Scientific Journal of Impact Factor (SJIF): 5.71
e-ISSN (O): 2348-4470
p-ISSN (P): 2348-6406

International Journal of Advance Engineering and Research Development

Volume 5, Issue 04, April -2018

DESIGN OF A FULLY ASSEMBLED CRANKSHAFT FOR A 1.2 LITER HORIZONTALLY OPPOSED 4 CYLINDER S.I. ENGINECHUKWUDI NODILIM EZEBUILO[1],SOHAIL SIDDIQI[2]

[1]Mechanical Engineering, Parul Institute of Engineering &Technology, Parul University
[2]Mechanical Engineering Parul Institute of Engineering &Technology, Parul University

Abstract-*This work aims to develop an analytical method of designing a fully assembled crankshaft for automotive application. The design will incorporate innovative octahedral fixture of the main journal shafts to the webs. This distinct feature of the design will basically improve the fatigue life of the design since most fatigue failures of assembled crankshafts occur at this region due to alternating slip-fret.Theories and formulas deduced in the analysis will be illustrated in the design of a crankshaft for a 1.2 litre horizontally opposed 4 cylinder spark ignition engine as charging power plant of a hybrid vehicle. The analytical method will evaluated the nominal sizes and dimensions based on stress analysis of the crankshaft configuration. Also analytically, the fatigue analysis will be performed to ensure that the part survives cyclic fluctuating stresses during service for infinite life of operation. Since the part will be subjected to high cycle of operation, the stress-life method of fatigue analysis will be adopted.For validation of the deduced analytical method, a finite element analysis of the solid modelled part will be computer-simulated with ANSYS workbench software. Then the results of the analytical method will be compared with that of FEA for conformity and agreement.The design and development of this innovation segmented crankshaft for the charging power unit of a hybrid vehicle will play a vital role in the green transition.*

Keywords*-crank shaft, connecting rod, stress,engine, combustion*

I. INTRODUCTION

The crankshaft is made up of three main parts. The first part is the main journal, the part of the crankshaft that is fixed on the engine block. The centrally located coaxial cylindrical main journals rotate in a set of supporting main bearings where it performs fixed center rotary motion and serves as output point of the engine for transmission input. The second part is the crankpin with its center offset from the center of the main journal. The action of the reciprocating pistons is transmitted to the crankpin through the connecting rod link. The crankpin performs an orbit circumferential motion about the center of the main journal at a radius known as the crank throw. A stroke of an IC engine is twice the crank throws. The big ends of the connecting rods are fixed with bearings which ride on the crankpins.The last part of the crankshaft is the web. The web is perpendicular to the axis of the crankshaft thereby forming the torque arm of the engine. The web rigidly connects the off axis main journal and crankpin to form the crankiness of the shaft.

The layout of the crankshaft of a multi cylinder IC engine is determined by the type of adopted cylinder arrangement. This paper aims to investigate a design of multi-piece crankshaft for a horizontally opposed four cylinder, four stroke SI engine of a modest total swept volume of 1.2 litres.

There are two types of possible horizontally opposed arrangement based on the configuration of the crankshaft. One type is a horizontally opposed boxer arrangement where the number of cranks on the shaft is the same as the number of cylinders. It implies that each conrod is connected to each designated crank pin. This paper dealt on horizontally opposed arrangement is where there is half the number of cranks on the shaft as the number of cylinders. It implies that each crankpin has two adjacent con rods attached on it. This type is also referred to as 180 degree V engines or flat V engines. Whereas the boxer crankshaft has better engine balance characteristics than flat V, this advantage becomes significant when the swept volume becomes comparatively large. The multi-piece crankshaft before assembly is comprised of two crankpins, four web pieces and three main journal pieces. The central main journal pieces will be designed to accommodate thrust bearings.

The aim of this work is to investigate and present an analytical method of design of a fully assembled multi-piece assembled crankshaft with octagonal mating surfaces. The design methodology is demonstrated in stress and fatigue analysis to deduce geometrical dimensions required in manufacture of the crankshaft for a 1.2 liter horizontally opposed four cylinder S.I engine. The multi-piece assembly technique offers effective mass production model that reduces man-hours in manufacture of engine crankshafts. The application of the design is expected to give improved engine performance and durability in comparison to existing cast or forged ones. The design will also reduce the cost of crankshaft production thereby lowering the cost of the I.C. engine.

