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Experimental And Numerical Analysis Of Crankshaft Used In Hero Honda Splendor Motorcycle

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Abstract—**In the project, 3-D finite element analysis was carried out on the modal analysis of crankshaft and the stress analysis of crankshaft to check the safety. The FEM software ANSYS workbench was used to simulate the analysis of crankshaft. The results of stress and deformation distributions and natural frequency of crankshaft were obtained by using ANSYS software. The experimental investigation also carried out for modal part and it validates with the FEM results.**

Index Terms- **Crankshaft, ANSYS, FEM results.**

I. Introduction

Crankshaft is among the largest components in internal combustion engines. It is employed in different types of engines, from small one cylinder lawn-mowers to large multi-cylinder diesel and petrol engines. Crankshaft is one of the most critically loaded components and experiences cyclic loads in the form of bending and torsion during its service life. This review is performed for optimization of crankshaft by considering various materials used for manufacturing of crankshaft, various manufacturing process used for manufacturing of crankshaft and opportunities available for optimization by various geometric changes in shape of crankshaft. Amongst all materials used for manufacturing of crankshaft the best material is selected and is manufactured by method which is most suitable and will reduce the cost of production. Crankshaft is one of the most important moving parts in internal combustion engine. It must be strong enough to take the downward force of the power stroked without excessive bending. So the reliability and life of internal combustion engine depend on the strength of the crankshaft largely. And as the engine runs, the power impulses hit the crankshaft in one place and then another. The torsion vibration appears when a power impulse hits a crankpin toward the front of the engine and the power stroke ends. If not controlled, it can break the crankshaft.

Strength calculation of crankshaft becomes a key factor to ensure the life of engine. Beam and space frame model were used to calculate the stress of crankshaft usually in the past. But the number of node is limited in these models. With the development of computer, more and more design of crankshaft has been utilized finite element method (FEM) 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. Finite element analysis allows an inexpensive study of arbitrary combinations of input parameters including design parameters and process conditions to be investigated.

Crankshaft is a complicated continuous elastomeric. The vibration performance of crankshaft has important effect to engine. The calculation of crankshaft vibration performance is difficult because of the complexity of crankshaft structure, the difficult determinacy of boundary condition. Dynamic matrix method and dynamic sub structural method combined with FEM were used to calculate the vibration of crankshaft. The method of three-dimensional finite element was carried to analysis dynamical characteristic of diesel crankshaft.

In the project, 3-D finite element analysis was carried out on the modal analysis of crankshaft and the stress analysis of crankshaft to check the safety. The FEM software ANSYS workbench was used to simulate the analysis of crankshaft. The results of stress and deformation distributions and natural frequency of crankshaft were obtained by using ANSYS software. The experimental investigation also carried out for modal part and it validates with the FEM results.

I.I Optimization of Crankshaft

For optimization of crankshaft various studies have been made on material selection and manufacturing process of crankshaft, it is found Crankshafts are typically manufactured from forged steel, nodular cast iron and austempered ductile iron (ADI). When forged steels are compared to cast iron and alloyed ductile iron used in crankshafts, the fatigue properties of forged steels are generally found to be better that that of cast iron. Also it is found that by using micro alloy steel it is possible to reduce the cost of production. Therefore replacement of conventional crankshaft by forged steel shaft will also result in optimization of cost.

Detailed dynamic load and stress analysis of the crankshaft investigated in this study was the subject of another paper. Finite element analysis was used to obtain the variation in stress magnitude at critical locations. The dynamics of the mechanism was solved using analytical techniques, which resulted in the load spectrum applied to the crank pin bearing. The load was applied to the FE model and the boundary conditions were defined according to the engine mount

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design. The analysis was performed over different engine speeds and as a result the critical engine speed and critical locations on the crankshaft were obtained. Stress variation over the engine cycle and the effect of torsion load in the analysis were also investigated. Results from FE analysis were verified.

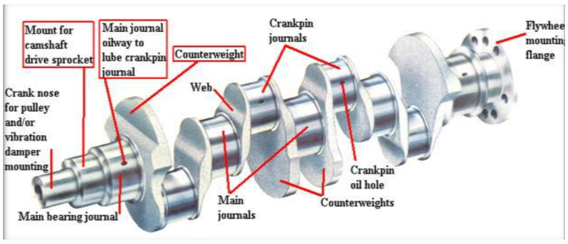

Fig. Generalized Crankshaft in the Engine Block

I.II Reasons of Failures Crankshaft

The most common reasons for crankshaft failures are below:

- Loss of effective lubrication. This can be due to contaminated lube oil, failed lube oil pumps, poor quality or incorrect specification lube oil.
- Over speeding of engines, or long term operation in a critical or forbidden rev range.
- Vibration loads on the crankshaft
- Faulty crankshaft damper, designed to remove excessive vibration from the crankshaft. Failure of proper operation can lead to excessive crankshaft vibration and fatigue.
- Engine power imbalance leading to fatigue failure, cyclic loading. This can be caused by poor maintenance or monitoring of engine power, or even poor quality fuel.
- Hydraulic locking of cylinders, flooding of cylinders with cooling water.
- Bearing misalignment, this can be detected early with proper crankshaft deflection measurement.
- Design faults, a common problem as more licenses are passed out to new shipyards. Incorrect or blatant ignorance of material compositions or poor manufacture of crankshaft can lead to early failure.
- Overloading of engine.

