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Optimization and Analysis of a Crankshaft in External Combustion Diesel Engine

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ABSTRACT

This project presents the design and optimization of crank shaft. The objectives of this project are to develop structural modeling, finite element analyzes and the optimization of the crank shaft for robust design. The structure of crank shaft was modeled utilized Pro-E software. Finite element modeling and analysis were performed using ANSYS software. Linear static analysis was carried out to obtain the stress/strain state results. The mesh convergence analysis was performed to select the best mesh for the analysis. The topology optimization technique is used to achieve the objectives of optimization which is to reduce the weight of the crank shaft. From the FEA analysis results, predicted higher maximum stress and maximum principal stress captured the maximum stress. The crank shaft is suggested to be redesign based on the topology optimization results. The optimized Crank shaft is lighter and predicted low maximum stress compare to initial design. For future research, the optimization should cover on material optimization to increase the strength of the Crank shaft. We also made thermal, vibration and fatigue analysis on this design.

KEYWORDS: *structural modeling, crank shaft, mesh convergence analysis, weight of the crank shaft, redesign, vibration and fatigue analysis on this design.*

I. INTRODUCTION

A crankshaft—related to crank—is a mechanical part able to perform a conversion between reciprocating motion and rotational motion. In a reciprocating engine, it translates reciprocating motion of the piston into rotational motion; whereas in a reciprocating compressor, it converts the rotational motion into reciprocating motion. In order to do the conversion between two motions, the crankshaft ank throws" or "crankpins", additional bearing surfacehas "crs whose axis is offset from that of the crank, to which the "big ends" of the crank shafts from each cylinder attach.It is typically connected to a flywheel to reduce the pulsation characteristic of the four-stroke cycle, and sometimes a torsional or vibrational damper at the opposite end, to reduce the torsional vibrations often caused along the length of the crankshaft by the cylinders farthest from the output end acting on the torsional elasticity of the metal. Most modern high speed production engines are classified as "over square" or short-stroke, wherein the stroke is less than the diameter of the cylinder bore. As such, finding the proper balance between shaft-stroking speed and length leads to better results.

II. ENGINE CONFIGURATION

The configuration, meaning the number of pistons and their placement in relation to each other leads to straight, V or flat engines. The same basic engine block can sometimes be used with different crankshafts, however, to alter the firing order. For instance, the 90° V6 engine configuration, in older days sometimes derived by using six cylinders of a V8 engine with a 3 throw crankshaft, produces an engine with an inherent pulsation in the power flow due to the "gap" between the firing pulses alternates between short and long pauses because the 90 degree engine block does not correspond to the 120 degree spacing of the crankshaft. The same engine, however, can be made to provide evenly spaced power pulses by using a crankshaft with an individual crank throw for each cylinder, spaced so that the pistons are actually phased 120° apart, as in the GM 3800 engine.

While most production V8 engines use four crank throws spaced 90° apart, high-performance V8 engines often use a "flat" crankshaft with throws spaced 180° apart, essentially resulting in two straight four engines running on a common crankcase. The difference can be heard as the flat-plane crankshafts result in the engine having a smoother, higher-pitched sound than cross-plane (for example, IRL IndyCar Series compared to NASCAR Sprint Cup Series, or a Ferrari 355 compared to a Chevrolet Corvette). This type of crankshaft was also used on early types of V8 engines. See the main article on crossplane crankshafts.

III. ENGINE BALANCE

For some engines it is necessary to provide counterweights for the reciprocating mass of each piston and crank shaft to improve engine balance. These are typically cast as part of the crankshaft but, occasionally, are bolt-on pieces. While counter weights add a considerable amount of weight to the crankshaft, it provides a smoother running engine and allows higher RPM levels to be reached.

IV. FLYING ARMS

In some engine configurations, the crankshaft contains direct links between adjacent crankpins, without the usual intermediate main bearing. These links are called *flying arms*.[20] This arrangement is sometimes used in V6 and V8 engines as it enables the engine to be designed with different V angles than what would otherwise be required to create an even firing interval, while still using fewer main bearings than would normally be required with a single piston per crankthrow. This arrangement reduces weight and engine length at the expense of less crankshaft rigidity.