NOMENCLETURES

Upper case letters

A - Area
AF - Stoichiometric air-fuel ratio
BP - Brake Power
C_v- Constant volume heat capacity
D - Diameter, Bore
E - Young's modulus
F - Force
I - Moment of inertia, Fit interference
IMEP- Indicated mean effective pressure
IP - Indicated power
K - Radius of gyration, Fatigue modifier
L - Stroke length
M - Moment
N - Rotational speed, Factor of safety
OQT - Oil quenched and tampered
P - Pressure
Q_{HV} - Heat value of petrol fuel
R - Ideal gas constant, Shear force
S - Fatigue strength
S_c - Bearing cap width
T - Temperature, Torque
V - Volume
W - Work
X_r - Residual burnt fraction
Z - Section modulus

Lower case letters

a - Crank radius
b - Width
d - Diameter
e - Extension length
h - Height
b - bush
bc - breath of connecting rod
c - combustion, clearance, compression
cc - center to center
ci - crankpin inner diameter
co - crankpin outer diameter
cp - crankpin
cr - connecting rod
cw - crankpin web
e - expansion, endurance
f - fuel, final, fatigue
i - inertia, initial
j - journal
l - Length
k_c- Specific heat ratio during compression
k_e-Specific heat ratio during expansion
k_f - Fatigue stress concentrating factor
k_t - Static stress concentrating factor
m - Mass.
n - connecting rod length to crank radius ratio
r_c - Compression ratio
r_e - Expansion ratio
t - Thickness

Greek letters

Ɵ - Fatigue ratio
η - Efficiency
θ - Crank angle
ρ - Density
σ - Normal stress
τ - Shear stress
φ - Angle at superposition
ω - Angular speed

Greek letters (contd).

α - Thermal expansivity
μ - Coefficient of friction

Subscripts

B - bending
G - gas
HP - horizontal primary
HS - horizontal secondary
T - torsion
VP - vertical primary
VS - vertical secondary
a - air, alternating
all - allowable
ne - allowable enduarance
m - mechanical, mixture, mean
mw - main journal –web
n - net
o - outer
p - piston group
r - residual, radial
s - swept
t - static
ut - ultimate
v - von mises
w - whipping, web
XX - about x–axis
YY - about y-axis
θ - tangential

II. ESTIMATION OF IN CYLINDER PRESSURE WITH RESPECT TO CRANK ANGLE

To initiate the stress analysis of the crankshaft, the distribution of the in cylinder gas pressure with respect to crank angle rotation must be evaluated if not given. The only data provided for the engine are volumetric swept volume of 1.2 liter, number of cylinder of four, mode of cylinder arrangement of horizontally opposed and the operating cycle of operation of four stroke spark ignition.

Hence the In cylinder pressure distribution has to be estimated as a prerequisite for the stress analysis of the crankshaft. The estimation of the gas pressure is based on calculations by a normalized ideal fuel-air cycle of the engine running on Atkinson cycle.

Real practical range of values for compression-expansion ratios, Bore to stroke ratio, connecting rod length to crank radius ratio, combustion efficiency, mechanical efficiency, specific heat ratios, lower heat value of fuel, stoichiometric air-fuel ratio, residual burnt fraction and operating constant speed of the engine are carefully chosen.

The ideal cycle is normalized by a factor to imitate real engine cycle to obtain the engine power rating and maximum in cylinder gas pressure. The normalized P-V diagram is curve fitted to a pressure-crank angle diagram to obtain the P-θ expression.

Engine parameters are assumed with practical values as follows –

I. Number of cylinders = 4
II. Total swept volume = 1.2 liter
III. Bore-Stoke ratio = 1
IV. Con-rod length-crank radius ratio, n = 3.5
V. Combustion efficiency, $\eta_c = 0.95$
VI. Mechanical efficiency, $\eta_m = 0.9$
VII. Compression ratio, $r_c = 9$
VIII. Expansion ratio, $r_e = 12$
IX. Compression specific heat ratio, $k_c = 1.35$
X. Expansion specific heat ratio, $k_e = 1.3$
XI. Ideal gas constant, R = 287 J/Kg.K
XII. Heat capacity at constant volume, C_v = 821 J/kg.K
XIII. Lower heat value of petrol, Q_{HV} = 44,000 KJ/kg
XIV. Stoichiometric air-fuel ratio , AF = 14.7
XV. Residual burnt fraction, $x_r = 0.03$