II. LITERATURE REVIEW

C.M.Balamurugan, R.Krishnaraj, Dr.M.Sakthivel evaluates and compares the fatigue performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and ductile cast iron. In this study a dynamic simulation was conducted on two crankshafts, cast iron and forged steel, from similar single cylinder four stroke engines. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The dynamic analysis was done analytically and was verified by simulations in ANSYS. Results achieved from aforementioned analysis were used in optimization of the forged steel crankshaft. Geometry, material and manufacturing processes were optimized considering different constraints, manufacturing feasibility and cost. The optimization process includes geometry changes compatible with the current engine, fillet rolling and result in increased fatigue strength and reduced cost of the crankshaft, without changing connecting rod and engine block.[1]

Farzin H. Montazersadgh says in his paper that finite element analysis was performed to obtain the variation of the stress magnitude at critical locations. The dynamic analysis resulted in the development of the load spectrum applied to the crankpin bearing. This load was then applied to the FE model and boundary conditions were applied according to the engine mounting conditions. Results obtained from the aforementioned analysis were then used in optimization of the forged steel crankshaft. Geometry, material, and manufacturing processes were optimized using different geometric constraints, manufacturing feasibility, and cost. The first step in the optimization process was weight reduction of the component considering dynamic loading. This required the stress range under dynamic loading not to exceed the magnitude of the stress range in the original crankshaft. Possible weight reduction options and their combinations were considered. The optimization and weight reduction were considered in an interactive manner and evaluated by manufacturing feasibility and cost. The optimization process resulted in an 18% weight reduction, increased fatigue strength, and a reduced cost of the crankshaft. [2]

Jaimin Brahmbhatt, Prof. Abhishek choubey suggested in his paper that a dynamic simulation is conducted on a crankshaft from a single cylinder 4- stroke diesel engine. A three-dimension model of diesel engine crankshaft is created using SOLID WORKS software. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. Simulation inputs are taken from the engine specification chart. The dynamic analysis is done using FEA Software ANSYS which resulted in the load spectrum applied to crank pin bearing. This load is applied to the FE model in ANSYS, and boundary conditions are applied according to the engine mounting conditions. The analysis is done for finding critical location in crankshaft. Stress variation over the engine cycle and the effect of torsion and bending load in the analysis are investigated. Von-Misses stress is calculated using theoretically and FEA software ANSYS. The relationship between the frequency and the vibration modal is explained by the modal and harmonic analysis of crankshaft using FEA software ANSYS. [3]

The optimization options their combination under a set of defined constraints, a comparison between the original forged steel crankshaft and the final optimized forged steel component. The main objective of this analysis was to optimize the weight and manufacturing cost of the forged steel crankshaft, which not only reduces the final production cost of the component, but also results in a lighter weight crankshaft which will increase the fuel efficiency of the engine. Optimization carried out on this component is not the typical mathematical sense of optimization, because variables such as manufacturing and material parameters could not be organized in a mathematical function according to the set of constraints such that the maximum or minimum could be obtained. In this case of optimization process, the final optimized geometry has definitely less weight than the original crankshaft but this does not mean that the weight could not be reduced further. In other words, this may not be the minimum possible weight under the set of constraints defined. As the main objective of this analysis, it was attempted to reduce the weight and final cost of the component by changing the crankpin geometry, Increasing the oil hole diameter and fillet radius, increasing the oil hole depth and changing the crank web geometry. The method describes the first step in the optimization process to reduce weight of the component considering dynamic loading, which means that the stress range under dynamic loading should not exceed the stress range magnitude in the original crankshaft. The optimization process was categorized in different stages and paper concludes with the sufficient reduction in weight and ultimately cost.[4]

Static analysis was conducted on a cast iron crankshaft from a single cylinder four stroke engine. Finite element analysis was performed to obtain the variation of the stress magnitude at critical locations. Three dimensional model of the crankshaft was created in Pro-E software. The load was then applied to the FE model and boundary conditions were applied as per the mounting conditions of the engine in the ANSYS. Results obtained from the analysis were then used in optimization of the cast iron crankshaft. This requires the stress range not to exceed the magnitude of the stress range in the original crankshaft. The optimization process included geometry changes without changing connecting rod and engine block. [5]

The problem occurred in single cylinder engine crank shaft. It consists of static structural and fatigue analysis of single cylinder engine crank shaft. It identifies and solves the problem by using the modeling and simulation techniques. The topic was chosen because of increasing interest in higher payloads, lower weight, higher efficiency and shorter load cycles in crankshaft. The main work was to model the crank shaft with dimensions and then simulate the crank shaft for static structural and fatigue analysis. The modeling software used is PRO-E wildfire 4.0 for modeling the crank shaft. The analysis software ANSYS will be used for structural and fatigue analysis of crank shaft for future work. The material for crank shaft is EN9 and other alternate materials on which analysis will be done are SAE 1045, SAE 1137, SAE 3140, and Nickel Cast Iron. The objective involves modeling and analysis of crank shaft, so as to identify the effect of stresses on crank shaft, to compare various materials and to provide possible solution. [6]

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III. Problem Statement

Problem Definition

1. Crankshafts are typically manufactured by casting and forging processes.
2. Manufacturing by forging has the advantage of obtaining a homogeneous part that exhibits less number of micro structural voids and defects compared to casting.
3. In addition, directional properties resulting from the forging process help the part acquire higher toughness and strength in the grain-flow direction.
4. While designing the forging process for crank-shaft, the grain-flow direction can be aligned with the direction of maximum stress that is applied to the component.
5. Crankshaft is a complicated continuous elastomer. The vibration performance of crankshaft has important effect to engine.
6. The calculation of crankshaft vibration performance is difficult because of the complexity of crankshaft structure, the difficult determinacy of boundary condition.
7. We are going to discussed with forged steel crankshaft.
8. We are doing finite element analysis for finding out the optimized product in crankshaft

Objectives

1. To perform complete study of Crankshaft of petrol engine of two wheeler for getting the dimensions, loading conditions etc.
2. To Create CAD - Model of crankshaft, using Pro-E software and FEM analysis by ANSYS Workbench.
3. To analyze the failures occurring in the crankshaft with the help of FEM.
4. To improve hardness of crank pin by providing coating of various material
5. To perform experimental investigation on the original crankshaft for actual loading condition
6. To compare the various results obtained from FEM and experimental tests and suggest the best suitable material for crankshaft.

