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Design and Static Structural Analysis of Crank Shaft

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Abstract: The stress analyses of a single-cylinder crankshaft are discussed using finite element method in this paper. Three-dimension models of single crankshaft and crank throw were created using Pro/ENGINEER software The finite element analysis (FEM) software ANSYS was used to analyze the distortion and stress status of the crank throw. The maximum deformation, maximum stress point and dangerous areas are found by the stress analysis of crank throw. The results would provide a valuable theoretical foundation for the optimization and improvement of engine design.

Keywords: Crankshaft, Finite Element Analysis, Modal Analysis, Stress Analysis.

I. INTRODUCTION

Crankshaft (i.e. a shaft with a crank) is a central component of any internal combustion engine and is used to convert reciprocating motion of the piston into rotatory motion or vice versa.. The crankshaft main journals rotate in a set of supporting bearings ("main bearings"), causing the offset rod journals to rotate in a circular path around the main journal centers, the diameter of which is twice the offset of the rod journals. The diameter of that path is the engine "stroke": the distance the piston moves up and down in its cylinder as shown in Fig.1. The big ends of the connecting rods ("conrods") contain bearings ("rod bearings") which ride on the offset rod journals. With the development of computer, more and more design of crankshaft has been utilized finite element method (FME) to calculate the stress of crankshaft. The application of numerical simulation for the designing crankshaft helped engineers to efficiently improve the process development avoiding the cost and limitations of compiling a database of real world parts.

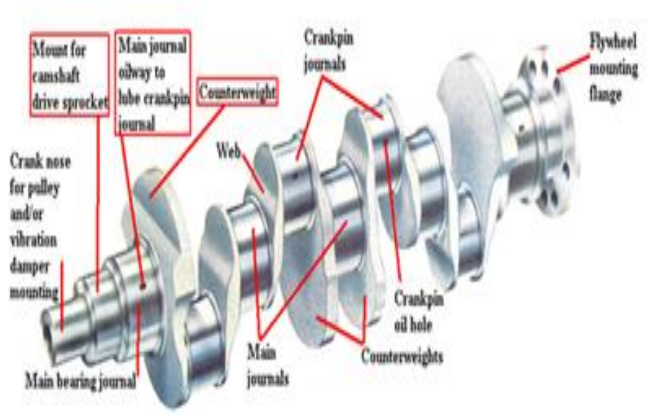


Fig.1. Crankshaft Nomenclature.

In contemporary racing crankshaft design, the requirements for bending and torsional stiffness (see the Stiffness vs. Strength sidebar) compete with the need for low mass moment of inertia (MMOI). Several crankshaft experts emphasized the fact that exotic metallurgy is no substitute for proper design, and there's little point in switching to exotics if there is no fatigue problem to be solved. High stiffness is a benefit because it increases the torsional resonant frequency of the crankshaft, and because it reduces bending deflection of the bearing journals. Journal Deflection can cause increased friction by disturbing the hydrodynamic film at critical points, and can cause loss of lubrication because of increased leakage through the greater radial clearances that occur when a journal's axis is not parallel to the bearing axis The crankshaft, depending upon the position of crank, may be divided into the following two types:1. Side crankshaft or overhung crankshaft, as shown in Fig 2.(a), and 2. Centre crankshaft, as shown in Fig.2 (b).

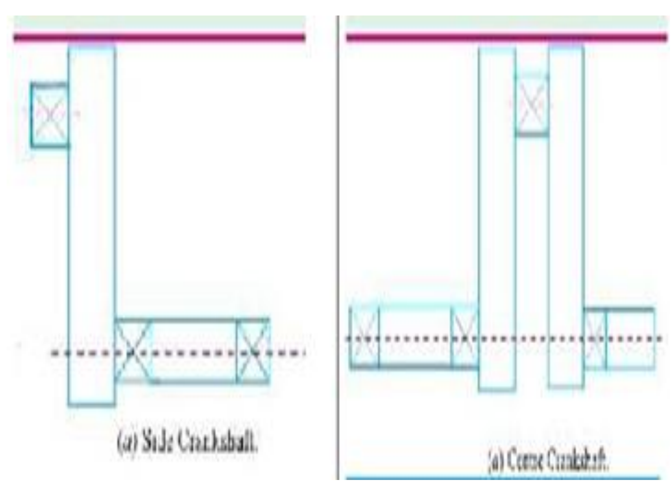


Fig.2. Position of Crankshaft.