The analysis of the engine based on normalized fuel-air cycle gives estimate values as
Brake power@ 3600 rpm = 42.64 KW
Engine Torque, T = 133 Nm
Peak in-cylinder pressure, P_G = 8.575 MPa. In-cylinder pressure distribution is deduced as

$$P = 11.2e^{-1.5\theta} \text{ MPa} \quad \ldots\ldots (1)$$

III. ESTIMATION OF GEOMETRIC DIMENSIONS OF BIG END OF THE CONNECTING ROD

In structural design of a crankshaft, the thickness of the big end of the connecting rod determines the length of the crankpin. Since the geometric dimension of the connecting rod is not given, the method of static superposition is used to deduce the approximate dimensions of the connecting rod. The design will aim at minimum allowable big end thickness to improve the rigidity of the crankshaft.

The following practical assumptions are made to enable estimated design of the crankshaft.

I. Material – SAE 1045 forged and heat treated.
II. Material yield strength – 1500 MPa
III. Material density P = 7850 kg/m^3
IV. Length of connecting rod = 3.5a = 0.127 m
V. Allowable crankpin pressure = 35 MPa
VI. Bush thickness t_b = 3mm
VII. Mass of piston group = 30% of swept mass of Aluminum.
VIII. Conrod I-section profile = 5t x 4t
IX. Design safety factor, N = 5

From the design of the conrod big end, the two dimensions required in the design of the crankshaft are

I. Conrod breath thickness, l_{bc} = 24mm
II. Crankpin diameter, d_{co} = 42mm

IV. SELECTION OF CRANKSHAFT MATERIAL

Selection of a suitable material for a machine part is a very critical design activity performed to balance economic, technical and quality constraints in the production of the machine part. In material selection for engine crankshaft, technical qualities like yield strength, ultimate tensile strength, machinability, hardenability, nitridability, Surface and core hardness, fatigue strength, ductility, impact resistance, corrosion resistance and temper-embrittlement resistance are properties desired in combination in varying proportions. Several design factors determine the order of importance which must be carefully chosen among several material properties. Generally appropriate selection of an alloy steel of the most desirable combination is aimed at.

In this work, the machinability and strength are paramount properties desired in the selected alloy steel. To meet the economic constraint criteria, the selected alloy steel of low alloying content with high availability from suppliers is needed. The material should be supplied as normalized hot rolled forms, billets for the webs and rods for the shafts. The sizing of the supplied material should be in closeness to the design dimensions for minimum machining. The material should be of low normalized strength for high machining tool life but high final strength after treatments.

Based on above considerations, the selected material medium carbon steel based on SAE grading is SAE 4130.
SAE 4130 is a low alloy medium carbon steel which contains chromium and molybdenum as strengthening agents. As well as high machinability, it is more responsive to treatments and can easily be welded. From the data of heat treated SAE 4130, the yield tensile strength is 1380mpa before surface hardening.This value will serve as the basis for stress analysis.Tabulated below are properties of the material.

Table 1. Chemical Composition of SAE 4130

Elements	Fe (Iron)	C (Carbon)	Cr (Chromium)	Mn (Manganese)
Percentage	97.3 – 98.22	0.28 – 0.33	0.8 – 1.1	0.4-0.6

Mo (Molybdenum)	P (Phosphorus)	S (Sulphur)	Si (Silicon)
0.15 – 0.25	Less than 0.035	Less than 0.04	0.15 – 0.35

Table 2. The physical and mechanical properties of SAE 4130 normalized

Properties	Density	Melting point	Thermal Conductivity	Ultimate tensile strength
Value	7850 kg/m^2	432 ^{0}C	42.7 W/mK	560 MPa
Linear expansivity	Yield tensile strength	Modulus of Elasticity	Shear Modulus	Poisson ratio
1.12 x 10^{-5} ^{0}C^{-1}	460 MPa	200 GPa	80 GPa	0.3

Elongation	Brinell Hardness	Rockwell Hardness B	Machinability
21.5%	217	95	70%

Table 3. Mean Mechanical Properties of heat treated SAE 4130

Properties	Treatment - OQT	Ultimate tensile strength	Yield tensile strength
Values	Oil quenched at 870^{0}C Tempered at 315^{0}C	1500 MPa	1380 MPa

Elongation		Brinell Hardness		Cyclic yield strength, S'_y	
11%		435		Fatigue Strain (1070 MPa) Hardening exponent, m = 0.13	
Properties	True Strain at fracture	Fatigue strength coefficient	Fatigue strength exponent	Fatigue ductility coefficient	Fatigue ductility exponent
Values	0.79	1695 MPa	-0.081	0.89	-0.69

V. INITIAL STRESS ANALYSIS

Crankshaft members are loaded in combination of torsion, bending, direct tension-compression or direct shear stresses depending on the part and location. For multi axial loading, the Distortion Energy failure criterion also known as Von-Mises theory which is suitable for ductile steels will be adopted in this analysis. The type of load acting on each member or sub member is specified before applying appropriate failure criterion. The analysis begins with gas pressure and the reciprocating forces as the source load which is transmitted to the crankpin through the connecting rod, from the connecting rod to the webs and finally to the main journals.

