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OPTIMIZATION OF CRANKSHAFT BY MODIFICATION IN DESIGN AND MATERIAL

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Abstract – Crankshaft is an important component of the internal combustion engine. *Due to huge loading and high number of fatigue cycles, crankshaft is prone to early damage and hence reducing the engine life. In this research paper, static structural analysis is performed on crankshaft for a 4-cylinder inline SI engine. A three- dimensional model of crankshaft is designed using SIEMENS NX 12.0 software. In order to study the effect of loading, Finite Element Analysis (FEA) is performed on ANSYS 18.1 software by applying load and constraints to the shaft according to engine working conditions. The analysis is performed for locating critical failure in crankshaft. The optimization includes the modification in the geometry of crankshaft which results in safe and efficient design of the crankshaft. The review of work on the crankshaft optimization and design is demonstrated. The materials, failure analysis, design parameters of the crankshaft analyzed here.*

***Key Words*: SI Engine, Crankshaft, Static Structural Analysis, Fatigue, ANSYS 18.1, SIEMENS NX 12.0, FEA.**

1.INTRODUCTION

Crankshaft is one of the key components in working of the internal combustion engine. It is moving component in the engine, which converts the reciprocating motion of the piston into a rotary motion. It consists of shaft parts, journals bearings and a crankpin bearing. This study was conducted on a 4-cylinder inline SI engine so the crankshaft should have capacity to take the maximum downward force during power stroke without excessive bending. Because life and durability of IC engine depend on the strength of the crankshaft. As the engine start functioning, the impulses hit the crankshaft. For smooth functioning of the engine, crankshaft should be designed and optimized to provide efficient working of engine. The geometry of a crankshaft is complicated, and function of its rotational position the varies load. Because of the dissimilar area of cross-sections of the crankshaft, it will be more dangerous to continue because the stiffness discontinuities will be causing maximum stress concentration. Mostly, the crankshaft fillet will be the highly stressed area and will require accuracy as the materials used to make crankshafts are typically sensitive to notch factors. In this research the fillet medication and the material used in the crankshaft will provide more safety to the engine, considering the results obtained by optimizing the crankshaft shows the safe and efficient working with the increase in the life of crankshaft. In the static structural analysis, the stress concentration of the crankshaft is described by Xiaorong Zhou et al. [1]. In the fillet of spindle neck the stress is mostly occurred and there is high stress at crankpin fillet. On the basis of stress analysis, the maximum strength can be achieved as per requirement. In this paper the analysis is carried out to obtain the maximum strength of crankshaft.

C.M. Balamurugan et.al [2] had designed the model in SOLID EDGE and further done analysis in ANSYS. In their paper comparison with two different material i.e. ductile cast iron and forged steel which increases the fatigue life. The optimization included geometry change with the current engine, fillet rolling which deducted the cost of the crankshaft. Analysis results obtained while the crankshaft subjected under static load having stresses and deformation.

J. Meng et.al [3] in their research paper designed crankshaft model in Pro-E software and analysis done in the ANSYS. The difference between the frequency and vibration modal is described by FEA analysis. This analysis resulted in theoretical foundation for optimization of engine design and by appropriate design the resonance vibration can be avoided and safety can be increased.

R. J. Deshbhratar et.al [4] have designed crankshaft by using Pro-E and analysis was done by ANSYS software. The maximum stress is seen in the fillet area and also around the center of crank pin. At most of the times crankshaft deforms because of bending under natural frequency as the maximum deformation is observed on crankpin neck.

Rinkle garg et.al [5] in their paper on crankshaft model which was designed in Pro-E software and used ANSYS software for simulation. It states that increase in strength of crankshaft reduces the maximum stress, strain and deformation. Which makes the engine more efficient in the working and the cost effective.

In the literature survey Mahesh L. Raotole et.al [6] have researched the life duration of crankshaft using FEA. They created model using MATLAB and used ANSYS for simulation. By carrying out dynamic analysis at various engine speed and at fillet radius, conclusion was that the crankshaft failure occurs in fillet region. So, designer should take into consideration that the fatigue life as most important factor of design.

K. Thriveni et.al [7] have performed static analysis on 4-stroke IC engine. Using CATIA modelling was done and analysis performed in ANSYS. They observed that maximum stress and deformation is at the neck of crankpin. And mostly maximum stress is developed at the fillet area.In all the literature review fatigue life is considered as important factor of design hence in this paper optimization is made to increase the life and safety of component.