Steps to Achieve the Above Objectives

1. Take the original crankshaft and collect all the dimensions to prepare CAD model.
2. Convert CAD model into neutral file so that it can be open in FEM software
3. Perform Modal and structural analysis using ANSYS Workbench software for existing and various different materials
4. Perform experiment tests using FFT analyzer and accelerometer.
5. Calculate theoretically the frequency ranges and stresses induced.
6. Compare results for getting best suitable material coat.

IV. Theoretical Analysis

Sample calculation for Modal Analysis for Alloy steel

Formula used is,

$$f\,(\mathrm{Hz}) = \frac{\lambda^2}{2\pi L}\sqrt{(EI/\rho A)}$$

Where

F= Natural Frequency in Hz

λ= Random no. generated using Scientific calculator (shift+ Ran #) = 0.9467

L= Length =0.2m

E= Youngs Modulus = 21000N/m^2

I = Inertia of crankshaft = 2186 X 10^{-3}kgm^3

ρ= Density of Material= 7800Kg/m^3

A = Cross Sectional Area =176.17 X 10^{-6} m^2

Putting all above values in equation

F= $(0.9467)^2$/ 2π$(0.2)^2$ X ((21000X 2186X10^{-3})/(7800X176.17X10^{-6}))$^{1/2}$

F= 3.56 X 182.77

F= 652.093 Hz

Similarly others are calculated as given in following table

	Alloy Steel	Mathematical Results for Frequency
Value of λ	0.9467	652.1
	0.9723	687.8
	1.0134	747.2
	1.52364	1689.1
	1.67348	2037.6
	1.72364	2161.6
	1.9587	2791.4
	2.1443	3345.7

	Aluminium Alloy	Mathematical Results for Frequency
Value of λ	0.9189	599.4
	0.957	650.2
	1.014	729.9
	1.524	1648.8
	1.725	2110.5
	1.824	2362.6
	2.175	3360.7
	2.235	3548.0

	Titanium Alloy	Mathematical Results for Frequency
Value of λ	0.92145	603.0
	0.962	665.2
	0.9978	707.1
	1.543	1693.0
	1.59	1803.1
	1.7348	2137.4
	2.015	2880.7
	2.0214	2901.9

	Nickel Alloy	Mathematical Results for Frequency
Value of λ	0.9364	611.7
	0.975	663.1
	1.104	717.2
	1.5343	1642.2
	1.625	1842.0
	1.703	2023.1
	2.014	2829.5
	2.074	3000.6

	Zinc Alloy	Mathematical Results for Frequency
Value of λ	0.9164	160.8
	0.9678	179.3
	1.029	200.7
	1.532	449.4
	1.5852	481.4
	1.779	603.9
	1.961	740.2
	2.152	890.4

Von Misses Stress analysis

For this project, fatigue is the main cause of crankshaft failure considered. Fatigue failure is defined as the tendency of a material to fracture by means of progressive brittle cracking under repeated alternating or cyclic stresses of intensity considerably below the normal strength. Although the fatigue is of a brittle type, it may take some time to propagate, depending on both the intensity and frequency of the stress cycles.

Here in this project, the cyclic stresses play an important role to determine whether the component is safe from fatigue failure. The other criteria used to determine the safety are von Mises stress, deformation and number of life cycles, strain along with the stresses. This problem can be solved analytically by using the expression for fatigue Endurance limit of crankshaft& by the distortion energy method.

Theory:

Here calculations are done in order to obtain values for Gas Force, Bending Moment,Section Modulus and Torque. Finally all these values are used in order to calculate the Von Mises Stresses.

Force on the piston:

Fpmax= Area of the bore ×Max. Combustion pressure

$$F_{pmax} = \frac{\pi}{4} \times D^2 \times P_{max}$$

$$\sin\varphi = \frac{\sin\theta}{L/R}$$

Ө=Maximum crank angle=35°

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

L = Stroke length

R = Crankshaft radius

$$F_Q = \frac{F_P}{\cos\varphi}$$

Tangential force

FT = FQ sin (θ + ϕ)

Radial force

FR = FQ cos (θ + ϕ)

Reactions at bearings (1&2) due to tangential force is given by,

$$H_{T1} = H_{T2} = \frac{F_T}{2}$$

Similarly, reactions at bearings (1&2) due to radial force is given by,

$H_{R1} = H_{R2} = \frac{F_R}{2}$

Mc = Max. bending moment on crank pin σ_b

Mc $= HR_1 \times b_2$

Tc = Max.twisting moment on crank pin

Tc $= HT_1 \times b_1$

$T_e = \sqrt{[(M_c)^2 + (T_c)^2]}$

Von-misses stress induced in the crankpin

Force on the piston:

Fpmax= Area of the bore ×Max. Combustion pressure

Fpmax = π/4×D²×Pmax

Fpmax= π/4×50^2×3.5

Fpmax=6.86 x E^3 N

Θ=Maximum crank angle=35°

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

L = Stroke length

R = Crankshaft radius

Sinφ=17×sin35/62.4

Φ=8.995°

$F_Q = \frac{F_p}{\cos\varphi}$

=6.86/cos8.995

Instead of 6.86 use 12.339 KN and calculate von miss stress

$F_Q = F_p = \frac{12.339}{\cos\varphi \quad \cos(80995)}$

= 12.55 kN

Tangential force,

$F_T = F_Q \sin(\theta+\phi)$

$= 12.55 \times \sin(35+8.995)$

$= 8.7163$ kN

Radial force,

$F_R = F_Q \cos(\theta+\phi)$

$= 12.55 \times \cos(35+8.995)$

$= 9.0314$ kN

Reactions at bearings (1 & 2) due to tangential force is given by,

$H_{T1} = H_{T2} = \frac{F_T}{2}$

= 8.7163/2

= 4.358 kN

Similarly, reactions at bearings (1&2) due to radial force is given by,

$H_{R1} = H_{R2} = \frac{F_T}{2}$

= 9.031/2

= 4.5155 kN

M_c = Max bending moment on crank pin

$M_c = H_{R1} \times b_2$

$= 4.5155 \times 67$

= 261.899 kN-mm

T_c = Max twisting moment on crank pin

$T_c = H_{T1} \times b_1$

$= 4.358 \times 67$

= 252.764 kN-mm

$T_e = \sqrt{[(M_c)^2 + (T_c)^2]}$

$= \sqrt{[(261.899)^2 + (252.764)^2}$

= 363.97kNmm

$M_{ev} = \frac{\sqrt{(K_b \times M_c)^2 + (3 \times (K_t \times T_c)^2)}}{4}$

$= \frac{\sqrt{(1 \times 261.899)^2 + (3 \times (1\times 252.746)^2)}}{4}$

= 261.899 kN-mm

$M_{ev} = \frac{\pi}{32} \times d^3 \times \sigma_{von}$

$261.899 \times 10^3 = \frac{\pi}{32} \times 67^3 \times \sigma_{von}$

$\sigma_{von} = 13.672$ N/mm^2

OR

From reference (17),

16 mm distance between two flanges of crank shaft where load is act.

Area=πd/2x16 =502mm^2

σxx =piston force/area of crank pin=6860/502=13.66 N/mm2

σyy=0

σzz=0

put this value in following equation

$\sigma_{von} = 1 \sqrt{(\sigma_{xx} - \sigma_{yy})^2 + (\sigma_{xx} - \sigma_{zz})^2 + 6(\sigma^2_{xy} + \sigma^2_{xz} + \sigma^2_{yz})}$

Von missies stress= 18.6 N/mm^2

4.4 Design of Crankshaft When The Crank Is At Dead Centre:

4.4.1 Design of crankpin:

We know that piston gas load (F_P),

Force on the crankshaft $F_P =$ Area of the bore x P_{max}

Force on crankshaft $(F_p) = \frac{\pi}{4} \times D^2 \times P_{max}$

4.5 Design of Crankshaft When The Crank Is At An Angle Of Maximum Twisting Moment

Force on the crankshaft $F_P =$ Area of the bore x P_{max}

Force on crankshaft $(F_p) = \frac{\pi}{4} \times D^2 \times P_{max}$

Fp= 6.860 KN

In order to find the thrust in the connecting rod (F_Q), we should first find out the angle of inclination of the connecting rod with the line of stroke (i.e. angle $\emptyset$).

We know that

$$\sin\emptyset = \frac{\sin\emptyset}{L/R} = \sin 35^0/5$$

Which implies $\emptyset$ = 6.59°

We know that thrust in the connecting rod

$$F_Q = \frac{F_P}{\cos\emptyset}$$

From this we have,

Thrust on the connecting rod F_Q = 6.905 KN

Now shear stress on the crankshaft for which it is safe

Shear stress = Force / Cross sectional Area

= 6.860KN/ 176.17mm^2

= 38.9396 N/mm2

= 38.9396 MPa

This value of shear stress is the limiting value. If the stress induced goes beyond this value for any material then that crankshaft will definitely fail. Hence for FEM structural analysis we need to check the value of stress to be lesser than this for safe conditions.

Sr No.	Alloys	Equivalent stress
1.	Alloy steel	17.461 MPa
2.	Titanium alloy	14.47 MPA
3.	Aluminium alloy	25.997 MPa
4.	Nickel alloy	32.453 MPA
5.	Zinc alloy	6.5241 MPa

V. NUMERICAL ANALYSIS

CATIA V5:

CATIA is very powerful tool. You can harness this power to capture the design intent of your models by acquiring an understanding of fundamental concepts that define the software and why exist. This lesson discusses these concepts in detail. You should keep them in mind as you progress through this chapter.

Design Concepts:

You can design many different types of models in CATIA. However, before you begin your design project, you need to understand a few basic design concepts:

Design Intent:

Before you design your model, you need to identify the design intent. Design intent defines the purpose and function of the finished product based on product specifications or requirements. Capturing design intent builds value and longevity into your products. The key concept is at the core of the CATIA feature based modelling.

Feature-Based Modelling:

CATIA part modelling begins with the creating individual geometric features one after another. These features become interrelated to other features as reference them during the design process.

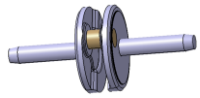
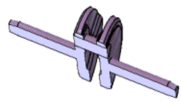
Parametric Design:

The interrelationships between features allow the model to become parametric. So, if you alter one feature and that change directly affects other related (dependent) features, then CATIA dynamically changes those related features. This parametric ability maintains the integrity of the part and preserves your design intent.

Associativity**:**

CATIA maintains design intent outside Part mode through associativity. As you continue to design the model, you can add parts, or electrical wiring. All of these functions are fully associative within CATIA. So, if you change your design at any level, your project will dynamically reflect the changes at levels, preserving design intent.

The assembly consists of crank webs, crankpin and needle bearing. The parts that are mentioned are drawn separately as mentioned above and then with help of various constraints, they are brought together and assembled.

