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Design and Analysis of EN19 Centre Crankshaft

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Abstract
Crankshaft is the one of the most important components in internal combustion engines, it converts the translation motion of piston generated due to gas force into rotational moment of engine shaft though oscillatory motion of connecting rod. It mostly experiences the bending and torsional moment. The present research is focused on investigation of alternative material for crankshafts. Material EN19 is considered in this study and strength performance is verified under different operating conditions. Analytical design calculations, 3D modelling and analysis of EN19 crankshaft is carried out under maximum bending moment and maximum torque. 3D modeling and analysis was done in Creo parametric software and Ansys respectively. The compressive stresses, Von-Mises stresses, shear stresses and deformation in crankshaft are found to be within the permissible limits. Result shows cost effective material EN19 can be used as one of the alternative materials for crankshaft as it satisfies the all-strength requirements.

Keywords
centre crankshaft, bending moment, torque, FEA, EN19 material

1 Introduction

The main component of IC engine which converts the translatory motion of piston into rotational moment through connecting rod is called crankshaft. There are two types of crankshafts mostly used in engines which are centered crankshaft and side crankshaft [1]. The main components of crankshaft are crank pin, which is connected to big end of connecting rod, shaft which is fitted in bearings with flywheel and last one is web which connects the crank pin and shaft. The side crankshaft consists of one web, and it is placed by used two bearings whereas center crankshaft consists of two webs and three bearings. Mostly side crankshafts are used in medium and large size engines whereas centered crankshafts are mostly used in automotive engine, stationary engine aircraft engine. The design of centered crankshaft is done by using the two cases such as maximum bending moment and no torsional moment when piston is at top dead centered position and second condition of maximum torsional moment when crank is at some angle to line of dead centers positions [2–4]. Force exerted by connecting rod on crankpin, weight of flywheel and resultant belt tension these are the three forces acts on crankshaft. There are so many materials for crankshaft out of which EN19 [5] is used. The FEA helps to find out stress and deformation [6–10] in structural components.

2 Design calculation

The design calculation of crankshaft is done according to the standard procedure for the design of the center crankshaft in "Design of machine element" written by V. B. Bhandari [1].

2.1 Input data

Table 1 shows the important inputs to design of crankshaft. If the dimensions of b, b_1, b_2, c, c_1, c_2 and l_c/d_c ratio are not given then it can be calculated by using empirical relation.

2.2 Case I - Centre crankshaft at top dead centered position

The assumptions we have made for this case are engine:

1. Engine is vertical and crank is at the TDC.
2. Belt drive is horizontal.
3. Crankshaft is simply supported on bearing 1, 2, 3.

2.2.1 Bearing reactions

The reaction on bearing 1, 2 and 3 are denoted by R_1, R_2 and R_3 respectively. The suffix letter v and h denotes vertical and horizontal component of reaction. The reaction at bearing 1 and 2 is due to forces acting on crank pin and reaction at bearing 2 and 3 is due to weight of flywheel and belt tension and it is denoted by R'_2 and R'_3 as shown in Fig. 1.

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Table 1 Input data

Parameters	Symbol	value
No, of cylinders	n	1
Maximum speed	N	10000 rpm
Maximum HP	HP	13.5 kW
Maximum gas pressure	$P_{\max}$ or p'	3 MPa
Maximum torque	$\mathrm{T}_{\max}$	14.2 N m
Bore	D	58 mm
Stroke length	L_s	58.7 mm
Displacement	V_s	155 cc
Crank length	r	29.35 mm
Length of connecting rod	L	100 mm
Rod ratio	(L/r)	3.41
Angle of inclination of crankshaft with line of dead centers	θ	35°
By empirical relation ($b = 2 \times D$)		
Distance between 1 and 2 bearings	b	116 mm
Distance between bearing 1 and crankpin center	b_1	58 mm
Distance between bearing 2 and crankpin center	b_2	58 mm
Distance between 2 and 3 bearings	c	200 mm
Distance between bearing 2 and flywheel center	c_1	100 mm
Distance between bearing 3 and flywheel	c_2	100 mm
Weight of flywheel	W	350 N
Total belt tension	$(P_1 + P_2)$	1500 N
l_c/d_c ratio	l_c/d_c	1