2. DESIGN FOR CRANKSHAFT

2.1 Material Selection

AISI 4140 cold worked steel also known as 42Cr4Mo2 (EN 19) steel. AISI 4140 is known as chromium-molybdenum alloy steel. Chromium provides good hardness, where the molybdenum has uniform hardness and good strength. The properties of AISI 4140 consist of high toughness, wear resistance and good ductility in the quenched condition and tempered condition. The AISI 4140 cold finished annealed alloy steel can be heated using different methods to obtain a various property, hence it is mostly used for forging because of its self-scaling properties. Crankshafts need more strength and stiffness to carry the loads in modern engines. Thus, the cold worked forged steel crankshafts offer higher strength and stiffness and the other material characteristics than the cast iron alternative.

2.2 Chemical Composition of Crankshaft

The material selected for crankshaft is AISI 4140 cold worked steel also known as 42Cr4Mo2 (EN 19) steel. The detailed composition of material is as below mentioned in Table 1

Table -1: Chemical Composition

Content	Percentage
C	0.35 - 0.45
Mn	0.5 - 0.8
Si	0.10 - 0.35
Ni	0 - 0.02

Mechanical Properties:

Hardness: BHN = 241 HRC: 22

Yield tensile strength: 485 MPa

Ultimate Tensile strength: 814 MPa

Elongation at breaking: 22.2 %.

2.3 Design Parameters Assumed

Below given specification in TABLE II used to design the crankshaft. The specifications are based on Honda City (2015 sedan).

Table -2: Vehicle Specification

Type	4-cylinder inline SI engine
Bore	73 mm
Stroke	89.4 mm
Net displacement ratio	1497 cc
Torque	145 Nm @ 4600 rpm

2.4 Design Procedure

Considering the chemical composition, design of the crankshaft is performed using theoretical calculations. The crankshaft design is made by considering following such as the position of crank. When the crank is having maximum bending moment, when the crank angle is at which twisting moment is maximum.

a) When Crank Is at the Dead Center.

At this point, the bending of shaft caused due to transmitted maximum force on crankpin in plane of crank by maximum gas pressure load of piston. The crankpin ends of the crankshaft will only be subjected to bending moment. Thus, when crank is at dead center, the twisting moment is negligible and the bending moment is maximum. The various forces that are acting on the crankshaft are indicated as below Fig 1

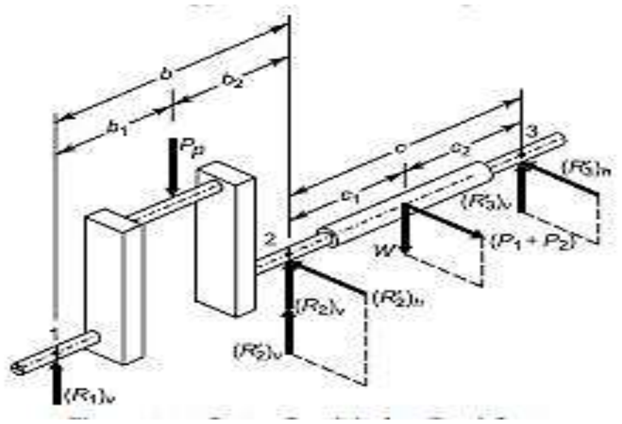

Fig 1: Crankshaft at dead Centre

Let, d = Piston diameter (mm), p = Maximum intensity of pressure N/mm²

we know gas load on piston

F_p = Area of bore x Max. Combustion pressure

$= \pi/4 \times d2 \times P_{max}$

(Where P_{max} = 65 bar = 6.5N/mm2)

F_p = 27.2 kN

Since, crankshaft is symmetrical and inline so force on bearing shaft,

$R1 = R2 = F_P/2 = 13.6$kN

b = 2D = 2 x 73

= 146 mm

b1 = b2 = b/2 = 73mm

b) Design of Crank Pin

Crankpin is subjected to shear stress due to twisting moment. Therefore, we are able to calculate the bending moment at center of crankpin and also twisting moment on crank pin .

Let, dc = Crank pin Diameter (mm)

Lc = Crank pin Length (mm)

σ_{allow} = Allowable bearing stress = 841 Mpa

Bending moment at the centre of crank pin is,

Mb = R1 x b1 = 992.982 Nm

We know that Mb = π/32 x (dc)2 x σb /FOS

dc = 35 mm

Now, the length of crank pin

lc = Fp / (dc x Pb)

= 25 mm

c) Design of Crank Web

The crank web is designed for eccentric loading. There are two stresses on the crank web, i.e. direct compressive stress and the bending stress.

w is width of crank web; t is thickness of web

from design data handbook

t = 0.45 x dc + 6.35

= 22.5 = say 25 mm

Also, width of crank web is,

w = 1.125 x dc +12.7

= 76mm = say 84 mm

L_w = 150mm (from design data handbook)

