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Design and Analysis of Crankshaft for Internal Combustion Engine

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ABSTRACT

In this project design and analysis of the crankshaft for the combustion engine. These components have a large volume component with complex geometry and need huge investment. These will be converts reciprocating or linear motion of the piston into a rotary motion. In this project the product is modeled in a 3D model with all available constraint by using advanced cad software CATIA-V5. this model will be converted to initial graphics exchange specification (IGES) format and imported to ANSYS workbench to perform static analysis. Finite element analysis (FEA) is performed to obtain the various stress and critical location of crankshaft under loads by using ANSYS software. This project helps to many researchers to select best material to production of crankshaft.

KEYWORDS: *Crankshaft, CATIA-V5, Initial Graphics Exchange Specification (IGES), ANSYS workbench and. Finite element analysis (FEA)*

I. INTRODUCTION

Internal combustion engine plays important in automobile, pumping, electric generation. marine etc. the obvious piston force applied to a crank shaft is the product of combustion pressure acting on the top of the piston. Many high-performance crankshafts are formed by the forging process. In this production process billet of suitable size is heated to appropriate forging temperature ranges from 1950˚F-2250˚F and then successively pounded or pressed the desire shape by squeezing the billet between pairs of dies under high pressure [1].

However, there will have major source of forces imposed on a crankshaft, namely Piston acceleration. The combined weight of the piston, ring package, wristpin, retainers, the connecting 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 connecting rod beam and big-end, crankshaft, crankshaft, bearings, and engine block are directly related to piston acceleration. The methods for dealing with those vibratory loads are covered in a dedicated article. Combustion forces and piston acceleration are also the main source of external vibration produced by an engine [2].

The steel alloys typically used in high strength crankshafts havebeen selected for what each designer perceives as the most desirable combination of properties. 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, nitride ability, 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 [3].

Many researchers have focused on design of the crankshaft. According to **Farzin H. Montazersadgh** and **Ali Fatemi's** journal dynamic simulation was acted on a crankshaft from a multi cylinder four stroke engine. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the engine specification chart. The dynamic analysis was done analytically and was verified by simulation in ADAMS which resulted in the load spectrum applied to crank pin bearing. This load was applied to the FE model in ANSYS, and boundary conditions were applied according to the engine mounting conditions. **Payer** et al. developed a two-step technique to perform nonlinear transient analysis of crankshafts combining a beam-mass model and a solid element model. Using FEA, two major steps were used to calculate the transient stress behavior of the crankshaft; the first step calculated time dependent deformations by a step-by- step integration using the new mark-beta- method. Using a rotating beam-mass- model of the crankshaft, a time dependent nonlinear oil film model [5]

In the second step those transient deformations were enforced to a solid- element- model of the crankshaft to determine its time dependent stress behavior. The major advantage of using the two steps was reduction of CPU time for calculations. This is because the number of degrees of freedom for performing step one was low and therefore enabled an efficient solution. Furthermore, the stiffness matrix of the solid element model for step two needed only to be built up once Literature survey is concluded, and we can move into next chapter to discussabout designing procedure. **Guagliano** et al. conducted a study on a marine diesel engine crankshaft, in which two different FE models were investigated. Due to memory limitations in meshing a three-dimensional model was difficult and costly. Therefore, they used a bi-dimensional model to obtain the stress concentration factor which resulted in an accuracy of less than 6.9 percent error for a centered load and 8.6 percent error for an eccentric load. This numerical model was satisfactory since it was very fast and had good agreement with experimental results [6].

II. Methodology

In the design and analysis of crankshaft project following steps to design crankshaft which bear high impact load piston steps are given below are

- Calculating forces on the crankshaft mathematically.
- Model the crankshaft in CATIA-V5.
- Save model file in IGES format.
- Export in the ANSYS software.
- Using Analysis module by inserting material and loads on the crankshaft.

