cooling for finer grain size and lower network ferrite content. From their research they concluded that laser notching exhibited best fracture splitting results, when compared with broached and wire cut notches. They optimized the fracture splitting parameters such as, applied hydraulic pressure, jig set up and geometry of cracking cylinder based on delay time, difference in cracking forces and roundness. They compared fracture splitting high carbon micro-alloyed steel (0.7% C) with carbon steel (0.48% C) using rotary bending fatigue test and concluded that the former has the same or better fatigue strength than the later. From a comparison of the fracture splitting high carbon micro-alloyed steel and powder metal, based on tension-compression fatigue test they noticed that fatigue strength of the former is 18% higher than the later.

## 3.FINITE ELEMENT ANALYSIS 3.1 INTRODUCTION TO F.E.M.

The finite element method combines in an elegant way the best features of the two approximate methods of analysis (viz.,) Functional approximation and Finite differences method .In particular finite element method can be explained through physical concept and hence it is most appealing to the engineer. And the method is amenable to systematic computer program and offers scope for application to a wide range of analysis problems .The basic concept is that a body or a structural may be divided into small elements of finite dimensions called finite elements. This process of dividing a elements is continuum into finite known discretisation. The original body or the structure is then considered as an assemblage of these elements connected at a finite number of joints called nodes or nodal points. Similar concept is used in finite difference method

The properties of the element are formulated and combined to obtain the solution for the entire body. This follows the concept used in Rayleigh-Ritz procedure of functional approximation. Only difference is that the approximation is made at the element level itself. Secondly in this method attention is mainly devoted to the formulation of properties of the constituent elements, instead of solving the problem for the entire structure or body in one operation. The procedure for combining the elements, solution of equations and evaluation of element strains & stresses are the same for any type of structural system or body. Hence the finite element method offers scope for developing general-purpose program with properties of various types of elements forming an element library and the other procedures of analysis forming the common core segment. This modular structure of the program organizations is well explained in large pictures of program packages in various disciplines of engineering

#### **3.2 CONCEPT OF A FINITE ELEMENT**

The finite element method is based upon the general principle known as going from part to whole .In engineering, problems may come which cannot be solved in closed form, that is as a whole. Therefore, we consider the physical medium as an assemblage of many small parts. Analysis of the basic part forms the first step towards a solution.



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This notion which in mathematical rather than physical, does not consider the body or the structure to be subdivided into separate parts that are re-assembled in the analysis procedure. Instead of that the continuum is zoned into regions by imaginary lines or planes inscribed on the body.

Using this concept, variational-procedure is applied in the analysis of the continuum by assuming a patchwork of solution or displacement models each of which applies to a single region.

The first decision the engineer must take is to select the shape or configuration of the basic element to be used in the analysis. This choice depends upon the geometry of the body or structure and also upon the number of independent space co-ordinates necessary to describe the problem. The finite element usually has a simple 1-D, 2-D or 3-D configuration.

# 4. MODELLING OF CONNECTING ROD INPUT DATA

Configuration of the engine to which the connecting rod belongs.

Crankshaft radius	48.5 mm
Connecting rod length	141.014 mm
Piston diameter	86 mm
Mass of the piston assembly	0.434 kg
Mass of the connecting rod	0.439 kg
Izz about the center of gravity	0.00144 kg M2
Distance of C.G. from crank end	
center	36.44 mm
Maximum gas pressure	37.29 Bar

Properties of connecting rod material.

Material Property	Unit	Scalar Value
Modulus of Elasticity	GPa	206.7
Poisson's Ratio	Unitless	0.30
Mass Density	kg/m <sup>3</sup>	7820

#### FE MODELING OF THE CONNECTING ROD

This chapter discusses geometry of connecting rod used for FEA, its generation, simplifications and accuracy.

Mesh generation and its convergence is discussed. The load application, particularly the distribution at the contact area, factors that decide load distribution, the calculation of the pressure constants depending on the magnitude of the resultant force, application of the restraints and validation of the FEA model are also discussed. Three FEM were used to determine structural behavior under three different conditions, namely, static load condition (static FEA).