Without Coating	With Coating Thickness 5 to 10 Micron
	
	
Crank Shaft Assembly without Coating	Crank Shaft with Zinc Coated

Crankshaft Materials and Properties of Coating Materials:

The steel alloys typically used in high strength crankshafts have been 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, 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 addition to alloying elements, high strength steels are carefully refined so as to remove as many of the undesirable impurities as possible sulfur, phosphorous, calcium, etc. and to more tightly constrain the tolerances, which define the allowable variations in the percentage of alloying elements. The highest quality steels are usually specified and ordered by reference to their AMS number (Aircraft Material Specification). These specs tightly constrain the chemistry, and the required purity can often only be achieved by melting in a vacuum, then re-melting in a vacuum to further refine the metal. Typical vacuum-processing methods are VIM and VAR. Vacuum Induction Melting (VIM) is a process for producing very high purity steels by melting the materials by induction heating inside a high-vacuum chamber. Vacuum Arc Remelting (VAR) is a refining process in which steels are remelted inside a vacuum chamber to reduce the amount of dissolved gasses in the metal.

Heating is by means of an electric arc between a consumable electrode and the ingot. There are other ultra-high-strength steels that are not carbon steels. These steels, known as "maraging" steels, are refined so as to remove as much of the carbon as possible, and develop their extreme strength and fatigue properties as a by-product of the crystalline structures resulting from the large amounts of nickel (15% and up) and cobalt (6% and up) they contain. These steels can achieve extreme levels of strength and maintain excellent levels of impact resistance. As far as I could determine, maraging alloys are not currently used for racing crankshafts but they have been used in certain extreme application connecting rods. In the high performance crankshaft world, the nickel-chrome-molyalloy SAE-4340 (AMS-6414) has been a favorite in both forged and billet applications. It is used because of its very high strength and fatigue properties, coupled with good ductility and impact resistance at high strengths. SAE-4340 contains a nominal 40 points of carbon and is often described as *the standard to which other ultra-high strength alloys are compared.* There is evidence that lower carbon content provides better impact resistance (reduced notch sensitivity) in certain alloys. The air-hardening nickel-chrome-molyalloy EN-30B is used in some high-end billet crankshafts, in both commercial and VAR forms. This alloy contains 30 points of carbon, and has a nickel content exceeding 4% (400 points). It has good impact resistance at high strengths and is often used in rock-drilling equipment and highly-stressed gears and transmission components. The fact that it can be air quenched to typical crankshaft core hardness is an added advantage because the distortions and residual stresses which result from oil quenching are avoided. Several manufacturers offer billet crankshafts in EN-30B. At least one US manufacturer of extreme duty crankshafts for NASCAR Cup, Top Fuel, Pro-Stock, early IRL, and other venues has selected a high-purity, lower-carbon version of the 43xx series of nickel-chrome-molysteels, a high-grade variant of E- 4330-M (AMS 6427). This material has a nominal 30 points of carbon and has become a favorite for oil drilling and jet engine components because of its very high toughness and impact resistance when heat-treated to high strengths. This manufacturer uses slight variations in the chemistry for different applications, but was understandably reluctant to discuss the variation specifics and how they affected the desired properties. The company maintains tight control over the entire process by purchasing its specific chemistry materials from a single, extreme-quality steel manufacturer, and by doing its heat-treating, cryogenic processing, ion-nitriding and high-tech inspection all in-house. The use of ion-nitriding allows the nitride process to be done subsequent to finish-grinding. The material which is currently viewed as the ultra-extreme crankshaft alloy is a steel available from the French manufacturer Aubert & Duval, known as 32-CrMoV-13 or 32CDV13. It is a deep-nitriding alloy containing 300 points of chrome, developed in the mid-nineties specifically for aerospace bearing applications. It is available in three grades. GKH is the commercial purity and chemistry tolerance. GKH-W is the grade having higher purity (VAR) and tighter chemistry tolerance. GKH-YW is the extremely pure grade (VIM - VAR) and is said to cost twice as much per pound as the -W grade.

According to data supplied by Aubert & Duval, fatigue-tests of the -W and –YW grades, using samples of each grade heat treated to similar values of ultimate tensile strength, show consistently that the -YW grade achieves a dramatic improvement (over 22%) in fatigue strength compared to the -W grade, and the endurance limit is claimed to be just a bit short of the yield stress, which is truly amazing. I have been told that, because of the extreme stress levels on Formula One crankshafts, most of them use the -YW grade, while the lower stress levels of a Cup crank allow the successful use of the -W grade. One well-known manufacturer (Chambon) has developed a process which allows the production of a deep case nitride layer in this alloy (almost 1.0 mm deep, as compared to the more typical 0.10 to 0.15 mm deeplayer). They say this deeper case provides a far less sharp hardness gradient from the >60 HRC surface to the 40-45 HRC core, which improves the fatigue and impact properties of the steel. It says that its deep-case process requires several days in the nitriding ovens, but the depth allows finish-grinding after nitriding, using a very sophisticated process to remove the distortions which occurred during the nitriding soak. No discussion of high-end crankshaft materials would be complete without mention of the ultra-high-strength alloy known as 300-M (AMS 6419). This alloy is a modification to the basic 4340 chemistry, in which a few more points of carbon are added (higher achievable hardness and strength), along with 170 points of silicon and 7 points of vanadium. The vanadium acts as a grain refiner, and the silicon enables the material to be tempered to very high strength (285 ksi) and fatigue properties, while retaining extremely good impact resistance and toughness. This material (300-M) is expensive and sometimes hard to get, since it is preferred for heavy aircraft landing gear components. It has been used by a few manufacturers for extreme duty Crankshafts and connecting rods as well as high-shock aircraft components. However, Several of the manufacturers I spoke with told me that they consider their favorite Materials to be much better than 300-M for crankshaft applications.