V. BASICS OF COMPOSITE MATERIALS

From the literature review for automotive application especially crankshafts are made by Composite materials.Metal Composite materials have found application in many areas of daily life for quite some time. Often it is not realized that the application makes use of Composite materials. These materials are produced from the conventional production and processing of metals. Here, the structure, which results from welding two types of steel by repeated forging, can be mentioned. Materials like cast iron with graphite or steel with a high carbide content, as well as tungsten carbides, consisting of carbides and metallic binders, also belong to this group of Composite materials. For many researchers the term metal matrix Composite s is often equated with the term light metal matrix Composite s (MMCs).

Substantial progress in the development of light metal matrix Composite s has been achieved in recent decades, so that they could be introduced into the most important applications. In traffic engineering, especially in the automotive industry, MMC shave been used commercially in fiber reinforced pistons and aluminum crank cases with strengthened cylinder surfaces as well as particle-strengthened brake disks .These innovative materials open up unlimited possibilities for modern material science and development. The characteristics of MMCs can be designed into the material, custom-made, dependent on the application.

Basis of Composite material selection

1. Composite materials exhibit superior mechanical properties such as high strength, toughness, young's modulus, fairly good fatigue and impact properties.
2. As fiber Composite are light weight materials, the specific strength and specific modulus are much higher than the conventional materials. Composite materials are
    1. E-glass/epoxy
    2. carbon epoxy
    3. E-glass/polyurethane

VI. MATERIAL PROPERTY

Aluminum and magnesium casting reinforced with fiber are being developed for high performance automotive application. while they are lighter and may offer better compressive strength, stiffness, and fatigue resistance than conventional engine materials .the material properties are shown in below table

Table 1Material Properties Of MMC

PROPERTIES	VALUE
Tensile Modulus along X direction, (Ex), MPa	20700
Tensile Modulus alongY,Zdirection, MPa	14400
Shear Modulus along XY direction, (Gxy), MPa	48000
Shear Modulus along YZ direction, (Gxy), MPa	5520
Shear Modulus along ZX direction, (Gxy), MPa	2760
Poisson ratio along XY direction, (NUxy)	0.244
Poisson ratio along YZ direction, (NUyz)	0.17
Poisson ratio along ZX direction, (NUzx)	0.3
Mass Density of the Material,(ρ),Kg/mm3	5e-6

LIFE

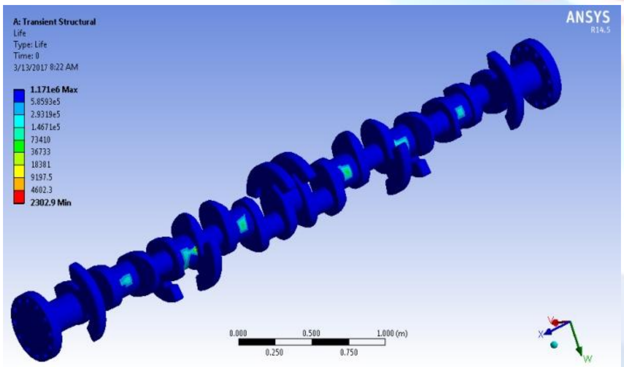

Fig 1 Model

MODEL

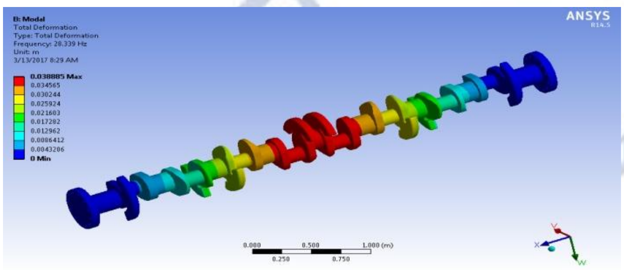

Fig.2 Analysis of Model

CALCULATIONS

Specifications:

B.H.P = 7.6
Bore = 49 mm
Stroke = 56 mm
Crank Radius = 56/2
= 28 mm
Piston Diameter = 48.5mm
Piston weight = 65g (0.065 kg)
RPM = 7500
Torque = 0.8kg m