#### **GEOMETRY OF THE CONNECTING ROD**

The connecting rod was digitized using a coordinate measuring machine. A solid model of the connecting rod, as shown in Figure was generated using CATIA the bolt-holes have been eliminated. The ross section of the connecting rod from failed components reveals that the connecting rod, as manufactured, is not perfectly symmetric. In the case of one connecting rod, the degree of non-symmetry in the shank region, when comparing the areas on either side of the axis of symmetry perpendicular to the connecting rod length and along the web, was about 5%. This non-symmetry is not the design intent and is produced as a Manufacturing variation. Therefore, the connecting rod has been modelled as a symmetric component. The connecting rod weight as measured on a weighing scale is 465.9 grams. The difference in weight between the weight of the solid model used for FEA and the actual component when corrected for bolt head weight is less than 1%.

This is an indication of the accuracy of the solid model.



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# MODEL

## Static FEA

Finite element mesh was generated using 3-D Structural Solid SOLID45 elements with various element lengths of 2.5 mm (8176 elements)



FIG MODEL OF CONNECTING ROD IN CATIA

# BOUNDARY CONDITIONS Loading

# Static FEA

The crank and piston pin ends are assumed to have a sinusoidal distributed loading over the contact surface area, under tensile loading, as shown in the Figure This is based on experimental results (Webster et al. 1983). The normal pressure on the contact surface is given by:

$$p = po \cos \Theta$$

The load is distributed over an angle of 1800. The total resultant load is given by:

$$P_t = \int_{\frac{-\pi}{2}}^{\frac{\pi}{2}} p_o(\cos^2\Theta) rtd\Theta = p_o rt\frac{\pi}{2}$$

Figure describes r, t and  $\Theta$ . The normal pressure constant po is, therefore, given by:

$$p_{o} = P_{t} / (r t \pi / 2)$$

The tensile load acting on the connecting rod, Pt, can be obtained using the expression from the force analysis of the slider crank mechanism. For compressive loading of the connecting rod, the crank and the piston pin ends are assumed to have a uniformly distributed loading through 1200 contact surface, as shown in Figure 3.9 (Webster et al. 1983). The normal pressure is given by:

$$\mathbf{p}=\mathbf{p}_{\mathrm{o}}\left(3.4\right)$$

The total resultant load is given by: The normal pressure constant is then given by:  $P_{i} = \frac{1}{2} \left( \frac{1}{2} + \frac{1}{2} \right)$ 

$$\mathbf{p}_{\rm o} = \mathbf{P}_{\rm c} \,/\,(\mathbf{r} \,\mathbf{t} \,\sqrt{3})$$

Pc can be obtained from the indicator diagram, such as the one shown in Figure of an engine. In this study four finite element models were analyzed. FEA for both tensile and compressive loads were conducted. Two cases were analyzed for each case, one with load applied at the crank end and restrained at the piston pin end, and the other with load applied at the piston pin end and restrained at the crank end. In the analysis carried out, the axial load was 26.7 kN (6 kips) in both tension and compression. The pressure constants for 26.7 kN are as follows:

#### **Compressive Loading:**

Crank End: po = 26700/ (24 x 17.056 x  $\sqrt{3}$ ) = 37.66 MPa Piston pin End: po = 26700/ (11.97 x 18.402 x  $\sqrt{3}$ ) = 69.98 MPa

#### **Tensile Loading:**

Crank End: po = 26700/ [24 x 17.056 x  $(\pi/2)$ ] = 41.5 MPa

Piston pin End: po = 26700/ [11.97 x 18.402 x ( $\pi$ /2)] = 77.17 Mpa

Since the analysis is linear elastic, for static analysis the stress, displacement and strain are proportional to the magnitude of the load. Therefore, the obtained results from FEA readily apply to other elastic load cases by using proportional scaling factor.



Tensile loading of the connecting rod (Webster et al. 1983).

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Compressive loading of the connecting rod (Webster et al. 1983).

### **RESULTS & DISCUSSION:**

# CASE-1: COMPERSSIVE LOAD ACTING ON CRANK SIDE

Load is applied on the on the big end of the connecting rod

Boundary condition is applied on pin end (gudgeon pin) of the connecting rod



#### **LOAD & BOUNDARY CONDITION**



## DEFORMATION OF CONNECTING ROD DUE TO THE COMPRESSIVE LOAD ACTING ON CRANK SIDE

Maximum deformation is 0.177128 mm



## VON-MISES STRESS DUE TO THE COMPRESSIVE LOAD ACTING ON CRANK SIDE

Maximum stress is 355.545 N/mm2 Minimum stress is 0.585078 N/mm2



VON-MISES STRAIN DUE TO THE COMPRESSIVE LOAD ACTING ON CRANK SIDE

Maximum strain is 0.423E-05 Minimum strain is 0.002288

#### **CASE-2: COMPERSSIVE LOAD ON PIN END**

Load is applied on the on the pin end (gudgeon pin) of the connecting rod

Boundary condition is applied on big end of the connecting rod



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## LOAD & BOUNDARY CONDITION