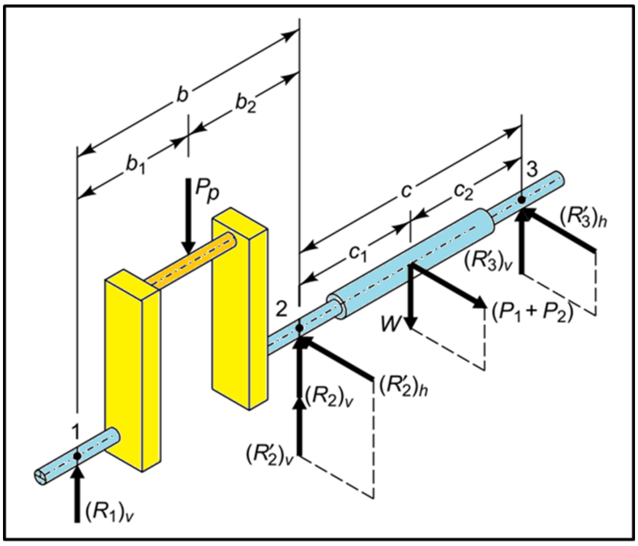

Fig. 1 Centre crankshaft at TDC (adapted from [1])

Force acting on piston (P_p) is:

$$P_p = \left(\frac{\pi D^2}{4}\right) P_{\max} = 7926.24 \text{ N} .$$

To calculate reaction at bearings is as follows:

$$(R_2)_v = \frac{P_p \times b_1}{b} = 3963.12 \text{ N}$$

$$(R_1)_v = \frac{P_p \times b_2}{b} = 3963.12 \text{ N}$$

$$(R_3')_v = \frac{W \times c_1}{c} = 175 \text{ N}$$

$$(R_2')_v = \frac{W \times c_2}{c} = 175 \text{ N}$$

$$(R_3')_h = \frac{(P_1 + P_2) \times c_1}{c} = 750 \text{ N}$$

$$(R_3')_h = \frac{(P_1 + P_2) \times c_2}{c} = 750 \text{ N} .$$

The resultant reactions at the bearing 1, 2 and 3 are given as follow:

$$R_1 = (R_1)_v = 3963.12 \text{ N}$$

$$R_2 = \sqrt{\left[(R_2)_v + (R_2')_v\right]^2 + \left[(R_2')_h\right]^2} = 4205.54 \text{ N}$$

$$R_3 = \sqrt{\left[(R_3')_v\right]^2 + \left[(R_3')_h\right]^2} = 770.15 \text{ N} .$$

2.2.2 Design of crank pin

Fig. 2 shows the nomenclature and loading of crank pin and as per assumption it is subjected to maximum bending moment.

To calculate the diameter of crank pin (d_c) is:

$$(M_b)_c = (R_1)_v \times b_1 = 229860.91 \text{ N mm}$$

$$d_c = \sqrt[3]{\frac{(M_b)_c \times 32}{\pi \times \sigma_b}} = 31.49 \text{ mm} .$$

Length of crank pin (l_c) is:

$$l_c = 1 \times d_c = 31.49 \text{ mm} .$$

To calculate allowable bearing pressure (P_b) is:

$$P_b = \frac{P_p}{d_c \times l_c} = 7.99 \text{ MPa} .$$

2.2.3 Design of left-hand crank web

From empirical relationship, the thickness of crank web is:

$$(t) = 0.7 \times d_c = 22.04 \text{ mm} ,$$

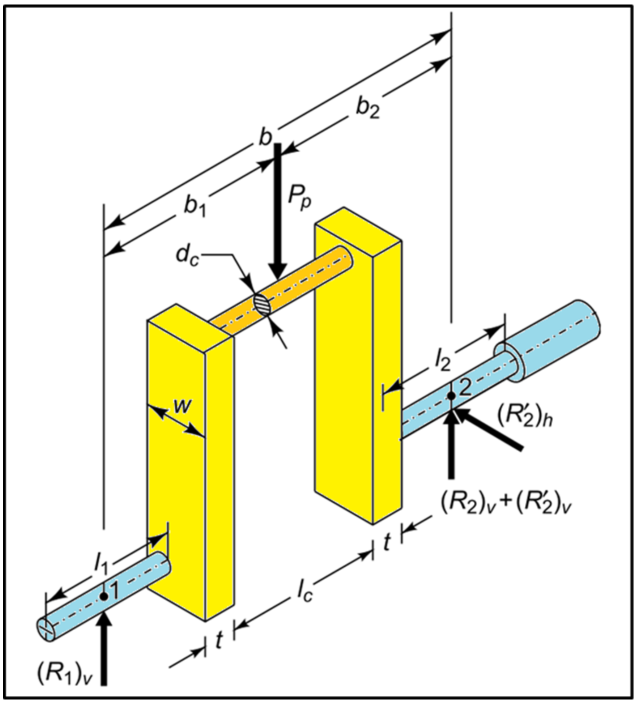

Fig. 2 Crank pin and web (adapted from [1])