The initial parameters requires for the analysis are:

1. Material is SAE 4130 with tensile yield strength of 1380 MPa.
2. Allowable crankpin bearing pressure is 35MPa.
3. Factor of safety is 5.
4. From equation .1, the distribution of in-cylinder pressure with respect to crank angle is $P = 11.2e^{-1.5\theta}$
5. Length of crankpin as twice the connecting rod breath and axial side play allowance of 2mm. Therefore $l_{cp} = 2\ (24) + 2 = 50$ mm.
6. Allowable tensile strength, $\sigma_{all} = 1380/5 = 276$ MPa.
7. Allowable shear strength, $\tau_{all} = 0.5\sigma_{all} = 138$ MPa
8. Crank angle at maximum torsion, $\theta_T = \mathrm{Tan}^{-1}(n) = 74^0 = 1.25$rad, where crank angle at maximum bending is taken as $\theta_B = 10^0 = 0.175$rad.
9. The angles between the cylinder axis and the line of connecting rod axis at both super positions as $\emptyset_B = 2.84^0$ and $\emptyset_T = 16^0$

During combustion, the crankshaft deflects in bending on gas pressure force with maximum deflection occurring at the center of the crankpin. It is assumed that the crankshaft is rigid enough for negligible values of deflection and slope at the supports. The expressions for maximum transmitted forces at both super positions of bending and torsion are

$$F_B = [11.2Ae^{-1.5\theta b} - 2m_r\omega^2 a(\cos\theta_b + \cos 2\theta_b/n)]/\cos\emptyset_b \ldots\ldots (2)$$
$$F_T = [11.2Ae^{-1.5\theta t} - 2m_r\omega^2 a(\cos\theta_t + \cos 2\theta_t/n)]/\cos\emptyset_t \ldots\ldots (3)$$

The reciprocating force is doubled because of the two opposed pistons are connected to one crankpin. Also the force is subtractive since the crank angle is less than 90^0 and the piston group is accelerating from zero on account of the gas pressure.

Assuming that the transmitted resultant force acts through the center of the crankpin supported at the web ends, the resolution of maximum shear force and bending moment resulting from the force, F_B is
R, maximum shear force at supports = $0.5F_B$.
M_B maximum bending moment at pin center = $0.25Rl_{cp}$
To reduce weight of the crankshaft, the pins and journals are designed hollow which also serves as lubrication oil way. The maximum diameter of the inner hole, d_{cpi} against shear failure is

$$d_{ci} = d_{co} - (4R/\pi\tau_{all})^{0.5} \ldots\ldots (4)$$

Then the section modulus of the crankpin, Z_{cp} is

$$Z_{cp} = \pi\ (d^4_{cpo} - d^4_{cpi})/32d_{cpo} \ldots\ldots(5)$$

Checking the crankpin for safety against failure, the bending stress σ_{cp}on the pin is

$$\sigma_{cp} = M_B/Z_{cp} < \sigma_{all}\ .$$

The extension, e of the crankpin into the web hole for fixture sufficient to withstand compressive crush of gas pressure is determined as

$e = R/\sigma_{cp}d_{co}$ .Therefore the total length of the crankpin is

$$L_{cp} = l_{cp} + 2e.$$

Unlike the crankpin, the state stress in the main journal is biaxial of torsion and bending stress modes in combination, the Von-Mises failure criterion will apply in the design of the main journal.
Taking center-to-center distance of the bore as 1.5D, then the length from bore center to main journal, l_{cc} axially is

$$l_{cc} = 0.5\ (\ 1.5D\ )$$

The journal is designed hollow with internal diameter of 8 mm for oil way. Then the maximum bending stress on the journal, σ_{xj} is calculated as

$$\sigma_{xj} = M_B/Z_j = 32\ F_B l_{cc} d_{oj}/\pi(d^4_{oj} - 8^4)\ \text{MPa} \ldots\ldots (6)$$

