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Optimization And Analysis of a Single-Cylinder Four-Stroke Engine Crankshaft for A 3-Wheeler Vehicle

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Abstract

The rising use of three-wheeler cars in commercial transportation in Nigeria has created a growing need for durable, cost-effective engine parts that can be adjusted to local operating conditions. The majority of the crankshafts that are in use today are imported and engineered to work on smoother roads, and thus they are prone to premature failure in such conditions in Nigeria, as the roads are harsh in nature, the ambient temperatures are high, the loads on the shock, and the lubrication are issues. This paper aimed at optimization and analysis of a crankshaft of a single-cylinder four-stroke engine of a TVS King Deluxe Plus three-wheeler, which is specifically designed to operate on the Nigerian roads. This methodology was physical measurement of crankshaft that existed, analytical calculation of bending and torsional stress, selection of material using mechanical properties, fatigue resistance, corrosion resistance, vibration damping and cost considerations. It was modeled in three dimensions in SolidWorks and Finite Element Analysis (FEA) was done in ANSYS to measure von Mises stress distribution, cumulative deformation as well as thermal strain when subjected to mixed loading of mechanical and thermal loads. A comparative evaluation was conducted on two candidate materials AISI 1045 steel and SG 700/2 ductile iron. The findings showed that higher levels of stress were recorded at the crankpin fillet areas. The optimized design was found to have improved stress distributions and fatigue performances and structural integrity. Based on the ANSYS results, SG 700/2 seemed to have better fatigue strength and demonstrated to be more efficient in vibration-damping. The local production viability was backed by cost analysis. The research paper offers a context-based solution to the design of the crankshaft, which improves reliability, lowers reliance on imported components, and promotes sustainable production of automotive components in Nigeria.

Keywords
Finite Element Analysis (FEA), Crankshaft, Computer-Aided Design (CAD), Fatigue, Von Mises Stress.

1. Introduction

The focus of this present study on computer-aided design, optimization through simulation, material selection, and optimization of geometry is in tandem with the increasing demand of mechanical resiliency, economical nature, and sustainability in the automotive sector of Nigeria. Over the years, the use of three-wheeler vehicles has become more rampant across the country and not just in the commercial sector as they've been widely known for, but the concept of three-wheeler vehicles has now also been implemented in carrier vehicles, personal vehicles, utility vehicles, and specialized vehicles. These parts are designed factoring in stresses based on the smooth road conditions of the regions in which they are produced. Overloading has proven to be one of the major causes of faults arising in mechanical systems, and this overload can occur as a result of cyclic load applied on parts of the mechanical systems, of which in vehicles, the terrain is a major contributor to this. Overloading may lead to extensive and excessive wear, breakdown of machinery, or structural damage. More power is required from the engine to travel on bad roads, hence the overload on the crankshaft which transmits the power. Overloading has proven to be one of the major causes of faults arising in crankshafts, from the damage on the oil lubrication system, to the physical stress overload on the crankshaft which can lead to cracks. The state of the roads lead to increased high stress concentrations and high operating oil temperature which will eventually lead to the wearing out of the crankshaft.

1.1 Objectives

The focus of this project is to optimize, and analyze a three-wheeler vehicle crankshaft that will deliver optimally with Nigerian environmental conditions and stresses. The goal is not to introduce new parts to the crankshaft but to develop an improved practical and cost-effective working model of a three-wheeler single-cylinder four-stroke crankshaft suitable for Nigerian roads. The case study is a TVS King Deluxe Plus model.

2. Literature Review

There is a need for optimization in the production of crankshafts in three-wheeler vehicles, seeing that most of the countries where these vehicles are produced are not subject to the same environmental and road conditions as many underdeveloped areas such as Nigeria. Hence, there is the need to design crankshafts which are more compatible with Nigeria's environmental conditions to get a better output and fewer failure cases. According to (O'Connor and Kleyner 2012), reliability is defined as the probability that an item will perform a required function without failure under stated conditions for a stated period of time.