Right and Left side of web will have same dimensions due to symmetry of loading.

d) Stresses Induced (von mises)

According to theory of distortion energy, the Von-Misses stress in the crank-pin is,

$M_{ev} = \sqrt{((K_b \times M_o)^2 + 3/(4) \times (K_t \times T_o)^2)}$

Where, K_b = 2 (combined shock and fatigue factor for bending)

and K_t = 1.5 (shock and fatigue factor for torsion)

$M_{ev} = \pi/32 \times (dc)^3 \times \sigma v$

σ_v = 230.73 MPa

$T_{ev} = \pi/16 \times (dc)^3 \times \tau$

τ = 137.69 MPa

3. DESIGN METHODOLOGY

3.1 Procedure of static Analysis

Modelling of crankshaft was performed in Siemens NX 12.0. According to the calculated dimensions from the theoretical steps, the crankshaft was modelled. Fillet of 5mm was provided at the joints of the spindle and web for even stress – strain distribution.

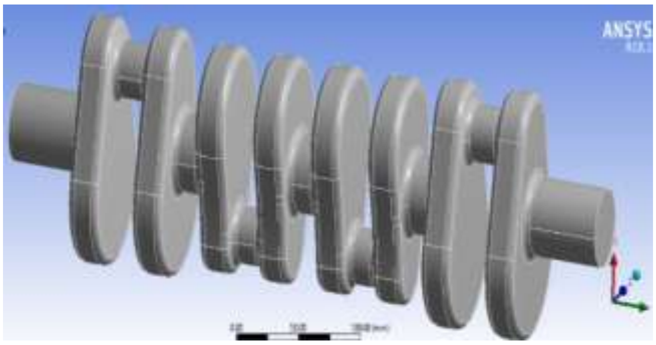

Fig -2: Crankshaft geometry

3.2 Pre- processing details

Before performing analysis on solver, it is necessary to pre process the model to get precise and authentic results. Since the geometry does not have highly complicated structures, hence a geometry cleans up was not conducted. Meshing is an important parameter which decides the quality of results obtained

Meshing of Crankshaft
Type of element: - Quadrilateral Dominant
Order of element: - Quadratic
Number of nodes: - 158272
Number of elements: - 46661
Quadrilateral dominant elements are selected because of the skewness offered. Since quadrilateral elements offer proper deformation under substantial loading, these elements are used in meshing to obtain proper results.

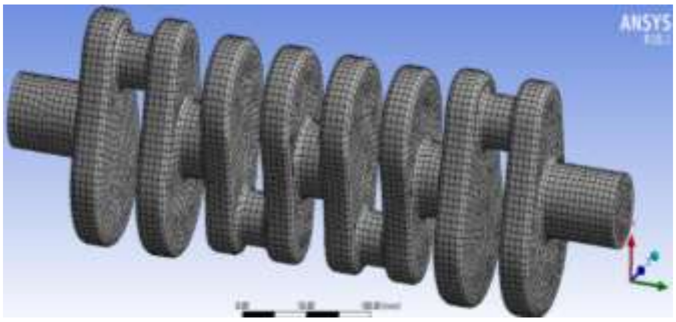

Fig -3: Meshed Model

3.3 Boundary conditions and load

Frictionless support is provided to the crankshaft according to the engine working condition. During power stroke, combustion take place in combustion chamber which causes crankshaft to rotate. The force is transferred from connecting rod to crankpin and load of 30000 N acting on crankshaft is shown in Fig. 4

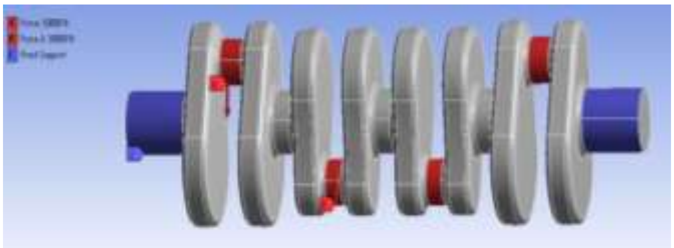

Fig -4: Loads on crankshaft

4. Results and Discussion

On applying boundary conditions and loads on the pre-processed model, following results were obtained.