And the maximum torsional stress, τ_{xyj} is calculated as

$$\tau_{xyj} = Td_{oj}/2J_j = 16F_T a d_{oj}/\pi(d^4_{oj} - 8^4) \text{ MPa} \quad \ldots (7)$$

The expression for Von misses yield criterion gives the equivalent uniaxial stress, σ_v as

$$\sigma^2_v = \sigma^2_{xj} + 3\tau^2_{xyj} \quad \ldots (8)$$

For safety the allowable yield stress, σ_{all} must be greater than equivalent Von-Mises stress, σ_v.

$$\sigma_v = [(32\, F_b l_{cc} d_o/\, \pi(d_o^4 - 8^4))^2 + 3(16F_t a d_o/\pi(d_o^4 - 8^4))^2\,]^{0.5} \quad \ldots (9)$$

The permissible bearing pressure, P_i on the journal is 35 MPa, and then the width of the bearing bed is deduced as

$$l_j = F_B/P_i.d_{o,max}$$

To find the width of the octagonal section on the main journal, the shear plane of the triangular wedge against torsion is assumed to be located at ½ the distance from the tip of the triangle to the base circle.

Hence the shear plane on the triangular wedge is
$0.25\,(d_{o,max} - d_{o,min})$

Therefore the width of the octagonal section, e_j for main journal is
$e_j = F_T\,/8(0.25\,(d_{o,max} - d_{o,min})\tau_{all}$.The total thickness of the web, t_w is $t_w = e + e_j$

The web is subjected to direct compressive stress at super position of 10^0 crank angle and bending stress at super position of 74^0crank angle. The minimum areas bearing the crush are the areas of ring section of the web parallel to the crankshaft axis. To determine the safe width of the web, b_w the thickness of these areas is added to the maximum diameter of the main journal hole. The minimum bearing area is located on the crankpin side. Hence the width of the bearing area b, is
$b = F_B/2(e)\sigma_{all}$and them $b_w = (d_{o,max} + 2b)$.

To check for bending, the width of the web adequacy for web bending is calculated as
$b_w= [6\, F_T a/\, t^2_w\, \sigma_{all}]^{0.5}$

The web height, h_w is
$h_w = 0.5(D + D_{o,max} + D_{co}) + 2b$

To reduce the cost of production, the counter weight for balancing the reciprocating forces is machined integral with the web as rectangular beam attachment. Let the length and breadth of the balancer be (0.08+2x) m and x mm respectively with the center of gravity at 0.5x mm. then the area and volume of the balancer are,
Area = $(0.08x + 2x^2)$ m^2, Volume = $0.015(0.08x+2x^2) = (0.0012x + 0.03x^2)$ m^2.
0.08 m and 0.015 m are deduced width and thickness of the web respectively.The mass and the radius of rotation of the balancer are
$|m| = 7850(0.0012x+0.03x^2) = (235.5x^2+9.42x)$ kg, $|r| = (0.03906+0.5x)$ m.
Condition for balance is, $(m_p a) = |m|.|r|$, hence
$(0.3)(0.0363) = (235.5x^2+9.42x)(0.03906+0.5x)$. This yields a cubic equation as

$$117x^3 + 13.908x^2 + 0.368x - 0.01089 = 0 \quad \ldots (10)$$

Solving the equation gives the dimensions of the counter weight.

VI.LIMITS OF FITS OF THE FITS OF THE ASSEMBLY

The critically stressed portion of the assembly crankshaft is the outer ring of the web at the main journal fit. The fitting required for the crankpin-web and main journal-web is medium drive fit specified as H7 – p6 fits.

The nominal diameter of the crankpin-web fit is d_{cw} and the nominal diameter of the main journal-web, d_{mw} is taken as 0.5 $(d_{max} + d_{min})$

From the table of Fit tolerance, the allowable size variation for H7 fit is, Δd = 0.025mm and 0.03mm respectively for cw, mw.

From the table of deviation, the lower deviation for p-fits is
δ = + 0.026mm, 0.032mm respectively for cw, mw fits.

For the cw fit the maximum and minimum hole sizes, D and shaft sizes, d is calculated as
$d_{cw-min} = d_{cw}$
$d_{cw-max} = d_{cw} + 0.025$
$D_{cw-min} = d_{cw} + 0.026$
$D_{cw-max} = d_{cw} + 0.026 + 0.025$
Maximum interference, $I_{cw\text{-}max} = 0.051$mm.