Internal combustion engines attain power output through crankshafts because these components function as the main connection between piston movements and engine power systems. The rotating motion of the pistons gets transformed by the crankshaft to rotational motion until it reaches the drivetrain (Heywood 2018). Increased stiffness of the crankshaft causes less flexing under pedaling which results in an improved power transfer. The major parameters which affect this are the crankshaft length, journal diameter, and crankpin overlap (Kane 2020). The crankshaft requires strong design and superior materials since cyclic loads increase its vulnerability to fatigue according to (Rezaei et al. 2010). The essential functions essential for its importance include correct crankshaft balancing functions as an NVH control system to diminish engine vibrations that brings better vehicle comfort and enhanced performance levels (Giri 2013).

The optimized design of crankshafts decreases several engine-related losses to improve fuel economy but also decrease emissions according to (Peters and Timings 2003). Temperature and geometry requirements for the crankshaft determine its capabilities in power distribution and load supervision through precise transmission of high combustion forces (Akıncı et al. 2012). Research investigates the use of micro-alloyed steels and titanium and composite-reinforced metals as alternative materials for demanding high-performance applications (Peters and Timings 2003). By adopting FEA, (Akıncı et al. 2012) conducted an experiment on stress distribution and critical failure point analysis to enhance geometric optimization while preventing weight increase. The assessment of natural frequencies and torsional vibrations and resonance potential make use of modal and harmonic analysis techniques (Giri 2013).

Through structural optimization and the selection of durable, low-cost materials, this study focuses on the structural optimization and cost-effective material selection of critical components in three-wheeled vehicles, particularly the single-cylinder four-stroke engine crankshaft designed for operation on Nigerian roads. Considering harsh service conditions such as pothole-induced shock loads, vibration, corrosion, and thermal stress, the components are modeled, simulated, and analyzed using suitable alternative materials that enhance durability while remaining affordable.

With the help of simulation software like AutoCAD, SolidWorks and ANSYS, it is now possible to test mechanical components in a more detailed way with structural, thermal and fatigue analysis. The parametric modeling is the fundamental functionality of modern CAD systems to ensure that designers can establish geometrical and working constraints, which generate flexible robust designs (Shah and Mäntylä 1995). Solid modeling and surface modeling techniques became mature to provide precise constructions of prismatic and intricate freeform forms that are used in aerospace as well as consumer product industries (Mantyla, 1988; Farouki, 1991). The interrelationship of CAD with CAM and CAE applications made this drafting tool a complete solution of engineering (Chang et al. 2005). FEA provides accurate and efficient solutions to stress analysis along with strain analysis and deformation analysis in the mechanical components (Zienkiewicz and Taylor 2005). The simulation of a single-cylinder diesel engine crankshaft shaft using ANSYS software established that the highest stresses were concentrated in the fillet area where the crankpin made contact with the web. These scholars pointed out that the form of transitions has a direct impact on the points of stress concentration (Patel et al. 2014). Through their work, FEA was applied to analyze a forged steel crankshaft. The researchers made a combination of the static and fatigue assessment that showed that the dynamic loads are what defines the resistance against fatigue damage in the component. The experimentally measured values of the strain gauges confirmed the simulation results that confirmed the simulation strategies (Balamurugan and Krishnaraj 2012).

The consequences of inadequate design are high levels of vibrations and likely fatigue breakdown. Vibration modes of the crankshaft were examined because they changed the crank throw setups. The modal analysis was used to verify that non-symmetrical crank throw arrangements reduced the point at which systems fail in the presence of resonance but remain stable at these conditions (Solano 2019). The study by Kumar et al. (2021) showed how optimized fillet radii and counter weights incorporated in the crankshaft design can lengthen its fatigue life when it is used in cyclic operation. Combining the computational design and analysis software, careful consideration of the material choice, and the correct manufacturing techniques offers an obvious way to the design of a sustainable and affordable three-wheeler crankshaft. The local environment conditions, choice of the appropriate materials, and crankshaft geometry can be utilized to create crankshafts that could perform best under the Nigerian road conditions.