The crankshafts are subjected to shock and fatigue loads. Thus material of the crankshaft should be tough and fatigue resistant. The crankshafts are generally made of carbon steel, special steel or special cast iron. The crankshafts are made by drop forging or casting process but the former method is more common. The surface of the crankpin is hardened by case carburizing, nitriding or induction hardening. In this report we will be concentrating upon the design of crankshaft used in Atul Shakti Limited for Mini Tempo. The engine runs on Single cylinders. The detailed parameters of the engine are mentioned in design chapter.

II. DESIGN OF CRANK SHAFT

A. Forces Imposed On A Crankshaft

We have selected engine is Combustion Ignition Diesel Engine. The obvious source of forces applied to a crankshaft is the product of combustion chamber pressure acting on the top of the piston. High-performance, contemporary high-performance Compression-Ignition (CI) engines can see combustion pressures in excess of 200 bar (2900 psi) which will produce a force of 16529 Kg acting on a small 4.00 inch diameter piston. This kind of huge force exerted onto a crankshaft rod journal which produces substantial bending and torsional moments and the resulting tensile, compressive and shear stresses. However, there is another major source of forces imposed on a crankshaft, namely Piston Acceleration. The combined weight of the piston, ring package, wristpin, retainers, the con-rod small end and a small amount of oil are being continuously accelerated from rest to very high velocity and back to rest twice each crankshaft revolution. Since the force it takes to accelerate an object is proportional to the weight of the object times the acceleration (as long as the mass of the object is constant), many of the significant forces exerted on those reciprocating components, as well as on the con-rod beam and big-end, crankshaft, crankshaft, bearings, and engine block are directly related to piston acceleration. Combustion forces and piston acceleration are also the main source of external vibration produced by an engine. In addition to these reciprocating forces and the resulting moments, there is a rotating mass associated with each crankpin, which must be counteracted. The rotating mass consists of the weight of the con-rod big end(s), conrod bearing(s), some amount of oil, and the mass of the crankshaft structure comprising the crankpin and cheeks. These rotating forces are counteracted by counterweight masses located in appropriate angular locations opposing the rod journals

B. Material Selection

Medium-carbon steel alloys are composed of predominantly the element iron, and contain a small percentage of carbon (0.25% to 0.45%, described as ‘25 to 45 points’ of carbon), along with combinations of several alloying elements, the mix of which has been carefully designed in order to produce specific qualities in the target alloy, including hardenability, nitridability, surface and core hardness, ultimate tensile strength, yield strength, endurance limit(fatigue strength), ductility, impact resistance, corrosion resistance, and temper-embrittlement resistance. The alloying elements typically used in these carbon steels are manganese, chromium, molybdenum, nickel, silicon, cobalt, vanadium, and sometimes aluminum and titanium. Each of those elements adds specific properties in a given material. The carbon content is the main determinant of the ultimate strength and hardness to which such an alloy can be heat treated. In converting the linear motion of the piston into rotational motion, crankshafts operate under high loads and require high strength.

Crankshafts require the following characteristics,

- High strength and stiffness to withstand the high loads in modern engines, and to offer opportunities for downsizing and weight reduction.
- Resistance to fatigue in torsion and bending.
- Low vibration.
- Resistance to wear in the bearing areas.

Thus the forged steel crankshafts offer higher strength and stiffness and the other material characteristics than the cast iron alternative.