the width of crank web is:

$$(w) = 0.7 \times d_c = 35.90 \text{ mm},$$

to calculate direct compressive stress (σ_c) is:

$$\sigma_c = \frac{(R_1)_v}{wt} = 5.01 \text{ MPa},$$

to calculate maximum bending moment (σ_b) is:

$$\sigma_b = \frac{6(R_1)_v \times \left[b_1 - \frac{l_c}{2} - \frac{t}{2}\right]}{wt^2} = 42.59 \text{ MPa},$$

the total compressive stress $(\sigma_c)_t$ is given by

$$(\sigma_c)_t = \sigma_c + \sigma_b = 47.6 \text{ MPa}.$$

2.2.4 Design of right-hand crank web

The left-hand and right-hand web dimensions are kept same because of balancing consideration point of view.

2.2.5 Design of shaft under flywheel

Fig. 3 shows the nomenclature and loading of crankshaft under flywheel.

The maximum bending moment in vertical plane $(M_b)_v$ is:

$$(M_b)_v = (R'_3)_v \times c_2 = 17500 \text{ N mm}.$$

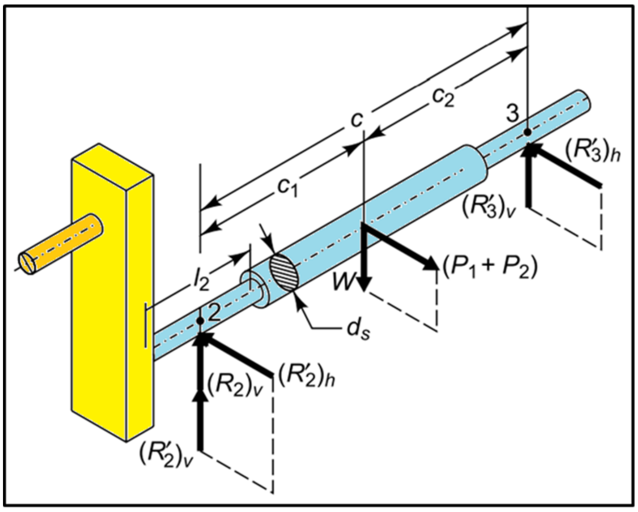

Fig. 3 Shaft under flywheel (adapted from [1])

The maximum bending moment in horizontal plane $(M_b)_h$ is:

$$(M_b)_h = (R'_3)_h \times c_2 = 75000 \text{ N mm}.$$

The resultant bending moment (M_b) is given as

$$M_b = \sqrt{(M_b)_v^2 + (M_b)_h^2} = 77014.61 \text{ N mm}.$$

To calculate diameter of shaft (d_s) is:

$$d_s = \sqrt[3]{\frac{M_b \times 32}{\pi \times \sigma_b}} = 21.87 \text{ mm}.$$

2.3 Case II - Centre crankshaft at angle of maximum torque

2.3.1 Components of forces on crank pin

As shown in Fig. 4 the crankshaft is rotate at an angle of θ from line of piston dead centers which causes two force tangential force (P_t) and radial component (P_r) on crank pin component on crankshaft. As the engine used in this research is petrol engine the maximum torque is obtain at the crank angle range of 25°–35° and we assume maximum torque obtain at angle of 35°.

To calculate the angle of inclination of connecting rod (ϕ) with line of dead centers is:

$$\phi = \sin^{-1}\left(\frac{\sin\theta}{(L/r)}\right) = 9.69°.$$

To calculate thrust acting on connecting rod (P_q) is:

$$P_q = \frac{P_p}{\cos\phi} = 8040.99 \text{ N}.$$

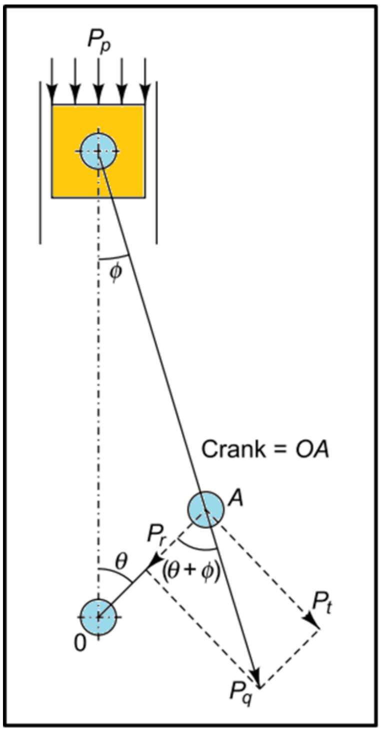

Fig. 4 Force acting on crank (adapted from [1])