3. Methods

3.1 Materials Used

a) Crankshaft: SG 700/2
b) Crankshaft: AISI 1045

Crankshafts are usually produced using forged carbon steel of which AISI 1045 is an affordable and suitable material commonly used for crankshafts. In this study, we are proposing another material and manufacturing process which is the casting of a ductile iron called SG 700/2 for the production of crankshafts. The research strategy was done based on real-world conditions, and the constraints faced such as material cost, machinability, thermal resistance, resistance to wear, corrosion, and vibration damping. The approach integrated thorough research and computation, Computer-Aided Design (CAD), and Finite Element Analysis (FEA) for performance validation.

3.2 Research Strategy

The first phase of this project was to source the crankshaft of a TVS King Deluxe Plus three-wheeler (Figure 1), and take live measurements of the crankshaft. After the measurements were taken, thorough research was done in order to select the best material suitable for the crankshaft. Comparison tables for SG 700/2, AISI 1045, and AISI 1040 were then drawn (Table 1- Table 8) for parameters such as material properties, corrosion, thermal fatigue, vibration damping, machinability, fatigue life, and cost analysis. (Figure 1- Figure 15)

Figure 1. A TVS King Deluxe Plus Crankshaft

Table 1. Measured Crankshaft Parts

S/N	Crankshaft Parts	Dimensions (mm)
1	Length of Connecting Rod	130
2	Crankpin Diameter	30
3	Length of Crankpin	54
4	Web Width	119.37
5	Web Thickness	13
6	Diameter of shaft at juncture of right arm	38
7	Diameter of shaft under flywheel (tapered end)	20.06
8	Diameter of shaft under flywheel (main shaft body)	23.3
9	Diameter of shaft under clutch	21
10	Length of shaft under flywheel (flywheel width)	50
11	Length of shaft under clutch (clutch width)	24
12	Length of main bearing	16
13	Center-to-center distance between bearing 1 and 2	70

Table 2. Mechanical Properties of Materials

Material	Ultimate Tensile Strength (MPa)	Yield Strength/0.2% Proof (MPa)	Density (kg/m^3)	Hardness (BHN)	Percentage Elongation %
SG 700/2	700	420 (0.2% Proof)	7100	225-305	2
AISI 1040	580	380	7850	217	18
AISI 1045	630	380	7870	229	15

Table 3. Crankshaft Material Performance Comparison

Material	Type	Corrosion Resistance	Vibration Damping	Thermal Fatigue Resistance	Machinability
SG 700/2	Ductile Cast Iron	Very Good	Excellent	Good	Excellent
AISI 1040	Medium-Carbon Steel	Very Poor	Good	Poor	Very Good
AISI 1045	Medium-Carbon Steel	Poor	Moderate	Moderate	Good

Table 4. Fatigue Analysis for Crankshaft Materials

Material	Fatigue Ratio (k)	Estimated Fatigue Strength $\sigma_e = k \cdot UTS$ (MPa)	Estimated Fatigue Life (Normal Duty)(km)	Estimated Fatigue Life (Nigerian Roads)(km)
SG 700/2	0.35	245	~120,000-150,000	~80,000-110,000
AISI 1040	0.48	278	~180,000-250,000	~140,000-200,000
AISI 1045	0.48	302	~180,000-250,000	~140,000-200,000

Table 5. Total Estimated Cost per unit kg

Material	Cost per kg (₦)	Estimated Manufacturing Cost (₦)	Total Estimated Cost per kg (₦)
SG 700/2	700-850	850 – 1,700	1,550 – 2,550
AISI 1040	600-750	250 – 650	850 – 1,400
AISI 1045	650-800	350 – 850	1,000 – 1,650

I. Engine Specifications

To optimize the crankshaft, we need to obtain the actual loads and stresses undergone by the crankshaft, and for this we must know the engine specifications.