C. Chemical Composition of Crankshft

The material selected for Atul Shakti Engine single cylinder crankshaft is 40Cr4Mo3. The detailed composition of material is as below mentioned in Table II,

TABLE I: Chemical Composition

Content	%
C	.48 - .52
P	0.03
Ni	0.2
Sn	0.2
B	.0003 - .0005
Mn	.75 - .90
S	.020 - .030
Mo	0.05
Al	.018 - .025
Si	.15 - .35
Cr	.15 - .25
Cu	.20 Max
V	.005 Max

Mechanical Properties:

Hardenability : 1.5mm – 56 to 63 HRC
5mm – 45 to 55 HRC
30mm – 20-29 HRC

Tensile Strength : 83 Kg/mm2

Yield Strength : 55 Kg/mm2

% Elongation : 14 % Min

Micro-Structure: Uniform tempered martensite with transferred ferrite content up to 10% Max. at core.

D. Design Parameters Assumed

Refer below specification in Table II details taken from Atul Shakti Ltd website to design crank shaft.

TABLE II: Crank Shaft Specification

TYPE	Four Stroke ,Air-cooled Diesel
Bore/Stoke, mm	86/68
No. Of Cylinders	1(One)
Displacement Ratio	395CC
Compression Ratio	18.01
Max Power	8.1hp @3600 r.p.m.
Max. Torque	16.7 Nm @ 2200-2800 r.p.m.
Air Cleaner	Dry Type
Oil Filter	Spin On
Fuel Filter	Paper Bement Type
Cooling System	Air Cooled
Oil Sump Capacity	1.4 Liter
Type Of Vehicle	Mini Tempo

E. Design Procedure

Based on the chemical composition of the material we will now design the crankshaft dimensions. Thus the design of crankshaft is to be made by considering the two position of crank. When the crank is at dead center (or when the crankshaft is subjected to maximum bending moment. When the crank angle is at which twisting moment is maximum.

When Crank Is At Dead Center: At this point of crank, the maximum gas pressure on the piston will transmit maximum force on the crankpin in the plane of the crank causing only bending of the shaft. The crankpin as we as ends of the crankshaft will only be subjected to bending moment. Thus, when crank is at dead center, the bending moment on the shaft is maximum and twisting moment is zero. The various forces that are acting on the crankshaft are indicated as below Fig.3. This engine crankshaft is a single throw and two bearing shaft is located 1 and 2. We can calculate the various forces acting on the crankshaft, connecting rod (F_p), Horizontal and vertical reaction on the shaft, and the resultant force at bearing 1 & 2 by below formulae.

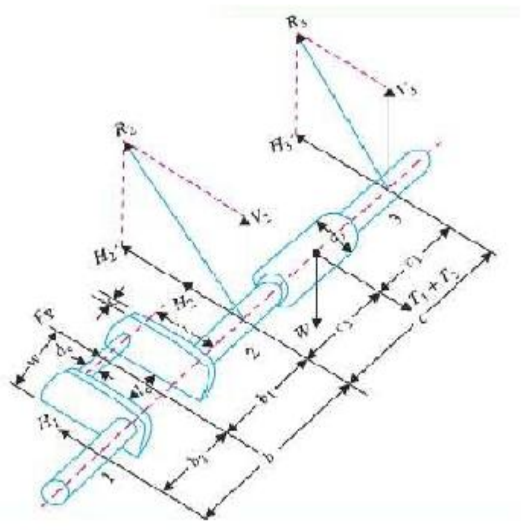


Fig.3. Crank at dead centre.

In the scenario we have not consider bearing reaction calculation at 3. We have given more focus for bearing reaction at 1 and 2 since true data is not available to calculate the reaction.