Physical properties	**Zinc alloy**	**Titanium alloy**	**Nickel alloy**
Density (G/Cm3)	6.65	4.43	8.89
Hardnes s Rockwell C	34	36	40
Poisson's Ratio (μ)	0.29	0.342	0.315
Modulus of Elasticity Gpa	12.4	113.8	220

Introduction to Finite Element Method (FEM):

5.4.1 Basic Introduction:

In mathematics, the finite element method (FEM) is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. It uses subdivision of a whole problem domain into simpler parts, called finite elements, and various methods from the calculus of variations to solve the problem by minimizing an associated error function. Analogous to the idea that connecting many tiny straight lines can approximate a larger circle, FEM encompasses methods for connecting many simple element equations over many small sub domains, named finite elements, to approximate a more complex equation over a larger domain.

1) Basic Concept:

The subdivision of a whole domain into simpler parts has several advantages:

- Accurate representation of complex geometry
- Inclusion of dissimilar material properties
- Easy representation of the total solution
- Capture of local effects.

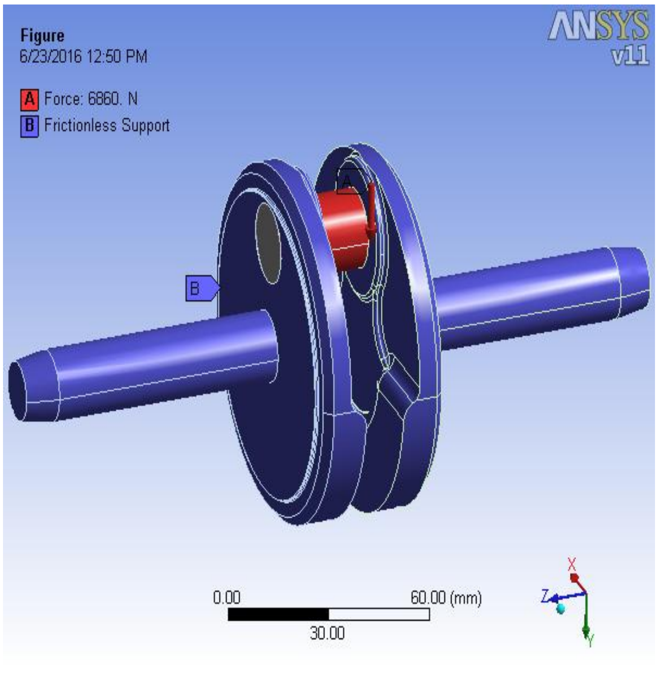
A typical work out of the method involves (1) dividing the domain of the problem into a collection of sub domains, with each sub domain represented by a set of element equations to the original problem, followed by (2) systematically recombining all sets of element equations into a global system of equations for the final calculation. The global system of equations has known solution techniques, and can be calculated from the initial values of the original problem to obtain a numerical answer. In the first step above, the element equations are simple equations that locally approximate the original complex equations to be studied, where the original equations are often partial differential equations (PDE). To explain the approximation in this Comparative analysis of fatigue failure in single cylinder petrol engine process, FEM is commonly introduced as a special case of Galerkin method. The process, in mathematical language, is to construct an integral of the inner product of the residual and the weight functions and set the integral to zero. In simple terms, it is a procedure that minimizes the error of approximation by fitting trial functions into the PDE. The residual is the error caused by the trial functions, and the weight functions are polynomial approximation functions that project the residual. The process eliminates all the spatial derivatives from the PDE, thus approximating the PDE locally with

- A set of algebraic equations for steady state problems.
- A set of ordinary differential equations for transient problems.

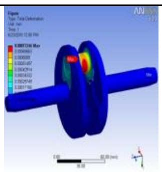
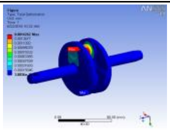
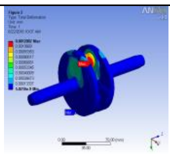
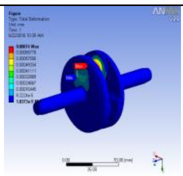
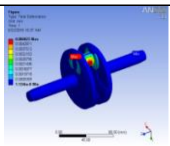
These equation sets are the element equations. They are linear if the underlying PDE is linear, and vice versa. Algebraic equation sets that arise in the steady state problems are solved using numerical linear algebra methods, while ordinary differential equation sets that arise in the transient problems are solved by numerical integration using standard techniques such as Euler's method or the Runge-Kutta method.

In step (2) above, a global system of equations is generated from the element equations through a transformation of coordinates from the sub domains' local nodes to the domain's global nodes. This spatial transformation includes appropriate orientation adjustments as applied in relation to the reference coordinate system. The process is often carried out by FEM software using coordinate data generated from the subdomains. FEM is best understood from its practical application, known as finite element analysis (FEA). FEA as applied in engineering is a computational tool for performing engineering analysis. It includes the use of mesh generation techniques for dividing a complex problem into small elements, as well as the use of software program coded with FEM algorithm. In applying FEA, the complex problem is usually a physical system with the underlying physics such as the Euler-Bernoulli beam equation, the heat equation, or the Navier-Stokes equations expressed in either PDE or integral equations, while the divided small elements of the complex problem represent different areas in the physical system. Comparative analysis of fatigue failure in single cylinder petrol engine FEA is a good choice for analyzing problems over complicated domains (like cars and oil pipelines), when the domain changes (as during a solid state reaction with a moving boundary), when the desired precision varies over the entire domain, or when the solution lacks smoothness.