Table 6. Engine Specifications

Parameter	Magnitude
Max. Brake Power	7.5 kW at 4750 RPM
Max. Brake Torque	15.0 Nm at 3000 RPM
Bore	62 mm
Stroke	66 mm
Displacement	199.26 cc
Compression Ratio	9.6:1
Connecting Rod Length	127mm
Connecting Rod Big-end Width	26mm

3.3 Design Calculations

Having selected the crankshaft material to be SG 700/2, we then compute the loads and stresses based on the measured crankshaft dimensions and the engine specifications for the case study, the TVS King Deluxe Plus petrol engine three-wheeler, and then apply the allowable bending and shear stress based on the selected factor of safety for the new material. Given;

Max. Power, BP = 7.5kW at 4750rpm
Max. Torque, T_{avg} = 15.0Nm at 3000rpm
Bore Diameter, D = 62mm
Length of stroke, l = 66mm
Length of connecting rod, L = 127mm
Engine displacement, V_d = 199.26cc
= 0.00019926m^3
= 0.19926dm^3

The factor of safety for the purpose of this study, considering Nigerian road conditions, will be 3.0 instead of the typical 2.0. The reason of the 1.0 increase is to avoid over designing as this part is a precision part. The materials AISI 1040 and 1045 are similar, however, AISI 1045 is the superior material hence we compare with SG 700/2.

Allowable bending stress, $S_{yt} = \dfrac{\text{Yield Strength}}{\text{FoS}}$ ... (1)

Allowable shear stress, $S_{sy} = 0.5S_{yt}$ ... (2)

Table 7. Allowable Stresses for the Proposed Materials

S/N	Material	Allowable Bending Stress, σ_b (MPa)	Allowable Shear Stress, τ (MPa)
1	SG 700/2	140	70
2	AISI 1045	126.67	63.34

Crankpin Diameter, d_c = 30mm
Length of Crankpin, l_{cp} = 28mm (portion of the crankpin between the webs)
Web Width, w = 119.37mm
Web Thickness, t = 13mm
Diameter of shaft at juncture of right arm, d_{s1} = 38mm
Diameter of shaft under flywheel, d_{sf} = 20.06mm
Diameter of shaft under clutch, d_{sc} = 21mm
Length of shaft under flywheel = 50mm (flywheel width)
Length of shaft under clutch = 24mm (clutch width)
Length of main bearing, L_m *(1 and 2)* = 16mm
Center-to-center distance between bearing 1 and 2, b = 70mm

Crank radius, r $= \dfrac{\text{Length of Stroke}}{2}$ ... (3)
$= 33mm$

L/r Ratio $= \dfrac{\text{Length of Connecting Rod}}{\text{Crank radius}}$ ... (4)
$= 3.85$

bmep, $p_m = \dfrac{6.28n_R T_{avg}}{V_d} = 0.946N/mm^2$ ... (5)

Where; $n_R = Number\ of\ crank\ revolutions\ for\ each\ power\ stroke\ per\ cylinder$

Maximum pressure, $p = 9 \times p_m = 8.6N/mm^2$ ... (6)

Case 1: Design of Crankshaft when Crank is at Dead Center

$Piston\ gas\ load, F_P = \dfrac{\pi}{4} \times D^2 \times p = 25.97\text{kN}$ ... (7)

Measured distance between bearing 1 and 2, $b = 70$mm

$$b_1 = b_2 = \frac{b}{2} = 35\text{mm} \quad \text{... (8)}$$

Due to piston gas load, there would be two horizontal reactions at H_1 and H_2 at bearings 1 and 2 respectively

$$H_1 = \frac{F_p \times b_1}{b} = 12.99\text{kN} \quad \text{... (9)}$$

$$H_2 = \frac{F_p \times b_1}{b} = 12.99\text{kN} \quad \text{... (10)}$$

Design of crankpin:

Let $d_c =$ diameter of crankpin
$l_{cp} =$ Length of crankpin (portion of the crankpin between the webs)
$\sigma_b =$ bending stress for crankpin

Bending moment at center of crankpin,

$$M_c = H_1 \cdot b_2 = 454.65kN.mm^2 \quad \text{... (11)}$$

$$\text{Also, } M_c = \frac{\pi}{32}(\text{dc})^3\sigma_b \quad \text{... (12)}$$

$$\sigma_b = 171.52N/mm^2$$

$$\text{Length of Crankpin, } l_{cp} = \frac{F_p}{d_c \times P_b} \quad \text{... (13)}$$

Bearing pressure, $P_b = 30.92N/mm^2$

Design of left-hand crankweb:

Maximum bending moment on crankweb,

$$M_\text{c} = H_2\left(b_2 - \frac{l_\text{cp}}{2} - \frac{t}{2}\right) = 188.36\text{kN} \cdot \text{mm} \quad \text{... (14)}$$

$$\text{Section modulus, } z = \frac{1}{6} \times w \times t^2 = 3362.26 \text{ mm} \quad \text{... (15)}$$

$$\text{Bending stress, } \sigma_b = \frac{M_\text{c}}{\text{Z}} = 5.8\text{N/mm}^2 \quad \text{... (16)}$$

Direct Compressive stress on the crankweb,

$$\sigma_c = \frac{H_1}{w \cdot t} = 8.4 \text{ N/mm}^2 \quad \text{... (17)}$$

$$\text{Total stress on } \sigma = \sigma_b + \sigma_c = 14.2 \text{ N/mm}^2 \quad \text{... (18)}$$

Since total stress on the crankweb is less than allowable bending stresses 140N/mm^2 and 126.67N/mm^2, the left hand crankweb design is safe.

Design of right-hand crankweb:

Dimensions of the left hand crankweb (thickness and width) are made equal to the left hand crankweb.

Design or shaft under the flywheel:

Let $d_{sf} =$ diameter of shaft in mm
$\sigma_b =$ bending stress in N/mm^2
$l_f =$ distance between center of bearing and center of flywheel
$l_\text{shaft} =$ length of shaft between bearing face and flywheel face

The mass of the flywheel is assumed to be 2.6kg (Within the range for small engines)

$$\text{Weight of flywheel, } W_\text{f} = \text{mass} \times \text{acceleration due to gravity} = 25.51\text{N} \quad \text{... (19)}$$

$$Length\ of\ bearing, L_\text{m} = 16\text{mm}$$

$$l_\text{f} = \frac{L_\text{m}}{2} + l_\text{shaft} + \frac{\text{Flywheel Width}}{2} = 41\text{mm} \quad \text{... (20)}$$

$$M_f = W_\text{f} \cdot l_\text{f} = 1045.91\text{N.mm} \quad \text{... (21)}$$

Also, bending moment on shaft (M_f),

$$M_f = \frac{\pi}{32}(d_\text{sf})^3\sigma_\text{b} \quad \text{... (22)}$$

$$\sigma_b = 1.32N/mm^2$$

Due to the weight of the flywheel acting downwards, there will be a vertical reaction V_1 at bearing 1.

$$V_1 = \frac{W_\text{f}(b + l_\text{f})}{b} = 40.46\text{N} \quad \text{... (23)}$$

Design or shaft under the clutch:

Let $d_{sc} =$ diameter of shaft in mm
$\sigma_b =$ bending stress in N/mm^2
$l_c =$ distance between the center of bearing and center of clutch
$l_{\text{shaft}} =$ length of shaft between bearing face and clutch face

The mass of the clutch is assumed to be 2.2kg (Within the range for small engines)

Weight of clutch, $W_{\text{c}} =$ mass $\times$ acceleration due to gravity $= 21.59\text{N}$ ... (24)

$Length\ of\ bearing, L_{\text{m}} = 16\text{mm}$

$$l_{\text{c}} = \frac{L_{\text{m}}}{2} + l_{\text{shaft}} + \frac{\text{clutch Width}}{2} = 34\text{mm} \quad \text{... (25)}$$

Bending moment of the shaft due to weight of clutch

$$M_C = W_{\text{c}} \cdot l_{\text{c}} = 734.06\text{N. mm} \quad \text{... (26)}$$

At dead center, tangential component of the gear is zero, however radial component still exists and causes bending

Bending moment of the shaft due to weight of clutch

$$M_{Frg} = F_{\text{rg}} \cdot l_{\text{G}} = 18.79\text{kN. mm} \quad \text{... (27)}$$

Resultant bending moment on the shaft, $M_{CG} = \sqrt{(M_C)^2 + \left(M_{Frg}\right)^2} = 18.81\text{kN. mm}$ ... (28)