Load calculation:

$$P_m = \frac{B.H.Px60}{LxAxN'}$$

$$N' = \frac{N}{2} = \frac{7500}{2} = 3750 \text{ rpm}$$

$$= \frac{7.6x60}{56x\frac{\Pi}{4}x48.5^2x3750}$$

$P_m = 1.175 \text{ x N / mm}^2$

$F_L = P_m \text{ x Area}$

$$F_L = 1.175 \text{ x} \frac{\Pi}{4} \text{x} (48.5)^2$$

$F_L = 2170 \text{ N}$

$$F_I = M_R x W^2 x r \left(Cos\theta + \frac{Cos2\theta}{n} \right)$$

$$n = \frac{l}{r} = \frac{100}{28} = 3.571$$

$\theta = 0^0$

$$F_I = 0.065 \text{ x} \frac{2\pi x7500}{60} \text{x } 0.028 \quad \frac{1+0.028}{0.100}$$

VII. RESULT AND DISCUSSION

Stress Analysis Comparision

The FEM value of conventional and MMC crank shaft is obtained and is shown in table .from the result it is clear that the conventional crank shaft stress are greater than the MMC crank shaft .The comparisons of stress in Composite crank shaft.

Table 1.1 comparisons of stress

Stroke	Conventional crank shaft	Composite crank shaft
Tensile stress(MPa)	793.109	436.78
Compressive stress(MPa)	765.408	426.735

VIII. FATIGUE ANALYSIS OF CONVENATIONAL CRANK SHAFT

The Table clearly indicates the crack nucleation starts from the cycle 10000 cycles at the stress value of 250MPa, this is because the crack nucleation starts only after certain value or stress which high enough to generate a crack and to propagate it. Then at 20000 cycle stress value decreases to 160MPa this is because due to the discontinuities, where as at 30000 cycle the stress value slightly rises as a result of fatigue propagation. After which the sudden down fall to 125MPa at 40000 cycles indicating inclusions and variations in cross-sectional area. At 50000 cycles the stress value reaches the highest stress value of 1175 MPa which is well above the working stress limit of 793.109. At this value the Crank shaft gets failed.

After then at 60000 cycles the stress values gradually decreases and reaches a low stress value remains even during the next cycle of loading till 100000 cycles of loading. Analysis up to 100000 cycles are tabulated below and the maximum stress observed as 1175MPa which exceeds the allowable ultimate strength 793.109 MP the failure occurs at 50000 cycles

Table 1.2 Cyclic loading Vs Stress value

Serial Number	CYCLIC LOAD in X-axis	STRESS VALUE in Y-axis
1.	0	0
2.	5000	0
3.	10000	325
4.	15000	250
5.	20000	200
6.	25000	315
7.	30000	440
8.	35000	250
9.	40000	100
10.	45000	625
11.	50000	**1175***
12.	55000	625
13.	60000	75
14.	65000	50
15.	70000	25
16.	75000	25
17.	80000	10
18.	85000	10
19.	90000	10
20.	95000	10
21.	100000	10

* Peak value exceeds the ultimate tensile strength, which implies 793.109MPa. At this Peak value 1175MPa crank shaft fails at 50000 cycles

IX. FATIGUE ANALYSIS OF COMPOSITE CRANK SHAFT

Taking number of cycles to failure N along x-axis and stress (MPa) along y-axis. Considering number of cycles up to 100000 cycles to failure and stress ranging 0 to 800 MPa. The table 14.3 clearly indicates the crack nucleation starts from the cycle 10000 cycles at the stress value of 250MPa, this is because the crack nucleation starts only after certain value or stress which high enough to generate a crack and to propagate it.

Then at 20000 cycle stress value decreases to 320MPa this is because due to the discontinuities, where as at 30000 cycle the stress value slightly rises as a result of fatigue propagation. After which the sudden down fall to 160MPa at 50000 cycles indicating inclusions and variations in cross-sectional area.