III. Calculation of Forces

At this position of the 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 well as ends of the crankshaft will be only subjected to bending moment. Thus, when the crank is at the dead Centre, the bending moment on the shaft is max. And the twisting moment is zero. The various forces that are acting on the crankshaft are indicated as below. This engine crankshaft is a single throw and three bearing shaft located at position 1,2&3. Let's us assume following data for engine and calculate the various forces acting on crank shaft connecting rod (Fp), Horizontal and vertical reactions on shaft, and the resultant force at bearing 2 & 3 by below formulae. Now the piston force

P_{max} = P × no of cylinders/1248 × 10^6 × 4000
= 55 × 4/1248 × 10^6 × 4000
= 44.07

Piston force Fp =π/4 × D^2 × P_{max}
=π/4 × $(69.6)^2$ × 44.07
= 167.67 KN

Assuming the distance between the bearings 1 &2 as
b = 2D=2 × 69.6=13902mm b1
= b2 = 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 = Fp/2 = 167.66/2 = 83.83 KN = H_2

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

we know that due to weight of flywheel acting downwards, there will be two vertical reactions V2 & V3 at bearings 2 & 3
V_2= V_1 = W/2 = 9.8/2 = 4.9 N

Since, the belt is absent in engine, neglecting the belt tension exerted by belt
i.e. T_1 + T_2 = 0

Now, let's design various parts of crankshaft

- 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 (Fp).

The crank web is subjected to the following stresses

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

We know that the thickness of crank web is t = 0.65 *d_c + 6.35= 0.65 × 90 + 6.35
= 64.85 = say65 mm

Also, width of crank web is,
W = 1.125 × d_c +12.7
= 1.125 x 90 +12.7
= 113.95 = say115 mm

The maximum bending moment on crank web is
M_{max} = H1 (b2 –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 crank shaft. To have optimum dimension of crank shaft let's assume length of crank web as.
l_c = 24 mm
and check whether these dimensions are suitable for the load exerted by the piston, & 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 = 13.2 mm

As compared to width of crank web thickness is more
Bending moment, M = 4275.33 kN-mm

Section modulus, $Z = 1/6 \times w \times t^2$

$= 1/6 \times 68 \times 13.2^2$

$= 1974.72\ mm^3$

Bending stress

$\sigma_b = M/Z$

$\sigma_b = 2.165\ KN/mm^2$

The compressive stress acting on crank web are

$\sigma_c = H_1 / (w \times t)$

$= 83.83 / (68 \times 13.2)$

$= 0.09339\ KN/mm^2$

A. The total stress acting on crank web is

$\sigma_T = \sigma_b + \sigma_c$

$= 2.2583\ KN/mm^2$

Thus, total stress on crank web is less than allowable bending stress of 83 N/mm^2
Hence, the design is safe

B. Design of right-hand crank web

From balancing point of view, the dimensions of right-hand crank web i.e. thickness and width are made equal to the dimensions of left-hand crank web.

C. Design of shaft under flywheel

There are two types of bending moments acting on shaft. Bending moment due to weight &, bending moment due to belt tension. Neglecting the belt tension lets design shaft diameter.

Let, d_s = diameter of crank shaft Since the length of bearings are equal $l_1 = l_2 = l_3$

$= 2(b/2 - l_c/2 - t)$

$= 2(139.2/2 - 24/2 - 13.2)$

= 88.8 mm

Assuming the width of flywheel = 200 mm

C = 88.88 + 200

= 288.88 mm

Considering the space for gearing and clearance,
Let C = 300 mm

Bending moment due to weight of fly wheel,

$M_b = V_3 \times C$

$= 4.9 \times 10^3 \times 300$

$= 1470 \times 10^3\ KN\ mm$

Also, the bending moment of shaft is

$M_s = \pi/32 \times d_s^3 \times \sigma\ allow$

$1470 \times 10^3 = \pi/32 \times d_s^3 \times 83\ d_s$

= 56.50mm

IV. MODELLING OF CRANKSHAFT

In this project modeling crankshaft in CATIA V5 as shown in figure 2. Creation of a 3-D model in CatiaV5R20 can be performed using three workbenches i.e.- sketcher- modeling and assembly. Sketcher is used two-dimensional representations of profiles associated within the part. We can create a rough outline of curves- and then specify conditions called constraints to define the shapes more precisely and capture our design intent. Each curve is referred to as a sketch object. a new sketch- chose Start Mechanical design part then select the reference plane or sketch plane in which the sketch is to be created. The sketch plane is the plane that the sketch is located on. The sketch plane menu has the following options:

Face/Plane: With this option- we can use the attachment face/plane icon to select a planar face or existing datum plane. If we select a datum plane- we can use the reverse direction button to reverse the direction of the normal to the plane.