Also, bending moment on shaft (M_{CG}),

$$M_{CG} = \frac{\pi}{32}(d_{\text{sc}})^3\sigma_{\text{b}} \quad \text{... (29)}$$

$\sigma_b = 20.69N/mm^2$

Due to the weight of the clutch acting downwards, there will be a vertical reaction C_2 at bearing 2.

$$C_2 = \frac{W_{\text{c}}(b + l_{\text{c}})}{b} = 40.46\text{N} \quad \text{... (30)}$$

Case 2: Design of Crankshaft When Crank is at an Angle of Maximum Twisting Moment

Let T_{avg} = Maximum torque measured at the crankshaft output (Max. Torque)
$T(\theta)$ = Instantaneous torque generated at a specific crank angle θ
T_{peak} = Maximum value for instantaneous torque usually located between 25º to 30º ATDC for petrol engines

Assuming a Gaussian Model,

$$T(\theta) = T_{\text{peak}} \cdot \exp\left(-\frac{1}{2}\left(\frac{\theta - \mu}{\sigma}\right)^2\right), \theta \in [360°, 540°] \quad \text{... (31)}$$

Where; μ = Crank angle where peak torque occurs (25º ATDC/385º crank angle assumed)
σ = Spread/width of the Gaussian assumed to be 35º
θ = Crank angle in degrees

Since torque is typically zero or negligible outside the power stroke,

Set $T(\theta) = 0$ for $\theta < 360°$ or $\theta > 54\theta°$

$$T_{\text{avg}} = \frac{1}{720}\int_{360}^{540} T(\theta)d\theta \quad \text{... (32)}$$

$T_{\text{peak}} = 80.1\text{Nm}$

∴ If torque peaks at 25°, peak instantaneous torque is 80.1 Nm

This peak torque is used to calculate the pressure at that point

$$T(\theta) = P(\theta) \cdot A \cdot r \cdot \sin(\theta) \quad \text{... (33)}$$

Where; $T(\theta) = T(385) = T_{\text{peak}}\ in\ Nmm$
$P(\theta) = Pressure\ at\ the\ top\ of\ piston\ for\ maximum\ torque\ condition\ in\ N/\text{mm}^2$
$A = Piston\ area\ in\ \text{m}^2$
$r = Crank\ radius\ in\ \text{m}$
$\theta = 385^0\ crank\ angle$

$P(\theta) = 2.0\ \text{N/mm}^2$

$P(\theta) = P'$

$$Piston\ gas\ load\ at\ maximum\ twisting\ moment, F_P = \frac{\pi}{4} \times D^2 \times P' = 6.04\text{kN} \quad \text{... (34)}$$

To find thrust in the connecting rod F_Q, we find the angle of inclination of the connecting rod with the line of stroke (ϕ)

$\sin\phi = \frac{\sin\theta}{L/r}$ ... (35)

$\phi = 6.31°$

Thrust in connecting rod, $F_Q = \frac{F_P}{\cos\phi} = 6.08\text{kN}$ ... (36)

Tangential force acting on the crankshaft due to piston load,

$F_T = F_Q \sin(\theta + \phi) = 3.16\text{kN}$ ... (37)

Radial force acting on the crankshaft due to piston load,

$F_R = F_Q \cos(\theta + \phi) = 5.2\text{kN}$ ... (38)

Due to radial force F_R, there would be two reactions at $b_1 \& b_2$

$H_{R1} = \frac{F_R \times b_1}{b} = 2.6\text{kN}$ ... (39)

$H_{R2} = \frac{F_R \times b_2}{b} = 2.6\text{kN}$ ... (40)

Due to tangential force F_{T}, there would be two reactions at $b_1 \& b_2$

$H_{\mathrm{T}_1} = \frac{F_{\mathrm{T}} \times b_1}{b} = 1.58\text{kN}$ ... (41)

$H_{T_2} = \frac{F_{\mathrm{T}} \times b_2}{b} = 1.58\text{kN}$ ... (42)

Design of Crankpin:

d_c = Diameter of crankpin in mm

Bending moment at the center of crankpin,

$M_c = H_{R1} \times b_2 = 91\text{kN} \cdot \text{ mm}$ ... (43)