# DEFORMATION DUE TO THE COMPRESSIVE LOAD ACTING ON PIN END

Maximum deformation is 0.109952 mm



VON-MISES- STRESS DUE TO THE COMPRESSIVE LOAD ACTING ON PIN END Maximum stress is 323.869 N/mm2 Minimum stress is 0.418468 N/mm2



# VON-MISES STRAIN DUE TO THE COMPRESSIVE LOAD ACTING ON PIN END

Maximum strain is 0.001569 Minimu m strain is 0.238E-05

## CASE 3: TANGENTIAL LOAD ON CRANK END

Tangential (Tensile) load is applied on the crank (big) end of the connecting rod



#### LOAD & BOUNDRY CONDITION



# DEFORMATION DUE TO TANGENTIAL LOAD ACTING ON CRANK END

Maximum deformation is 0.211723 mm



# VON-MISES STRESS DUE TO TANGENTIAL LOAD ACTING ON CRANK END

Maximum stress is 781.609 N/mm2 Minimum stress is 2.14456 N/mm2



# VON-MISES STRAIN DUE TO TANGENTIAL LOAD ACTING ON CRANK END

Maximum strain is 0.00408 Minimum strain is 0.244E-04

#### **CASE-4: TANGENTIAL LOAD ON PIN END**

Tangential load is applied on the pin end (gudgeon pin) of the connecting rod

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## LOAD & BOUNDRY CONDTIONS



# DEFORMATION DUE TO TANGENTIAL LOAD ACTING ON PIN END

Maximum deformation is 0.185451 mm



# VON-MISES STRESS DUE TO TANGENTIAL LOAD ACTING ON PIN END

Maximum stress is 818.678 N/mm2 Minimum stress is 0.83181 N/mm2



# 0), for any given crank angle $\theta$ , the orientation of the connecting rod is given by:

# $O \xrightarrow{\mathbf{r}_1} \Theta \xrightarrow{\mathbf{r}_2} \mathbf{r}_0$

The analytical vector approach has been discussed. With reference to Figure below, for the case of zero offset (e =

5.17 Vector representation of slider-crank mechanism.

Free body diagram and vector representation. (a) Free body diagram of connecting rod. (b) Free body diagram of piston

 $\beta = \sin -1\{-r1 \sin \theta / r2\}$ 

Angular velocity of the connecting rod is given by the expression:

 $\omega 2 = \omega 2 k$ 

 $\omega 2 = -\omega 1 \cos \theta / [(r2/r1)2 - \sin 2\theta] 0.5$ 

Note that bold letters represent vector quantities. The angular acceleration of the connecting rod is given by:

# VON-MISES STRAIN DUE TO TANGENTIAL LOAD ACTING ON PIN END

Maximum strain is 0.005327 Minimum strain is 0.586E-05

## **COMPRESSION FOR 4 TYPES:**

**ANALYTICAL APPROACH:** 

S.NO	CASE	DEFORMATION (mm)	VON- MISSES STRESS (N/MM2)
1	COMPERSSIVE LOAD ON CRANKSIDE	0.177	355.54
2	COMPERSSIVE LOAD ON PIN END	0.1099	323.86
3	TANGENTIAL LOAD ON CRANK END	0.211	781.609
4	TANGENTIAL LOAD ON PIN END	0.185	818.67



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$$\label{eq:alpha2} \begin{split} \alpha 2 &= \alpha 2 \ k \\ \alpha 2 &= (1/\cos\beta) \ [ \ \omega 12 \ (r1/r2) \ sin\theta \ - \omega 22 \ sin\beta \ ] \end{split}$$

Absolute acceleration of any point on the connecting rod is given by the following equation:

a =  $(-r1 \ \omega 12\cos\theta - \omega 22 \ u \ \cos\beta - \alpha 2 \ u \ \sin\beta) \ i + (-r1 \ \omega 12\sin\theta - \omega 22 \ u \ \sin\beta + \alpha 2 \ u \ \cos\beta) \ j$ 

Acceleration of the piston is given by:

 $ap=(-\omega 12 r1 \cos\theta - \omega 22 r2 \cos\beta - \alpha 2 r2 \sin\beta) i + (-\omega 12 r1 \sin\theta - \omega 22 r2 \sin\beta + \alpha 2 r2 \cos\beta) j$ 

Forces acting on the connecting rod and the piston are shown in Figure 2.2.