For the mw fit the maximum and minimum hole and shaft sizes are
$D_{mw\text{-}min} = d_{mw}$
$D_{mw\text{-}max} = d_{mw} + 0.030$
$D_{mw\text{-}min} = d_{mw} + 0.032$
$D_{mw\text{-}max} = d_{mw} + 0.030 + 0.032$
$I_{mw\text{-}max} = 0.062$mm

The fit- pressure compresses the shaft and expands the hole thereby weakening the hole ring. The fit pressures are expressed are expressed as

$$P = [(c^2 - b^2)(b^2 - a^2)/2b^3(c^2 - a^2)].\ IE \quad \ldots\ldots(11)$$

Where b = nominal diameter of the fit = d_o
a = inner diameter of the shaft = d_i
c = outer diameter of hole ring for web = $d_o + 2b$

$P_{cw} = [(66^2 - 42^2)(42^2 - 30^2)/2(42)^3(66^2 - 30^2)](0.061)(2x10^5)$
$= 44.6$MPa
$P_{mw} = [(78.12^2 - 52.06^2)(52.06^2 - 8^2)/2(52.06^3)(78.12^2 - 8^2)](0.062)(2 \text{ x } 10^5)$
$=65.3$ MPa
Torque transmitted as a result of m-w fit in the web is
$T_{mw} = 2\pi.\mu b^2 lP_,$ where mean friction coefficient, $\mu = 0.14$

This indicates that the torque ratio is 0.17 (T_T/T_{mw}) as compared to a value not exceeding 0.14 for a pure press fitted assembly. At near the close value for pure fit, it is assumed that the prevailing torque ratio is adequate to arrest slip-fret action at the octahedral mating of m-w fit.

The radial stress on the hole rings are
$\sigma_{cw\text{-}r} = -P_{cw}$
$\sigma_{mw-r} = -P_{mw}$
The tangential stresses on the hole rings deduced from the expression

$$\sigma_\theta = (c^2 + b^2)\ P/(c^2 - b^2) \quad \ldots\ldots (12)$$

The resulting Von-Mises stress due to fit pressure is given as

$$\sigma_{r\theta} = 0.707[(\delta_\theta - \delta_r)^2 + \delta^2{}_\theta + \delta^2{}_r]^{1/2} \quad \ldots\ldots (13)$$

The critical equivalent stress due to fit press, $\sigma_{mw\text{-}r\theta}$ should be less than material yield strength, σ_{all} for safety.
To determine the temperature difference required during assembly of the fits, the expression for area expansion required to accommodate the maximum interference with allowance is given as
$\Delta T = (1/\alpha_A A_f)(A_f - A_i)$

Where α_A is the area expansitivity of material = $2\alpha_L = \alpha_A$
The resulting dimensions for the press fit is specified as
C-W fit: Hole diameter = $42.00 \pm^{0.025}_{0.000}$ mm
Shaft diameter = $42.026\pm^{0.025}_{0.000}$ mm
M-W fit:
Hole diameter = $50.00\pm^{0.030}_{0.000}$ mm based on the minimum diameter before machining octahedron.
Shaft diameter = $54.12 \pm^{0.030}_{0.000}$ mm based on the maximum diameter before machining octahedron.

VII. FATIGUE ANALYSIS

As earlier stated, the crankshaft is variably loaded in repeated cyclic manner. Failures in crankshafts result from fluctuating stresses yet analysis reveals that the prevailing maximum stresses are well below the material yield strength. Such failures termed fatigue occur in machine parts after repetitions of load cycle over a high number of times. Hence explicit fatigue checks should be performed in addition to the implicit initial stress analysis for a successful design. The parts of the crankshaft are designed for infinite life at high cycles of over 10^6.

Before calculating the fatigue strength of the material under operating conditions, the fatigue ratio and several fatigue modifying factors need to be specified. These factors are;

1. Θ_f, fatigue ratio for ductile steels = 0.506
2. K_a, surface factor for machined parts = $4.45\,(S_{ut})^{-0.265}$, $K_a = 4.45\,(1500)^{-0.265} = 0.64$
3. K_b = Size factor based on minimum diameter = $1.24d^{-0.107}$, $K_b = 1.24(42)^{-0.107} = 0.83$
4. K_c = loading factor for rotational bending is unity. $K_c = 1$
5. K_d, temperature factor is taken as unity since the mean operating temperature (250^0C) is below tamper temperature (400^0C). $K_d = 1$