Now force on Piston,

F_p = Area of the bore x Max. Combustion pressure

$= \pi/4 \text{ x } D_2 \text{ x } P_{max}$

(Where P_{max} = 25 bar = 2.5N/mm2)

$= \pi/4 \text{ x } (86)2 \text{ x } 2.5$

F_p = 14.52KN

Assuming the distance between the bearing 1 & 2 as

b = 2D = 2*69.6 = 13902 mm

$b_1 = b_2 = b/2 = 69.6$

We know that due to piston gas load, there will be two equal horizontal reactions H_1 & H_2 at bearings 1 & 2 respectively..i.e

$$H_1 = F_p/2 = 167.66/2 = 83.83 \text{ kN} = H_2$$

Assuming that the length of bearing to be equali.e. $c_1=c_2=c/2$

F. Design of Crank Pin

Crankpin is also subjected to shear stress due to twisting moment. Thus we can calculate bending moment at centre of crankpin and twisting moment on crank pin and the resultant moment.

Let,

d_c = Diameter of crank pin in mm

l_c = length of crank pin in mm

σ_{allow} = allowable bearing stress for crank pin = 83kg/mm2

Bending moment at the centre of crank pin is,

$M_c = H_1 * b_2 = 83.83 * 69.6 = 5834.53$ KN.mm

We know that

$M_c = \pi/32 * (d_c)3 * \sigma_b$

$5834.56 \text{ x } 103 = \pi/32 * (d_c)3 * 83d_c$

= 89.46 mm say 90 mm

Now, the length of crank pin $l_c = F_p/(d_c * p_b)$

= 167.67 x 103 / (90 * 10) -- (say p_b =10)

= 186.28 mm

G. Design of Left Hand Crank Web

The crank web is designed for eccentric loading. There will be two stresses acting on the crank web, one is direct compressive stress and the other is bending stress due to piston gas load (F_p). The crank web is subjected to the following stresses:

- Bending stresses in two planes normal to each other
- Direct compressive stress and
- Torsional stress

We know that the thickness of crank web is

t = 0.65 *dc + 6.35

= 0.65* 90 + 6.35

= 64.85 = say 65 mm

Also width of crank web is,

w = 1.125 * dc +12.7

= 1.125 * 90 +12.7

= 113.95 = say 115 mm

The maximum bending moment on crank web is

M_{max} = H1 (b_2 – l_c/2 - t/2)

= 83.83 (69.6- 186.28/2-65/2)

= - 4697.83 kN mm

The bending moment is negative; hence the design is not safe. Thus the dimensions are on higher side.

Now let's assume,

d_c = 45 mm

Hence, l_c = 372.57 mm

This is very high, which will require huge length of crankshaft. To have optimum dimension of crankshaft; lets assume length of crank web as,

L_c = 24mm and check whether these dimension are suitable for the load exerted by the piston and other forces,

Now,

T = 35.6 &

W = 63.32 = say 68 mm

This thickness is also on higher side, let's assume thickness of crank web as

T =13.23 mm

As compared to width of crank web thickness is more Bending moment,

M =4275.33 KN.mm

Section Modulus

Z = 1/6 * w * t2

= 1/6 * 68 *13.2^2

Z = 1974.72 mm^3

Bending Stress ób = M/Z

ób= 2.165 KN/mm2

The compressive stress acting on crank web are

óc = H_1/(w * t)

= 83.33/(68* 13.2)

= .0939 KN/mm2

The total stress on crank web is less than allowable bending stress of 83 N/mm2.