To calculate tangential force component (P_t) is:

$$P_t = P_q \sin(\theta + \phi) = 5655.16 \text{ N}.$$

To calculate radial force component (P_r) is:

$$P_r = P_q \cos(\theta + \phi) = 5716.37 \text{ N}.$$

2.3.2 Bearing reaction

The Fig. 5 shows the forces acting on the crankshaft for maximum toque condition. The reaction on bearing 1, 2 and 3 are denoted by R_1, R_2 and R_3 respectively. The suffix letters v and h denote vertical and horizontal component of reaction. The reaction at bearing 1 and 2 is due to forces acting on crank pin and reaction at bearing 2 and 3 is due to weight of flywheel and belt tension and it is denoted by R'_2 and R'_3.

To calculate reactions of bearing are as follows:

$$(R_2)_h = \frac{P_t \times b_1}{b} = 2827.58 \text{ N}$$

$$(R_1)_h = \frac{P_t \times b_2}{b} = 2827.58 \text{ N}$$

$$(R_2)_v = \frac{P_r \times b_1}{b} = 2858.18 \text{ N}$$

$$(R_1)_v = \frac{P_r \times b_2}{b} = 2858.18 \text{ N}$$

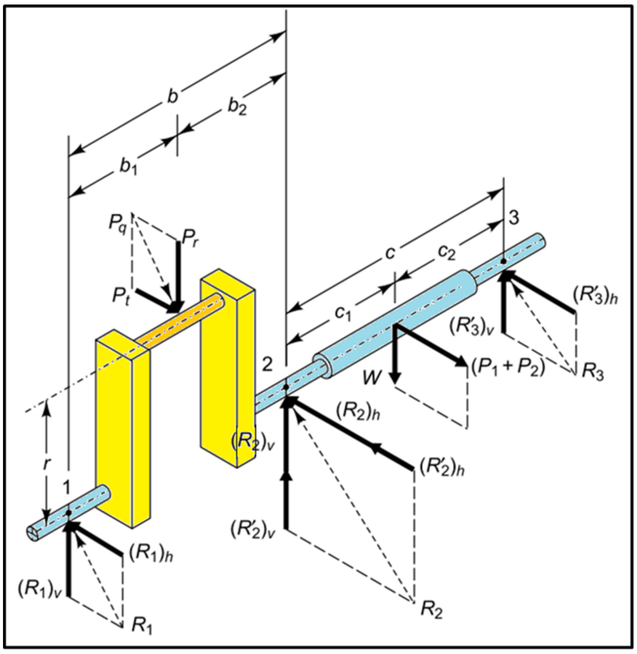

Fig. 5 Centre crankshaft at angle of maximum torque (adapted from [1])

$$(R'_3)_v = \frac{W \times c_1}{c} = 175 \text{ N}$$

$$(R'_2)_v = \frac{W \times c_2}{c} = 175 \text{ N}$$

$$(R'_3)_h = \frac{(P_1 + P_2) \times c_1}{c} = 750 \text{ N}$$

$$(R'_2)_h = \frac{(P_1 + P_2) \times c_2}{c} = 750 \text{ N}.$$

The resultant reactions at the bearing 1, 2 and 3 are given as follow:

$$R_1 = \sqrt{\left[(R_1)_v\right]^2 + \left[(R_1)_h\right]^2} = 4020.50 \text{ N}$$

$$R_2 = \sqrt{\left[(R_2)_v + (R'_2)_v\right]^2 + \left[(R_2)_h + (R'_2)_h\right]^2} = 4690.34 \text{ N}$$

$$R_3 = \sqrt{\left[(R'_3)_v\right]^2 + \left[(R'_3)_h\right]^2} = 770.15 \text{ N}.$$

2.3.3 Design of crank pin

As shown in Fig. 6 the load is applied at a crank pin at an angle which causes the bending as well as twisting of crank pin.