The fatigue strength of machined plain pin and journal is

$S_e = k_a\, k_b\, k_c\, k_d\, \Theta_f\, S_{ut} = (0.64)(0.83)(1)(1)(0.506)1500 = 403$ MPa

6. For fatigue safety factor of 1.2, the allowable fatigue strength, $S_{all}=S_e/1.2 = 335$ MPa

For lubrication of the bearings, it is assumed that the crankpin and main journal have transverse hole diameters of 4mm and 5mm respectively. Inner holes in these parts are linked by 5mm diameter hole drilled in longitudinally at the midsection of the web. Due to transverse oil holes, the fatigue stress concentration factors are required to evaluate prevailing stresses in the parts.

Data in terms of graphs or tables are used to find the static stress concentration factors (k_t) which is applied in relation to deduce the fatigue stress concentration factors (k_f). The expression relating static and fatigue stress concentration is

$$K_f = K_t / [1 + 2(k_t - 1)\sqrt{a}/(\sqrt{r}k_t)] \quad \ldots\ldots(14)$$

$\sqrt{a} = 174/S_{ut}$ for transverse holed part

r = radius of the hole

The fatigue stress on the crankpin due to bending is

$\sigma_{fcp} = K_f\, M_b/Z_{cp}$.For safety against fatigue failure, σ_{fc} should be less than S_{ne} (335 MPa).

The failure criterion adequate for the combined loading in the main journal is Mises-Gerber, where the alternating equivalent stress is given as

$$\sigma_a = (16/\pi d^3)[4(k_f\, M_b)^2 + 3(k_{fs}T_t)^2]^{0.5} \ldots\ldots(15)$$

Let $d = d_0(1 - (d_i/d_o)^4)^{1/3} \approx d_o$ for hollow shafts.

The alternating equivalent stress is compared with Gerber locus give as

$$\sigma_a/S_e + (\sigma_m/S_{ut})^2 = 1 \quad \ldots\ldots (16)$$

In horizontally opposed engine arrangement, the mean stress, $\sigma_m = 0$, so the logic reduce to

$$\sigma_a/S_e = 1 \quad \text{or} \quad \sigma_a \leq S_e \text{ for design.} \quad \ldots\ldots (17)$$

The alternating stress should be less than the material fatigue strength for safety of the main journal against fatigue failure.

VIII. ANALYTICAL RESULTS

Table 4. Results of dimensions

Member	No of pieces	Dimensions (nominal)
Crankpin	2	Outer diameter = 42 mm Inner diameter = 30 mm Length= 60 mm Transverse oil hole = 4mm
Central main journal	1	Outer diameter = 54.12 mm Apothem of octal-prism = 50 mm Inner diameter =8mm Length = 48.9mm Length of machined ends = 10mm Transverse oil hole = 5mm
End main journal	2	Length = 34.45 mm Single end machining, non-drill through centre
web	4	height =110mm width =80mm thickness =15mm oil way drill diameter
Counter weights	Integral with webs	Length = 120 mm Width = 20mm Thickness = 15 mm

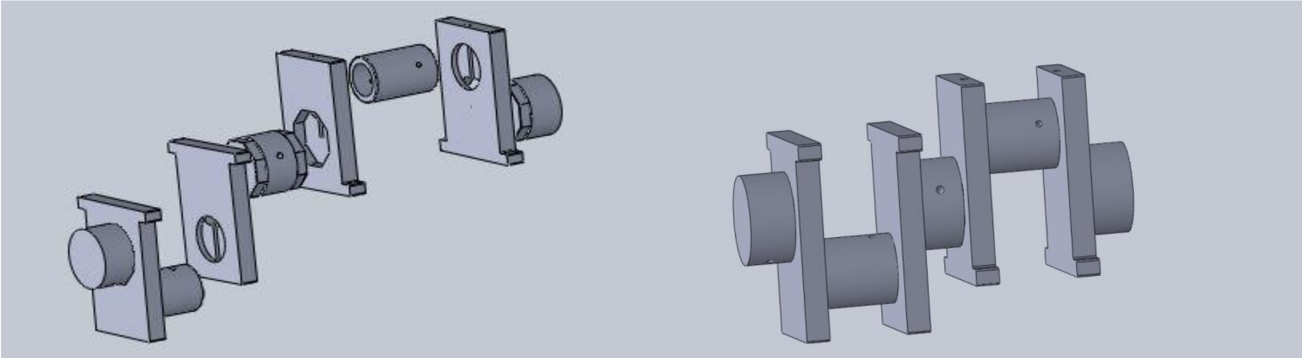

Figure 1. Exploded model and assembled model of the multi piece crankshaft

IX. CONCLUSION

The successful achievement of the design goals will offer the following benefits to the power plant;