Hence, Design is safe

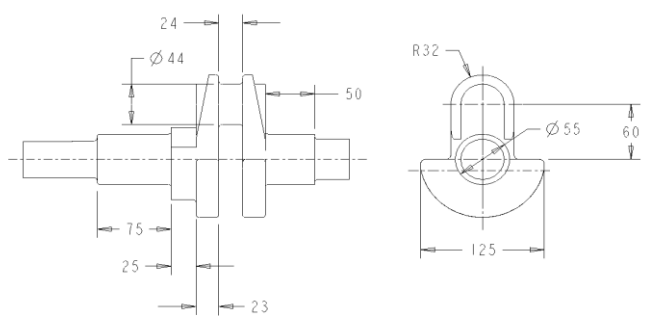
H. Design of Right Hand Crank Web

From balancing point of view, the dimension of right hand crank web i.e. thickness and width are made equal to the dimension of the left hand crank web. When the crank is at an angle of maximum twisting moment; the twisting moment on the crankshaft will be maximum when the tangential force on the crank (FT) is maximum. The maximum value of tangential force lies when the crank is at angle 30º to 40º for constant pressure combustion engines (i.e. diesel engines). When the crank is at angle at which the twisting moment is maximum, the shaft is subjected to twisting moment from energy or force stored by flywheel. The above design parameters can be cross checked for the factor of safety while designing by considering the crankshaft at an angle of maximum twisting moment. If the factor of safety is more than 1 then the design is safe. Considering this, we have to various forces acting on crankshaft at different twisting angles.

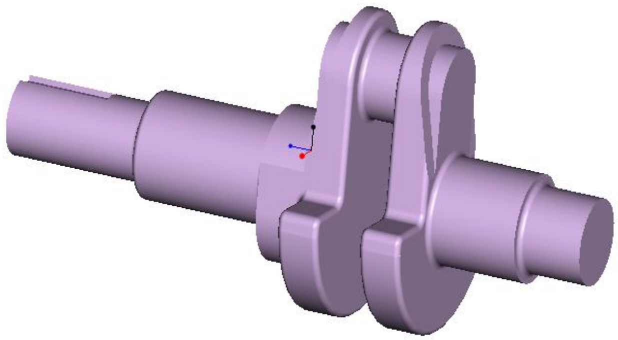
III. CRANKSHAFT MODELLING AND ANALYSIS

A. Crankshaft Drawing/Model

As per calculation, we have created the Proe model of the crankshaft. The final dimensions of the crankshaft are as given below Fig.4,



(a)



(b)

Fig.4. 2D&3D of Crankshafts.

B. Crankshaft FEA Static Structural Analysis

Introduction: For FEA analysis we have used ANSYS workbench as shown in Fig.5.

Input for FEA: Pro-e Solid model OR Parasol file of Crankshaft.

Material properties:

Material Type: Forged Steel

Designation: 40CrMo4

Yield strength (MPa): 680

Ultimate tensile strength (MPa): 850

Elongation (%):13

Poisson ratio:0.3

Pre-processor: Meshing Details of the Crankshaft.

Mesh Statics:

Type of Element: Tetrahedrons

Number of Nodes: 56279

Number of Elements: 44106

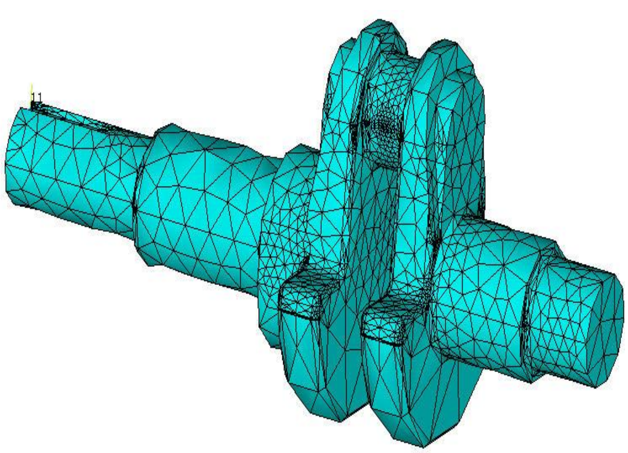
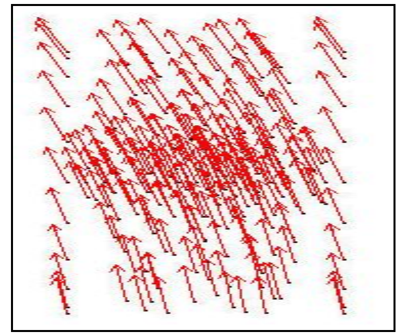


Fig.5. FEA analysis of crankshaft.