XC-YC- YC-ZC- and ZC-XC: With these options- we can create a sketch on one of the WCS planes. If we use this method- a datum plane and two datum axes are created as below.

Displays the structure of the part, assembly, or drawing. Select an item from the feature manager design tree to edit the underlying sketch, edit the feature, and suppress and un suppress the feature or component, for example. A meeting is an aggregate of or extra components, additionally known as components, inside one solid works record. Your role and orient components the use of mates that form family members among additives.

- This lesson discusses the following:
- Adding components to a meeting
- Transferring and rotating additives in an assembly
- Growing display states in an assembly

"Feature" is an all-encompassing term that refers to all solids, bodies and primitives used in Solid works Form Features are used to supply detail to the model in the form of standard feature types. These include hole, Extrude Boss/Cut, Swept Boss/Cut, Fillet. We can also create our own custom features using the User Defined option. All of these features are associative.

Reference Feature sallow creating reference planes, reference lines and reference points. These references can assist in creating features on cylinders, cones, spheres and revolved solid bodies. Reference planes can also aid in creating features at angles other than normal to the faces of a target solid. Dress up Feature options lets modify existing solid bodies and features.

These include a wide assortment of options such as edge fillet, variable fillet, chamfers, draft, offset face, shell and tapers. Surface design lets us create surface and solid bodies. A surface body with zero thickness, and consists of a collection of faces and edges that do not close up to enclose a volume.

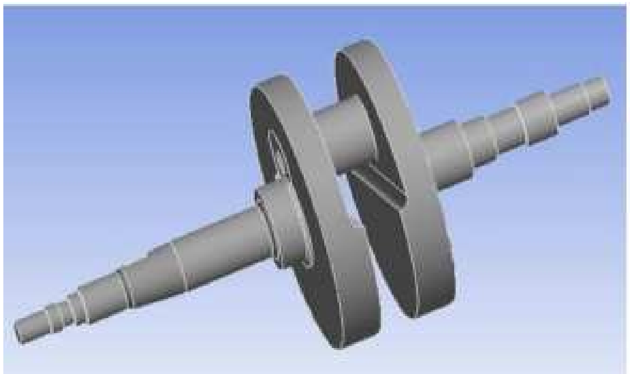

Fig. 1. Final Model of Crankshaft

V. Results and Discussion

1. Total Deformations

In the Details of "Total Deformation" window, expand the Results node, if it is not already expanded. Note that the maximum and minimum deformations displayed are respectively.

For Medium Steel Alloy

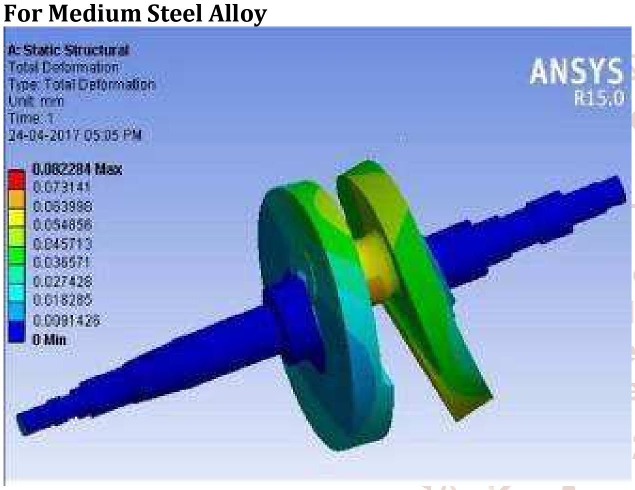

Fig.2. The Values of Total Deformation Obtained from The Legend Display in Color Bands.