The maximum bending moment (M_b) and twisting moment (M_t) are calculated as

$$M_b = (R_1)_v \times b_1 = 165774.65 \text{ N mm}$$

$$M_t = (R_1)_h \times r = 82989.42 \text{ N mm}.$$

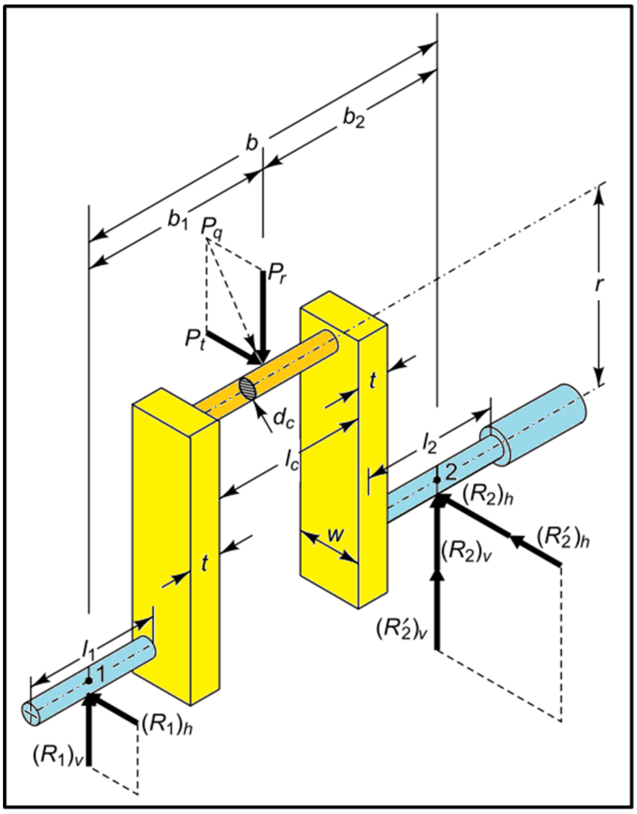

Fig. 6 Crank pin and web (adapted from [1])

To calculate diameter of crank pin (d_c) is:

$$d_c = \left(\frac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2}\right)^{\frac{1}{3}} = 28.67 \text{ mm} .$$

Length of crank pin (l_c) is:

$$l_c = 1 \times d_c = 28.67 \text{ mm} .$$

To calculate allowable bearing pressure (P_b) is:

$$P_b = \frac{P_q}{d_c \times l_c} = 9.77 \text{ MPa} .$$

2.3.4 Design of shaft under flywheel

As shown in Fig. 5 the forces acting on flywheel i.e., self-weights and belt tension.

Bending moment (M_b) due to reaction R_3 is:

$$M_b = (R_3) \times c_2 = 77015 \text{ N mm} .$$

Twisting moment (M_t) due to tangential component P_t is:

$$M_t = P_t \times r = 165978.84 \text{ N mm} .$$

To calculate diameter of shaft (d_s) is:

$$d_s = \left(\frac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2}\right)^{\frac{1}{3}} = 28.56 \text{ mm} .$$

2.3.5 Design of shaft at the juncture of right-hand crank web

The empirical relationships are:

$$t = 0.7 \times d_c = 20.08 \text{ mm}$$

$$w = 1.14 \times d_c = 33.56 \text{ mm} .$$

Bending moment in vertical plane $(M_b)_v$ and in horizontal plane $(M_b)_h$, twisting moment M_t, and the complete bending moment M_b are:

$$(M_b)_v = (R_1)_v \left[b_1 + \frac{l_c}{2} + \frac{t}{2}\right] - P_r\left[\frac{l_c}{2} + \frac{t}{2}\right]$$
$$= 96084.333 \text{ N mm}$$

$$(M_b)_h = (R_1)_h \left[b_1 + \frac{l_c}{2} + \frac{t}{2}\right] - P_t\left[\frac{l_c}{2} + \frac{t}{2}\right]$$
$$= 93562.96 \text{ N mm}$$

$$M_t = P_t \times r = 165978.84 \text{ N mm}$$

$$M_b = \sqrt{\left[(M_b)_v\right]^2 + \left[(M_b)_h\right]^2} = 134112.74 \text{ N mm} .$$

The diameter of shaft at juncture of right-hand crank web (d_{s1}) is:

$$d_{s1} = \left(\frac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2}\right)^{\frac{1}{3}} = 30.06 \text{ mm} .$$

2.3.6 Design of right-hand crank web

The bending moment $(M_b)_r$ due to radial component is:

$$(M_b)_r = (R_2)_v \left[b_2 - \frac{l_c}{2} - \frac{t}{2}\right] = 96084.33 \text{ N mm} .$$