- Reduce bearing failures in automotive crankshafts by feasible utilization of full ring metal bearings.
- Improve the ease of crankshaft manufacture by utilizing press fitting assembly
- Due to light weight and ease of manufacture, the adaptation of the crankshaft in automotive industry reduces cost of production and man hours of production.
- Reduce weight of the crankshaft offers improved engine performance and overall engine fuel conversion efficiency.
- Increasing engine reliability reduces the engine maintenance cost.

The work aims to proffer an effective mass production technique that reduces production time in manufacture of crankshafts. The application of this design is expected give improved engine performance and durability in comparison to existing cast or forged ones. It is also expected that this design will reduce the cost of crankshaft production thereby lowering the cost of the engine.

REFERENCES

[1] Y.Zhang, B McClain and X.D. Fang "Design of interference fits via finite element method", International journal of mechanical sciences 42(2000) 1835 -1850.july 1999
[2] C.E. Truman and J.D.Booker "Analysis of a shrink failure on a gear hub- shaft assembly", Elsevier, Science Direct, Engineering failure analysis, 14(2007)557-572. May 2006
[3] M. Y. Sun, S. P. Lu, D. Z. Li, Y. Y. Li, X. G. Lang and S. Q. Wang. "Three dimensional finite element method simulation and optimization of shrink fitting process for large marine crankshaft", Elsevier-Material and Design 31(2010) 4155-4161
[4] F.H Montazersadgh and A. Fatemi "Dynamic load and stress analysis of a crankshaft",SAE International journal, 2007-01-0258.2007
[5] RinkleGarg and Sunil Baghla "Finite element analysis and optimization of crankshaft design", International journal of engineering and management research, volume-2 Issue -6, pages 26-31. December 2012
[6] S.S. Shelkle, C.L. Dhamejani and A.S .Gadhave "Modelling and analysis of the crankshaft using ANSYS software", International journal of engineering science and computing, IJESC, volume - 6 Issue -6.pages 6796 -6801,June 2016
[7] M53 paper publication "Calculation of crankshafts for I.C. engines", 2nd Revision, January 2011
[8] A. Solanki and J. Dodiya "Design and stress analysis of crankshaft for single cylinder 4 stroke Diesel engine", IJRASET vol. 2, issue no 5, May 2014
[9] S. Ho, Y. Lee, H. Kang and C. Wang "Optimization of a crankshaft rolling process for durability", Elsevier-International Journal of Fatigue 31(2009) 799-808
[10] M. Fonte, V. Anes, P. Duarte, L Reis and M. Freitas "Crankshaft failure analysis of boxer diesel motor", Elsevier-Engineering Failure Analysis 56(2015) 109-115
[11] M. Leitner, Z. Truncali, R. Steiner and F. Grun "Multi axial fatigue strength assessment of electro slag remelted 50CrMo4 steel crankshafts", Elsevier-International Journal of Fatigue 100(2017) 159-175
[12] F. J. Espadafor, J.B. Villanueva and M.T. Garcia "Analysis of a diesel generator crankshaft failure", Elsevier-Engineering Failure Analysis 16(2009) 2333-2341

Books

[13] J.E. Shigley and C. R. Mischke "Mechanical Engineering Design", McGraw Hill 6th Ed, 2003
[14] "Mark's Standard Handbook for Mechanical Engineers", McGraw Hill 11th Ed, 2010
[15] J. B. Heywood, "Internal Combustion Engine Fundamentals", McGraw Hill 2nd Ed 1988
[16] M. Kutz "Mechanical Engineers' Handbook", John Wiley 2nd Ed 1998
[17] W.W. Pulkrabek "Engineering Fundamentals of Internal Combustion Engines", Prentice Hall 1994
[18] K. Houg and B. Dondlinger "Vehicular Engine Design", Springer 2nd Ed 2016
[19] A. Kolchin and V. Demidov "Design of Automotive Engines", Mir 2nd Ed 1984
[20] J. A. Collins, H. R. Busby and G. H. Staab "Mechanical Design of Machine Elements and Machines" John Wiley 2002
[21] R. B. Patil "Machine Design II" Tech Max GTU 2015

.