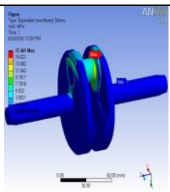
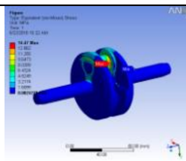
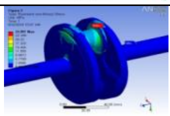
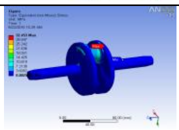
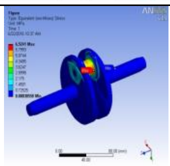
Physics Type	Structural
Analysis Type	Static Structural
Reference Temp	42. °C
Object Name	Analysis Settings
State	Fully Defined
Save ANSYS db	No
Delete Unneeded Files	Yes
Nonlinear Solution	No
Object Name	Force
Magnitude	6860. N (ramped)



Force Analysis of Crankshaft

Alloys	Figure	Total deformation results
Alloy steel 41Cr4		7.7246×10^{-4} mm
Titanium alloy		1.4262×10^{-3} mm
Aluminium alloy		1.2002×10^{-3} mm
Nickel alloy		7.4×10^{-4} mm
Zinc alloy		4.823×10^{-3} mm

Total Deformation of Crankshaft

Alloys	Figure	Equivalent Stress results
Alloy Steel 41Cr4		17.461 MPa
Titanium Alloy		14.47 MPa
Aluminium Alloy		25.997 MPa
Nickel Alloy		32.453 MPa
Zinc Alloy		6.5241 MPa

Equivalent Stress Result of Crankshaft

Analytical Observations of Alloy Steel, Titanium Alloy, Aluminum Alloy in Hz

Freq.	Alloy Steel	Titanium Alloy	Aluminum Alloy
1.	642.1	618.69	585.8
2.	697.5	683.09	676.2
3.	754.7	741.52	696.2
4.	1679.6	1621.6	1733.6
5.	1913.3	1879.4	1909.0
6.	2092.9	2049.5	2469.4
7.	2935.6	2874.8	3005.7
8.	3023.9	2979.3	3189.6

Analytical Observations of Nickel Alloy, Zinc Alloy

Freq.	Nickel Alloy	Zinc Alloy
1.	612.7	169.5
2.	669.2	183.5
3.	725.0	198.4
4.	1603.5	443.3
5.	1837.8	502.9
6.	2008.1	550.6
7.	2816.7	772.2
8.	2907.7	794.3

VI. RESULT ANALYSIS AND COMPARISON

Alloy Steel 41Cr4			
Mode Shapes	**Mathematical Results**	**FEA ANSYS Results**	**% Error**
	Frequency in Hz		
1	652.1	642.1	1.5
2	687.8	697.5	-1.4
3	747.2	754.7	-1.0
4	1689.1	1679.6	0.6
5	2037.6	1913.3	6.1
6	2161.6	2092.9	3.2
7	2791.4	2935.6	-5.2
8	3345.7	3023.9	9.6

TITANIUM ALLOY			
Mode Shapes	**Mathematical Results**	**FEA ANSYS Results**	**% Error**
	Frequency in Hz		
1	603.0	618.69	-2.6
2	665.2	683.09	-2.7
3	707.1	741.52	-4.9
4	1693.0	1621.6	4.2
5	1803.1	1879.4	-4.2
6	2137.4	2049.5	4.1
7	2880.7	2874.8	0.2
8	2901.9	2979.3	-2.7

ALUMINUM ALLOY			
Mode Shapes	Mathematical Results	FEA ANSYS Results	% Error
	Frequency in Hz		
1	599.4	585.8	2.3
2	650.2	676.2	-4.0
3	729.9	696.2	4.6
4	1648.8	1733.6	-5.1
5	2110.5	1909.0	9.5
6	2362.6	2469.4	-4.5
7	3360.7	3005.7	10.6
8	3548.0	3189.6	10.1

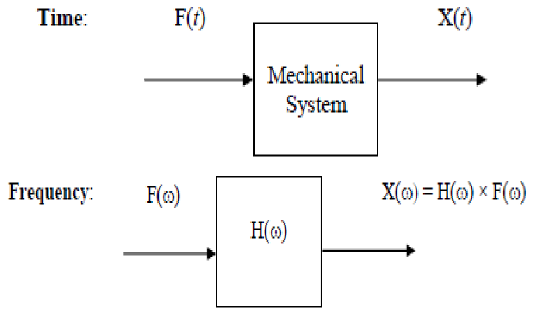
NICKEL ALLOY			
Mode Shapes	Mathematical Results	FEA ANSYS Results	% Error
	Frequency in Hz		
1	611.7	612.7	-0.2
2	663.1	669.2	-0.9
3	717.2	725.0	-1.1
4	1642.2	1603.5	2.4
5	1842.0	1837.8	0.2
6	2023.1	2008.1	0.7
7	2829.5	2816.7	0.5
8	3000.6	2907.7	3.1

ZINC ALLOY			
Mode Shapes	Mathematical Results	FEA ANSYS Results	% Error
	Frequency in Hz		
1	160.8	169.5	-5.4
2	179.3	183.5	-2.3
3	200.7	198.4	1.1
4	449.4	443.3	1.4
5	481.4	502.9	-4.5
6	603.9	550.6	8.8
7	740.2	772.2	-4.3
8	890.4	794.3	10.8

VII. EXPERIMENTATION

The experimental modal analysis (EMA) means the extraction of modal parameters (frequencies, damping ratios, and mode shapes) from measurements of dynamic responses. Basically, it is carried out according to both input and output measurement data through the frequency response functions (FRFs) in the frequency domain, or impulse response functions (IRFs) in the time domain. For mechanical engineering structures, the dynamic responses (output) are the direct records of the sensors.

EMA has grown steadily in popularity since the advent of the digital FFT (Fast Fourier Transformation) spectrum analyzer in the early 1970's (Schwarz & Richardson). In this paper, we will make FRF measurements with a FFT analyzer, modal excitation techniques, and modal parameter estimation from a set of FRFs (curve fitting). Experimental modal parameters (frequency, damping, and mode shape) are also obtained from a set of FRF measurements. The FRF describes the input-output relationship between two points on a structure as a function of frequency. Since both force and motion are vector quantities, they have directions associated with them. Therefore, an FRF is actually defined between a single input DOF (point &direction), and a single output DOF. FRF is defined as the ratio of the Fourier transform of an output response (X (ω)) divided by the Fourier transform of the input force (F (ω)) that caused the output (See Fig. 6.1). An FRF is a complexes valued function of frequency. Actually FRF measurements are computed in a FFT analyzer.