The bending stress due to radial component $(\sigma_b)_r$ is:

$$(\sigma_b)_r = \frac{(M_b)_r}{Z} = \frac{6 \times (M_b)_r}{wt^2} = 42.60 \text{ MPa} .$$

The bending stress due to tangential load $(\sigma_b)_t$ is:

$$(\sigma_b)_t = \frac{(M_b)_t}{Z} = \frac{6 \times P_t\left[r - \frac{d_{s1}}{2}\right]}{wt^2} = 21.48 \text{ MPa} .$$

The direct compressive stress due to radial component of force $(\sigma_c)_d$ is:

$$(\sigma_c)_d = \frac{P_r}{2wt} = 4.24 \text{ MPa} .$$

The total maximum compressive stress (σ_c) is:

$$\sigma_c = (\sigma_b)_r + (\sigma_b)_t + (\sigma_c)_d = 68.92 \text{ MPa} .$$

2.3.7 Design of left-hand crank web

The left-hand and right-hand web dimensions are kept same because of balancing consideration point of view.

2.3.8 Design of crankshaft bearing

Because of flywheel weight, belt tension and force on crank pin the bearing 2 subjected to maximum stresses and its reaction is given as:

$$R_2 = \sqrt{\left[(R_2)_v + (R_2')_v\right]^2 + \left[(R_2)_h + (R_2')_h\right]^2} = 4690.34 \text{ N}.$$

The length of bearing is calculated by bearing consideration:

$$l_2 = \frac{R_2}{d_{s1} \times P_b} = 15.97 \text{ mm} \approx 16 \text{ mm}.$$

From design calculation the dimension of crankshaft for 3D modelling are $d_c = 32$ mm, $l_c = 32$ mm, $t = 23$ mm, $w = 36$ mm, $d_s = 30$ mm, $l_2 = 16$ mm.

3 Finite element analysis of crankshaft

3.1 Geometry

The Fig. 7 shows the geometry of center crankshaft which is model in Creo parametric.

3.2 Meshing

The tetrahedral meshing is used with 8 mm element size as shown in Fig. 8. The total number of element and nodes obtained in meshing is 7027 and 11668 respectively.

3.3 Boundary conditions

As shown Fig. 9 the simple support is applied at the 3 bearings location as per our assumption.

3.4 Loading in case I

A vertical load of 7926.24 N is applied at crank pin and weight of pulley and belt tension of 350 N and 1500 N is applied in flywheel location respectively in Fig. 10.