At 65000 cycles the stress value reaches the highest stress value of 650MPa which is well above the working stress limit of 436.78. At this value the Crank shaft gets failed.After then at 80000 cycles the stress values gradually decreases and reaches a low stress value remains even during the next cycle of loading till 100000 cycles of loading.Analysis up to 100000 cycles are tabulated below and the maximum stress observed as 650MPa which exceeds the allowable ultimate strength 436.78 MP the failure occurs at 65000 cycles

Table 1.3 : Cyclic loading Vs Stress value

Serial Number	CYCLIC LOAD in X-axis	STRESS VALUE in Y-axis
1.	0	0
2.	5000	0
3.	10000	250
4.	15000	200
5.	20000	125
6.	25000	300
7.	30000	320
8.	35000	290
9.	40000	240
10.	45000	200
11.	50000	160
12.	55000	400
13.	60000	480
14.	65000	**650***
15.	70000	450
16.	75000	300
17.	80000	80
18.	85000	10
19.	90000	10
20.	95000	10
21.	100000	10

* Peak value exceeds the ultimate tensile strength, which implies 436.735MPa. At this Peak value 650 MPa crank shaft fails at 65000 cycles.

X. FATIGUE ANALYSIS COMPARISION

After the fatigue life analysis made between the conventional Crank shaft with Composite crank shaft, to infer the solution in various viewpoints.

Initially the conventional Crank shaft subjected to analyzing without the material replacement gives us the result as it has the fatigue life up to 50000 cycles reaching a maximum stress of 1175MPa which is higher than the working stress, 795.109MPa.

Subjecting the Composite Crank shaft to the fatigue analysis the result obtained is satisfactory. During 65000 cycle Crank shaft reaches maximum stress of 650MPa where it fails and the stress value decreases. This clearly confers that the Composite crank shaft as its fatigue life up to 65000 cycles, during the next cycle of loading the stress value reaches 450MPa and for further cycles of loading it remains in the low range of stress value. Apart form stress and fatigue analysis, there is weight reduction during the material replacement.

XI. WEIGHT REDUCTION

The strength to weight ratio of Composite crank shaft is greater than that of the conventional crank shaft and more over the density of Composite crank shaft is less than that of the Composite crank shaft. The mass of Composite crank shaft is found to be 72.17 grams than the conventional rod of mass 115.48 grams. So by replacing the material of conventional crank shaft by Composite 37.5% of weight is reduced.

Percentage Reduction in Weight by Material Replacement

$$= \frac{(115.48 - 72.17)}{115.48} = 37.5\%$$

XII. CONCLUSION

The conventional crank shaft used in the engines was replaced with a Composite crank shaft . The conventional crank shaft and the Composite crank shaft were analyzed by finite element methods. From the results, it is clear that the stress induced in the Composite crank shaft is found to be lower than that of the conventional crank shaft.

Composite crank shaft material is replaced for good fatigue strength, minimizing weight and without violating the limiting constraint formed by induced stress. A reduction of 37.5% weight is achieved when a conventional crank shaft is replaced with Composite crank shaft under identical conditions of design parameters.

REFERENCES

[1] Roger rsbb,1996, “fatigue failure of a crank shaft “,engineering failure analysis, vol.3, pp.no. 13-28.
[2] Fantino,2002,”crank shaft made Composite crank shaft”, patentno.us5154096.
[3] Jean-perre maumas,1992,”crank shaft made of Composite materials”,patent no.us154,098.
[4] Person et al,2005,”use of a transmission component ofcomposites Composite (MMC) material”,.
[5] Janasrasback,” Composite in engines”,MX Composite s AB,www.mxComposites.com
[6] Folger,1987 “design, fabrication and performance of fiber FP/composites Composite crank shafts”,SAE.Inc.
[7] Berthed ,1998,“optimized crank shaft to enable higher engine performance and cost reduction”,SAE.Inc
[8] Karl kainer,2006,”basics of composites Composite s”,cutom-made materials for automotive and aerospace engineering, ISBN 3-527-31360
[9] ErolSancaktar, Mathieu Gratton,1999, ‘Design, Analysis and Optimization of Composite Leaf Spring for Light Vehicle Applications’. Composite Structures. 44. 195-204.
[10] Autar k.kaw,1997,”mechanics of Composite materials”,CRC Press.
[11] j.n reddy,2003,”an introduction to the finite element methed”, TATA McGRAW HILL.
[12] L.J. segerlind,1984,”applied finite element analysis”,,john wiles.
[13] s.s roa,1989,”finite element methed in engineering”, pergamon press.
[14] chandrapatta and belagundu,1997,”finite element in engineering”,prevate lid.
[15] cook,1999,”davis etal,1999,”concept and application of finite element analysis”,wiley,john and sons.