Time and Frequency Domain

Exciting Modes with Impact Testing

Impact testing is a fast, convenient, and low cost way of finding the modes of machines and structures. All the tests were performed at the (name of your company), in the Mechanical engineering LAB, at LGMT - CRITT. The following equipment is required to perform an impact test:

1. An ***impact hammer*** with a load cell attached to its head to measure the input force

2. An ***accelerometer*** to measure the response acceleration at a fixed point & direction

3. A 2 channel ***FFT analyzer*** to compute FRFs.

4. ***Post-processing modal software*** for identifying modal parameters and displaying the mode shapes in animation.

	
Vibrometer or accelerometer	Impact hammer / feedback hammer
	
Signal receives from accelerometer and Impact hammer to FFT	Curve fit or graph generated on laptop screen

VIII. RESULTS AND COMPARISON

MODAL ANALYSIS

Alloy Steel 41Cr4			
Mode Shapes	Mathematical Results in Hz	FEA ANSYS Results in Hz	% Error
1	652.1	642.1	1.5
2	687.8	697.5	-1.4
3	747.2	754.7	-1.0
4	1689.1	1679.6	0.6
5	2037.6	1913.3	6.1
6	2161.6	2092.9	3.2
7	2791.4	2935.6	-5.2
8	3345.7	3023.9	9.6

TITANIUM ALLOY			
Mode Shapes	Mathematical Results in Hz	FEA ANSYS Results in Hz	% Error
1	603.0	618.69	-2.6
2	665.2	683.09	-2.7
3	707.1	741.52	-4.9
4	1693.0	1621.6	4.2
5	1803.1	1879.4	-4.2
6	2137.4	2049.5	4.1
7	2880.7	2874.8	0.2
8	2901.9	2979.3	-2.7

ALUMINUM ALLOY			
Mode Shapes	Mathematical Results in Hz	FEA ANSYS Results in Hz	% Error
1	599.4	585.8	2.3
2	650.2	676.2	-4.0
3	729.9	696.2	4.6
4	1648.8	1733.6	-5.1
5	2110.5	1909.0	9.5
6	2362.6	2469.4	-4.5
7	3360.7	3005.7	10.6
8	3548.0	3189.6	10.1

NICKEL ALLOY			
Mode Shapes	Mathematical Results in Hz	FEA ANSYS Results in Hz	**% Error**
1	611.7	612.7	-0.2
2	663.1	669.2	-0.9
3	717.2	725.0	-1.1
4	1642.2	1603.5	2.4
5	1842.0	1837.8	0.2
6	2023.1	2008.1	0.7
7	2829.5	2816.7	0.5
8	3000.6	2907.7	3.1

ZINC ALLOY			
Mode Shapes	Mathematical Results in Hz	FEA ANSYS Results in Hz	% Error
1	160.8	169.5	-5.4
2	179.3	183.5	-2.3
3	200.7	198.4	1.1
4	449.4	443.3	1.4
5	481.4	502.9	-4.5
6	603.9	550.6	8.8
7	740.2	772.2	-4.3
8	890.4	794.3	10.8

IX. RESULTS AND DISCUSSION

Mathematical Comparison of Existing and Modified Crankshaft

Frequency Range for Mode	Existing	Modified After Coating
	Alloy Steel 41cr4 Crank Shaft	Zinc Alloy Crank Shaft
	Mathematical in Hz	Mathematical in Hz
1.	640.0	160.8
2.	687.9	179.3
3.	747.2	200.7
4.	1689.1	449.4
5.	2037.7	481.4
6.	2161.7	603.9
7.	2791.7	740.2
8.	3345.8	890.4

Frequency Range for Mode	Existing	Modified After Coating
	Alloy Steel 41cr4 Crank Shaft	Zinc Alloy Crank Shaft
	FEA Value in Hz	FEA Value in Hz
1.	642.1	169.5
2.	697.5	183.5
3.	754.7	198.4
4.	1679.6	443.3
5.	1913.3	502.9
6.	2092.9	550.6
7.	2935.6	772.2
8.	3023.9	794.3

Frequency Range for Mode	Existing	Modified After Coating
	Alloy Steel 41cr4 Crank Shaft	Zinc Alloy Crank Shaft
	FEA Value in Hz	FEA Value in Hz
1.	642.1	169.5
2.	697.5	183.5
3.	754.7	198.4
4.	1679.6	443.3
5.	1913.3	502.9
6.	2092.9	550.6
7.	2935.6	772.2
8.	3023.9	794.3

X. CONCLUSIONS & RECOMMENDATIONS

In this project, the crankshaft model was created by CATIA V5 for modeling the crank shaft. Then, the created model was import into ANSYS software for static structural analysis.

FEA Results for equivalent (Von-Misses) stress, shows that all selected material are below the limiting stress value of crankshaft and hence can be used for testing. Also the results of modal analysis FEA results match with the experimental and theoretical calculation for validation of model. The value of Von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe.

The analysis of the crank shaft was done using five different materials. These materials are Alloy Steel 41Cr4, Titanium Alloy, Al Alloy, Zinc alloy. The comparison of analysis results of all five materials will show the effect of stresses on different materials and this will help to select suitable material for coating.

Zinc Alloy can be used or coating can be done as it shows better results and also economically cheaper. Force analysis results also shows that it is having more stress bearing capacity. Induced stress is also less so it can be preferred.

ACKNOWLEDGMENT

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