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 mc is the mass of the connecting rod. Forces at the piston pin and crank ends in X and Y directions are given by:

FBX = -(mpaP + Gas Load)

FAX = mc ac.gX - FBX

 $FBY = [mc \ ac.gY \ u \ cos\beta - mc \ ac.gX \ u \ sin\beta + Izz \ \alpha2 + FBX \ r2 \ sin\beta] / (r2 \ cos\beta)$  $FAY = mc \ ac.gY - FBY$ 

#### Details of 'slider-crank mechanism

	Crank		Rod AB
	OA	Connecting	Slider B
Calculated Mass			
(kg)	0.0243	0.0477	0.0156
Calculated			
Volume (mm3)	3116	6116	2000

IXX (kg-mm2)	0.12	0.24	0.26
IYY (kg-mm2)	21.9	165.4	0.65
IZZ(kg-mm2)	21.9	165.4	0.65
IXY (kg-mm2)	0	0	0
IZX (kg-mm2)	0	0	0
IYZ (kg-mm2)	0	0	0

Length (mm)	100	200	20
Width (mm)	5	5	10
Depth (mm)	6	6	10

Crankshaft radius	48.5 mm
Connecting rod length	141.014 mm
Piston diameter	86 mm
Mass of the piston	
assembly	0.434 kg
Mass of the connecting rod	0.439 kg
Izz about the center of	
gravity	0.00144 kg m2
Distance of C.G. from	
crank end center	36.44 mm
Maximum gas pressure	37.29 Bar

## RESULTS OF FINITE ELEMENT STRESS ANALYSIS

The load analysis was carried out to obtain the loads acting on the connecting rod at any given time in the loading cycle and to perform FEA. Most investigators have used static axial loads for the design and analysis of connecting rods. However, lately, some investigators have used inertia loads (axial load varying along the length) during the design process. A comparison between the two is needed and is discussed in this chapter. Connecting rods are predominantly tested under axial fatigue loading, as it was the case for the connecting rod investigated in this project (Afzal, 2004).

The maximum and minimum static loads can simulate the fatigue testing range. As a result, FEA was carried out under axial static load with no dynamic/inertia loads. In order to capture the structural behaviour of the connecting rod under service operating condition, quasi dynamic FEA was also performed. Quasi-dynamic FEA results differ from the static

FEA results due to time varying inertia load of the connecting rod which is responsible for inducing bending stresses and varying axial load along the length.

The results of the above mentioned analyses are presented and discussed in this chapter with a view to use them for optimization. This chapter discusses the stress-time history at critical locations, selection of load



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or the loads under which the connecting rod should be optimized, comparison of the quasi-dynamic with static stress analysis results and obtaining the bending stress magnitude and load ratios.

#### **CONCLUSION:**

This project Design of steel forged connecting rods. The connecting rod chosen for this project belonged to a mid size sedan. First, the connecting rod load calculated as per analytical Approach. Load analysis was performed based on the input from specification, which comprised of the crank radius, piston diameter, the piston assembly mass, and the pressure-crank angle diagram, using analytical techniques. FEA was then performed using the results from load analysis to gain insight on the structural behaviour of the connecting rod. The following conclusions can be drawn from this study:

1) There is considerable difference in the structural behaviour of the connecting rod between axial fatigue loading and dynamic loading (service operating condition). There are also differences in the analytical results obtained from fatigue loading simulated by applying loads directly to the connecting rod and from fatigue loading with the pins and interferences modelled.

2) Bending stresses were significant and should be accounted for. Tensile bending stresses were about 16% of the stress amplitude (entire operating range) at the start of crank end transition and about 19% of the stress amplitude (entire operating range) at the shank centre. Bending stresses were negligible at the piston pin end. The tress ratio varies from -0.14 at the extreme end of the connecting rod cap to -1.95 at the crank end transition, under service operating conditions considering the entire load range.

3) The stress multi axially is high (the transverse component is 30% of the axial component), especially at the critical region of the crank end transition. Therefore, multi axially fatigue analysis is needed to determine fatigue strength. Due to proportional loading, equivalent stress approach based on von Misses criterion can be used to compute the equivalent stress amplitude. The load is 26.7 kn in all the cases.

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