Taking moment on crankpin

$T_c = H_{\mathrm{T}1} \times r = 52.14\text{kN} \cdot \text{ mm}$ ... (44)

Equivalent twisting moment on crankpin,

$T_e = \sqrt{(M_c)^2 + (T_c)^2} = 104.88 \times 10^3\text{N} \cdot \text{ mm}$ ... (45)

Equivalent twisting moment T_e,

$104.88 \times 10^3 = \frac{\pi}{16}(d_c)^3\tau$ ... (46)

Where; τ = Shear stress in N/mm^2

$\tau = 19.79 N/mm^2$

Design of Shaft Under Flywheel:

d_{sf} = diameter of the shaft in mm

Bending moment on the shaft will be the same as earlier calculated,

$M_{\mathrm{sf}} = 1045.91\text{N} \cdot \text{ mm}$

Twisting moment on shaft, $T_{\mathrm{sf}} = F_{\mathrm{T}} \times r = 104.28.\times 10^3 \text{ N} \cdot \text{ mm}$ ... (47)

Equivalent Twisting Moment on Shaft, T_{ef}

$T_{\mathrm{ef}} = \sqrt{(M_{\mathrm{sf}})^2 + (T_{\mathrm{sf}})^2} = 104.29 \times 10^3 \text{ N} \cdot \text{ mm}$ ... (48)

Equivalent of Twisting moment T_{ef},

$104.29 \times 10^3 = \frac{\pi}{16}(d_{\mathrm{sf}})^3\tau$ ... (49)

$\tau = 65.8 \text{ N/mm}^2$

From the above, taking the already calculated value of $d_s = 38$ mm, the induced Shear stress is less than the allowable sheer stress for SG 700/2 of 70N/mm^2 but slightly greater than that of AISI 1045 of 63.34N/mm^2, hence, the design in safer for SG 700/2.

Design of Shaft Under Clutch:

d_{sc} = diameter of the shaft in mm

r_{p} = pitch circle radius

L_{G} = distance between center of bearing and center of gear teeth = 11.5mm

Pitch circle diameter, $D_p = m \times Z = 36mm$ ... (50)

Where; m = module (assumed to be 2.25 based on similar engines an corresponds with design)

Z = number of teeth = 16

Tangential force on the shaft due to gear

$F_{\mathrm{tg}} = \dfrac{T_{\mathrm{peak}}}{r_{\mathrm{p}}} = 4488.89N$ ... (51)

Radial force on the shaft due to gear

$F_{\mathrm{rg}} = F_{\mathrm{tg}} \times \tan(\phi) = 1633.83N$ ... (52)

Where; $\phi =$ Pressure angle (assumed to be 20^0, typical for similar modern gears)

Resultant gear force, $F_G = \sqrt{\left(F_{\mathrm{tg}}\right)^2 + \left(F_{\mathrm{rg}}\right)^2} = 4776.98N$ ... (53)

$M_{\mathrm{v}} = M_{\mathrm{c}} + \left(F_{\mathrm{rg}} \times L_{\mathrm{G}}\right) = 19.53k\mathrm{N} \cdot \mathrm{mm}$ ... (54)

Horizontal bending moment on the shaft, M_{h}

$M_{\mathrm{h}} = F_{\mathrm{tg}} \times L_{\mathrm{G}} = 51.6319.53k\mathrm{N} \cdot \mathrm{mm}$ ... (55)

Resultant bending moment on the shaft, M_{sc}

$M_{\mathrm{sc}} = \sqrt{(M_{\mathrm{v}})^2 + (M_{\mathrm{h}})^2} = 55.21k\mathrm{N} \cdot \mathrm{mm}$ ... (56)

Twisting moment on shaft, $T_{\mathrm{sc}} = F_{\mathrm{T}} \times r = 104.28.\times 10^3\ \mathrm{N} \cdot \mathrm{mm}$ ... (57)

Equivalent Twisting Moment on Shaft, T_{ec}

$T_{\mathrm{ec}} = \sqrt{(M_{\mathrm{sc}})^2 + (T_{\mathrm{sc}})^2} = 118.02 \times 10^3\ \mathrm{N} \cdot \mathrm{mm}$ ... (58)

Equivalent of Twisting moment T_{ec},

$118.02 \times 10^3 = \dfrac{\pi}{16}(d_{\mathrm{sc}})^3 \tau$ ... (59)

$\tau = 64.91\ \mathrm{N/mm^2}$

From the above, taking the already calculated value of $d_s = 38$ mm, the induced shear stress is less than the allowable sheer stress for SG 700/2 of 70N/mm^2 but slightly greater than that of AISI 1045 of 63.34N/mm^2, hence, the design in safer for SG 700/2.