Loading on Crankshaft:

Below are radial Loads (F_y) = 10.68N (Load is applied at Modes of geometry)



Below are Tangential Loads (F_z) = 10.05N

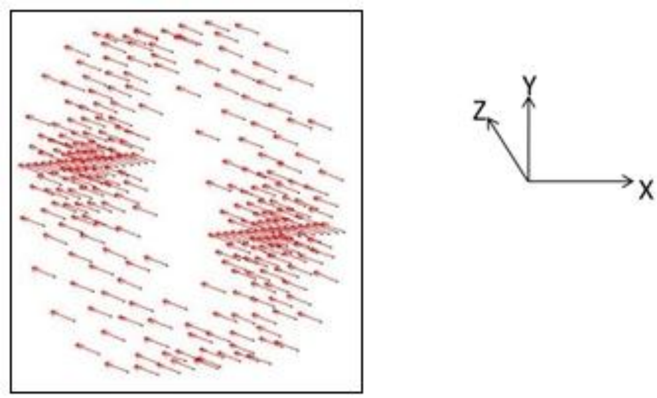


Fig.6. Radial load & Tangential load.

Radial load & Tangential load is as shown in Fig.6.

FEA Results: Represent two views of crankshaft in order to view magnitude of stress (Fig. 7).

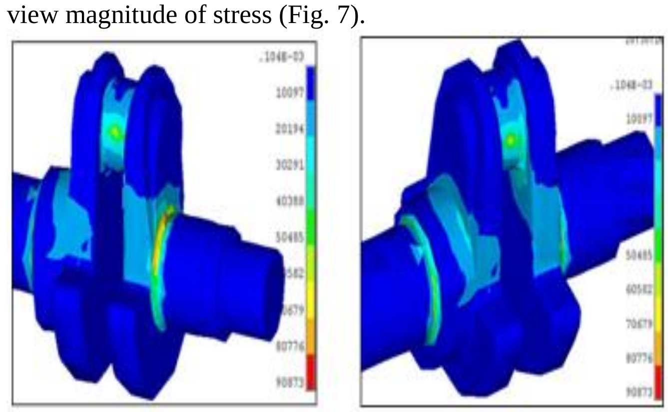


Fig.7. FEA Results: Magnitude of stress.

Interpretation:

- Max stress occurs at the joints besides the bearing region.
- Introducing a fillet will help reduce the stresses.

IV. CONCLUSION

The stresses are well below the YS of the material – safe to use the component. Refer below table for stress analysis results of analytical and FEA method.

TABLE III: FEA Results

Type of Stress (N/mm2)	Theoretical	FEA Analysis
von-Mises	112.25	114.73
Shear Stress	50.15	57.50

Above Results Shows that FEA Results matches with the theoretical calculation so we can say that FEA is a good tool to reduce time consuming theoretical Work. The maximum deformation appears at the center of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks and near the central point Journal. The edge of main journal is high stress area. The Value of Von-Misses Stresses that comes out from the analysis is far less than material yield stress so our design is safe and we should go for optimization to reduce the material and cost.

V. ACKNOWLEDGMENT

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VI. REFERENCES

[1] Atul Shakti Engine Limited; http://www .atulauto.co .in/ site/vehicle/atul-shakti- smart/pickup-van-standard.html.
[2] C.Gaier and H.Dannbauer – "A Multi axial fatigue analysis method for Ductile, semiductile and brittle materials" The Arabian Journal for Science and Engineering, Volume 33, Number 1B - 18 March 2006.
[3] M.Zoroufi and Fatemi– "A Literature review on durability evaluation of Crankshafts including comparisons of competing manufacturing Processes" Vol. 12, pp. 38-53.
[4] Yu Gongzh.i, Yu Hongliang., Duan Shulin., 2011, "Crankshaft Dynamic Strength Analysis for Marine Diesel Engine,"Third International Conference on Measuring Technology and Mechatronics Automation.
[5] Machine Design by Sri R S Khurmi and Sri. J K Gupta, Chapter-32 Internal Combustion Engine parts.
[6] http://knol.google.com].