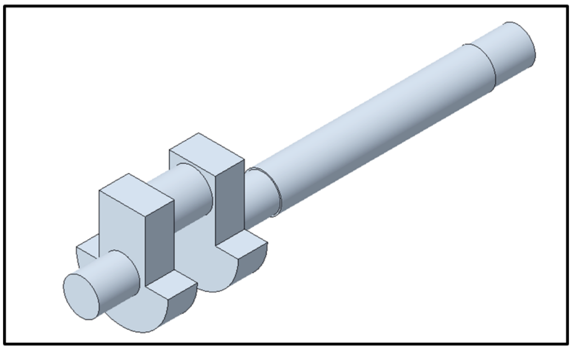

Fig. 7 Geometry

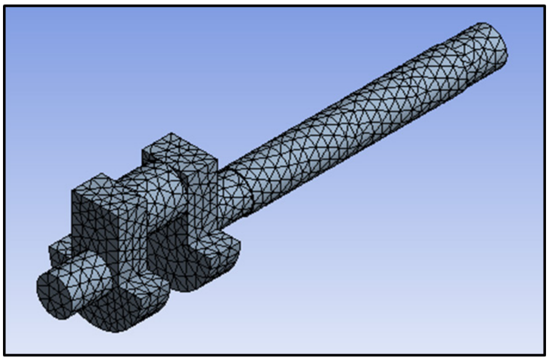

Fig. 8 Meshing

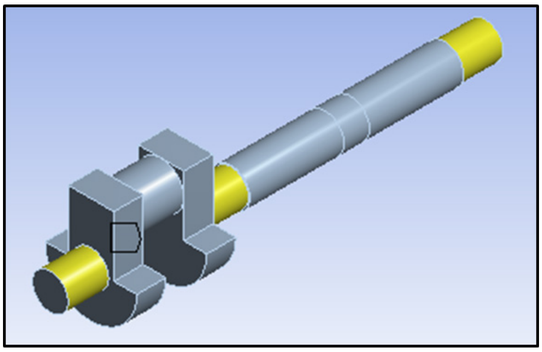

Fig. 9 Boundary conditions

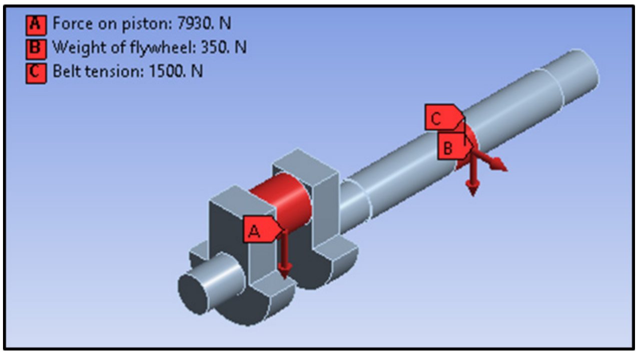

Fig. 10 Loading in case I

3.5 Deformation in case I

The Fig. 11 shows the deformation of crankshaft at maximum bending moment i.e., when piston is at top dead center position. The maximum deformation of 0.10329 mm is occurring at web of crankshaft.

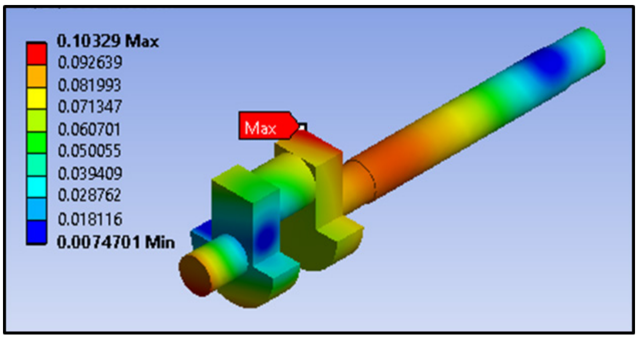

Fig. 11 Deformation in case I

3.6 Von-Mises stress in case I
As shown in Fig. 12 the Von-Mises stress generated inside the crankshaft is 59.064 MPa at a junction of web and shaft.

3.7 Shear stress in case I
The shear stress of 31.918 MPa is generated inside the crankshaft as shown in Fig. 13 for maximum bending stress condition.

3.8 Loading in case II
Fig. 14 shows the loading of crankshaft in case II where load is applied on crank pin when crankshaft is rotate at an angle of 35°.