For Ni Cr Mo Steel Alloy

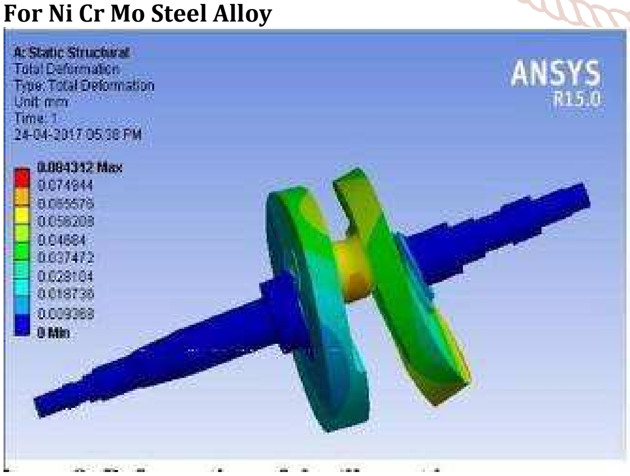

Fig. 3. The Values of Total Deformation Obtained from The Legend Display in Color Bands.

2. Equivalent Stress

In the Details of "equivalent stress" window, expand the Results node, if it is not already expanded. Note that the maximum and minimum deformations displayed are respectively

For Medium Steel Alloy

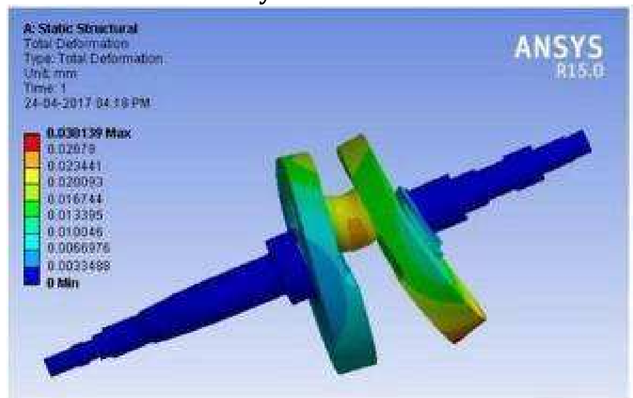

Fig.4. The Values of Equivalent Stress Obtained from The Legend Display in Color Bands.

For Ni Cr Mo Steel Alloy

Fig. 5. The Values of Equivalent Stress Obtained from The Legend Display in Color Bands

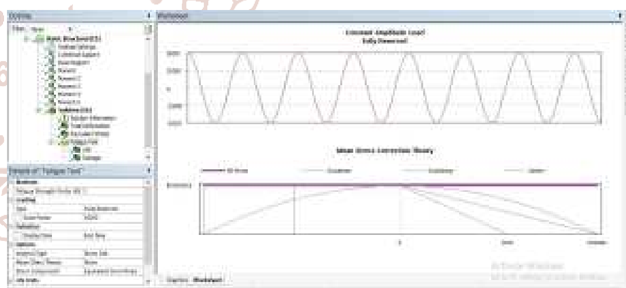

Fig. 6. Strain Life Graphs

Therefore, there is significant difference in the stresses and deflection in springs under the same static load

Life

In the Details of "life" window, expand the Results node, if it is not already expanded. Note that the maximum and minimum deformations displayed are respectively.

Damage

In the Details of "damage" window, expand the Results node, if it is not already expanded. Note that the maximum and minimum deformations displayed are respectively.

VI. Conclusion

- The Model weight was reduced after change of materials from 10.93 kg to 9.773 kg at a density Kg/m^3 from the table 7.1 and 7.2.
- From the above results it is suggest that the design modification was acceptable.

- In the working of the engine, the engine generates 10 to 12 torque at beginning it may vary by increase of acceleration. Because of continuously cycle or stokes the crankshaft gets deforms. At a peek of cycles, the crankshaft is unable to generate smooth transmission of the power to rear wheel.
- In this project we consider the composite material in real time MEDIUM STEEL ALLOY and replace the material with other material name NI CR MO STEEL ALLOY so that the weight and deformations are decreased and life increase.
- by analysis the life of the crankshaft is increased.
- weight was reduced

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