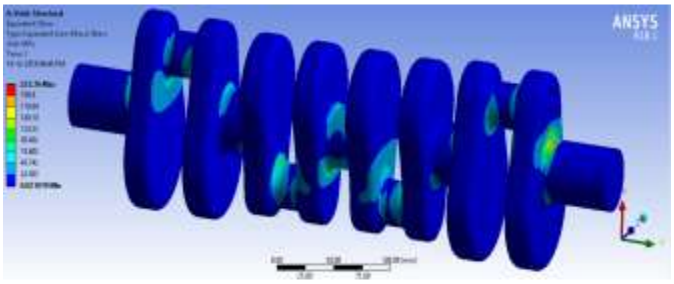

Fig -5: Von mises stress

Fig. 5 shows Von Mises Stress occurring in the crankshaft. It is observed that maximum amount of shear stress is at the spindle and web joint which is 223.76 MPa.

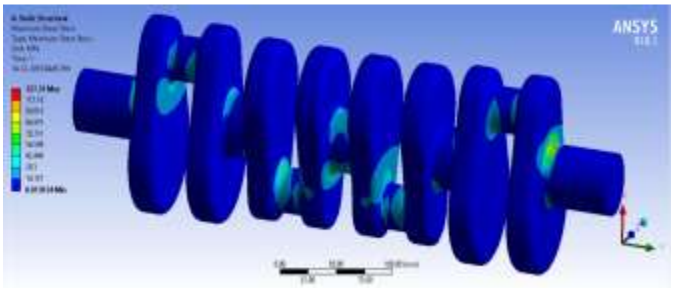

Fig -6: Shear stress

In Fig 6, shear stress occurring in the crankshaft is reduced. In this case too, it is observed that maximum shear stress is occurring at joints between spindle and the crank web and it is found out to be 127.31 MPa.

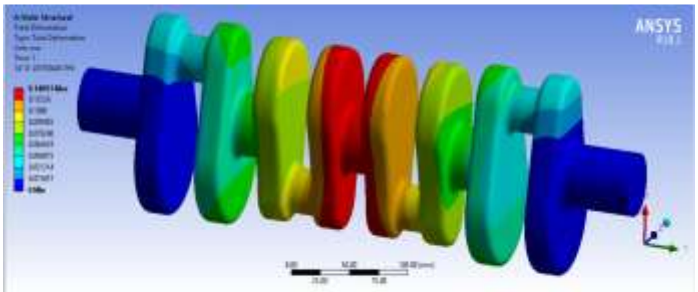

Fig -7: Total Deformation

Fig 7 represents the total deformation occurring in the crankshaft under the given maximum loading. The deformation occurring is very minimum and occurs between the central spindle joining the intermediate webs. The maximum deformation is 0.14091 mm.

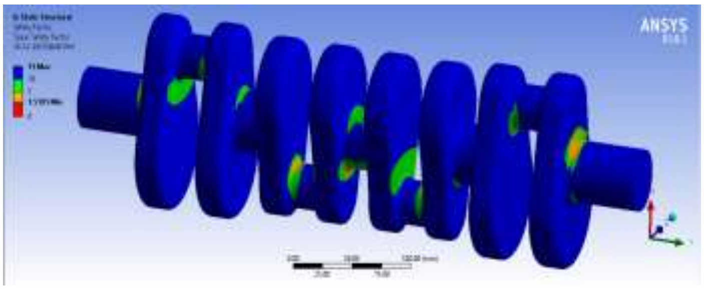

Fig -8: Factor of safety

Fig 8 depicts the Factor of Safety (FOS) obtained in the crankshaft under fatigue loading. Since reverse cyclic loads are imposed on crankshaft, it is important to calculate fatigue strength of crankshaft. The FOS calculated is for infinite life cycles and is found out to be 1.5195. The least value of FOS defines the total FOS of the component.

Table -3: RESULT TABLE

Stresses	Theoretical (MP_a)	FEA (MP_a)
Von mises	230.73	223.76
shear	137.69	127.31

The comparison between the theoretical and FEA results clearly state that the stress obtained are very close and the optimization result in the less stress to developed and the property of material induces less shear stress. Due to this the working become more efficient and the safety of machine is thus increased.

5. CONCLUSIONS

The Von mises stress in the crankshaft is far more less than that of yield stress of the material. Shear stress in the crankpin is also within permissible limits. Thus, it can be concluded that design of crankshaft considered to be safe under the loading conditions.

Deformation of the crankshaft is 0.15 mm which is too negligible in rotating motion of the crankshaft. Since the deformation is negligible, crankshaft leads to smooth running and increasing the engine life expectancy and better power delivery.

Due to fillets at the end of crankpin and the crank web, stress is evenly distributed which leads to less stress concentration at the joints. Hence providing fillet will lead to a safer design than that of without fillet joining. The Safety Factor (FOS) is found to be 1.5, which results in assurance of safety of the crankshaft in cases of uneven and more loading than that of ideal loading.

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