3.9 Deformation in case II
The deformation of 0.11019 mm is occurring at a web of crankshaft as shown in Fig. 15.

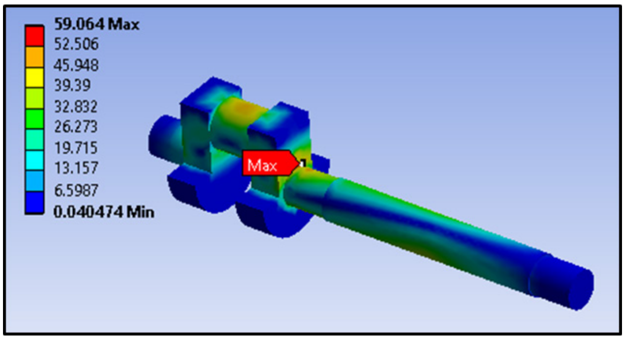

Fig. 12 Von-Mises stress in case I

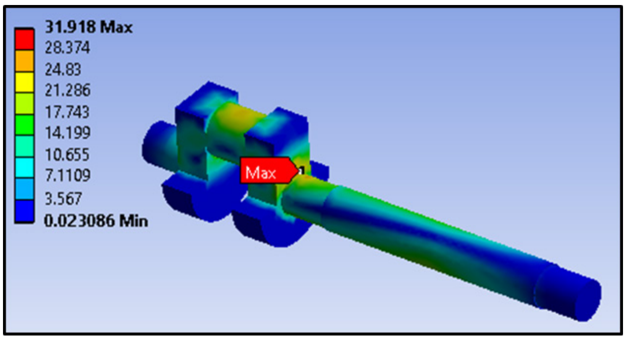

Fig. 13 Shear stress in case I

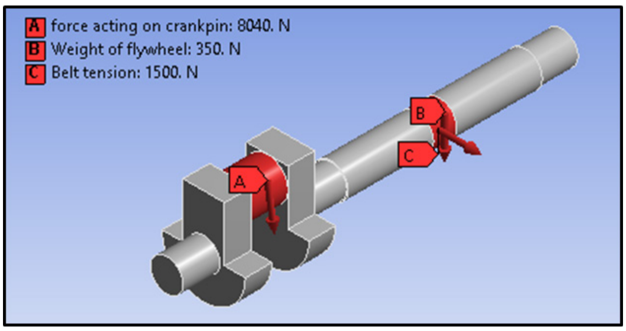

Fig. 14 Loading in case II

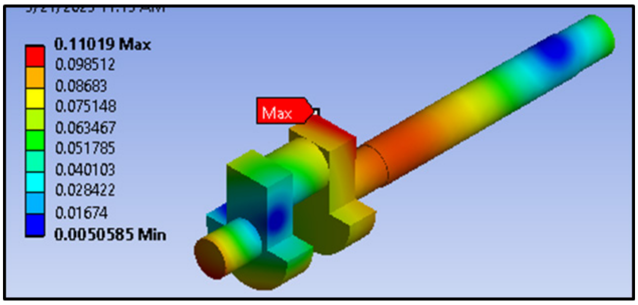

Fig. 15 Deformation in case II

3.10 Von-Mises stress in case II
As shown in Fig. 16 the Von-Mises stress of 61.982 MPa is occurring at junction of web and shaft.

3.11 Shear stress in case II
The shear stress of 33.452 MPa is occurring at the junction of web and shaft as shown in Fig. 17.

4 Result
Table 2 shows the result of crankshaft in analytical as well as FEA. The stress generated in both cases in FEA is close to analytical design. The shear stress calculated in FEA in case I is 31.91 MPa and in case II is 33.45 MPa

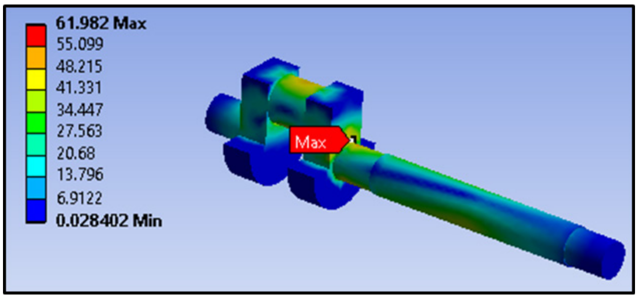

Fig. 16 Von-Mises stress in case II

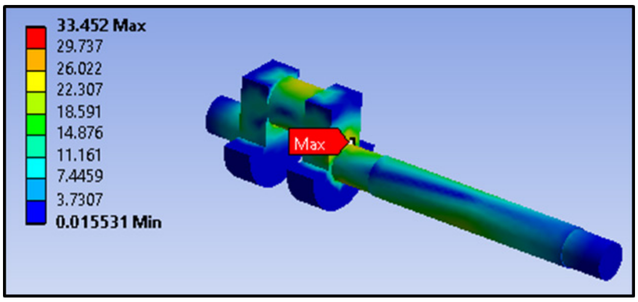

Fig. 17 Shear stress in case II

Table 2 Results

Sr. no.	Parameters	Analytical	FEA
1	Total compressive stress/ Von-Mises stress Case I	47.60 MPa	59.06 MPa
2	Total compressive stress/ Von-Mises stress Case II	68.92 MPa	61.98 MPa

which is very less as compared to material properties, so it confirms the design is safe. Also, the deformation in both the cases is very less. In general materials used to produce crankshafts are EN8 and AISI E4340 with ultimate strength of 415 MPa and 470 MPa respectively. In this study proposed material is EN19 with ultimate and yield strength of 870 MPa and 660 MPa respectively. Proposed material EN19 is low-cost material as compared to EN8 and AISI E4340. Performance comparison with previous studies [6–9] and current analysis shows EN19 is one of the alternative materials for crankshaft as it satisfies the all-strength requirements.

5 Conclusion

Design and analysis of center crankshaft is carried out in two critical conditions that is maximum bending moment and maximum torque. A cost-effective material EN19 is considered for crankshaft and strength performance is verified. The compressive stresses, Von-Mises stresses, shear stresses and deformation in crankshaft are found to be within the permissible limits. Result shows stresses generated in maximum torque condition are higher and crank pin diameter found to be optimum in condition of maximum bending moment, so designing of crankshaft in both conditions are important. The higher stresses are observed in maximum torque condition at a junction of web and shaft, that can be optimized by providing a taper at this location. The equal web thickness and width helps to keep balancing in rotation of crank. In center crankshaft the middle bearing experiences the maximum stresses so this bearing must be carefully selected. Result shows cost effective material EN19 can be used as one of the alternative materials for crankshaft as it satisfies the all-strength requirements.

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