Design of the shaft at the juncture of the left-hand crank arm:

This arm is used to design the shafts at the junctures between the webs because it faces the most twisting loads from the gear, and clutch

Let d_{s1} = Diameter of the shaft at the juncture of the right-hand crank arm

Reaction due to radial gear force on the shaft

$R_{\mathrm{gv}} = \dfrac{F_{\mathrm{rg}}(b + L_{\mathrm{G}})}{b} = 1902.25N$ ... (60)

Reaction due to tangential gear force on the shaft

$R_{\mathrm{gh}} = \dfrac{F_{\mathrm{tg}}(b + L_{\mathrm{G}})}{b} = 5226.4N$ ... (61)

Resultant forces at bearing 2, $R_2 = \sqrt{\left(H_{T_2} + R_{\mathrm{gh}}\right)^2 + \left(H_{R_2} + R_{\mathrm{gv}} + C_2\right)^2} = 8.18kN$ ... (62)

Bending moment at juncture of the right-hand crank arm

$M_{s1} = R_2\left(b_2 + \dfrac{l_{cp}}{2} + \dfrac{t}{2}\right) - F_Q\left(\dfrac{l_{cp}}{2} + \dfrac{t}{2}\right) = 329.35 \times 10^3\ \mathrm{N} \cdot \mathrm{mm}$ ... (63)

Twisting Moment at the juncture of the left-hand crank arm

$T_{s1} = F_T \times r = 104.28 \times 10^3\ \mathrm{N} \cdot \mathrm{mm}$ ... (64)

Equivalent Twisting moment at the juncture of the right-hand crank arm

$T_{\mathrm{e}} = \sqrt{(M_{s1})^2 + (T_{s1})^2} = 345.47 \times 10^3\ \mathrm{N} \cdot \mathrm{mm}$ ... (65)

Equivalent Twisting Moment T_e,

$345.47 \times 10^3 = \dfrac{\pi}{16}\left(d_{s_1}\right)^3 \tau$ ... (66)

$\tau = 32.07\mathrm{N}/mm^2$

From the above, taking the already calculated value of $d_s = 38$ mm, the induced shear stress is less than the allowable sheer stresses 70N/mm^2 and 63.34N/mm^2, hence, the design in safe for both materials.

Design of crankshaft bearings:

Bearing 2 is subjected to the heaviest loads and faces maximum stress.

the reaction at this bearing, $R_2 = 8.18kN$

$\therefore$ Total Bearing Pressure $= \dfrac{R_2}{l_{\mathrm{m}} \cdot d_{s1}} = 13.46\mathrm{N/mm^2}$ ... (67)

The bearing pressure is slightly above the safe limit for main journal bearings in four-stroke gas (petrol) engines of $5 - 8.5\ \mathrm{N/mm^2}$ according to Khurmi & Gupta using the already measured value of $d_{s1} = 38$mm, however, recent

engines use bimetal and tri-metal plates for their bearing which can withstand up to 40 – 50 N/mm^2 of pressure in their bearings.
Design of left-hand crankweb:
Let σ_{bR} = Bending Stress in the radial direction
σ_{bT} = Bending Stress in the tangential direction
Bonding moment due to radial component,

$$M_R = \left(H_{R_2} + R_{\mathrm{gv}} + C_2\right)\left(b_1 - \frac{l_{cp}}{2} - \frac{t}{2}\right) = 251.66 \times 10^3 \text{ N} \cdot \text{ mm} \qquad \text{... (68)}$$

Also, Bending moment

$$M_R = \sigma_{bR} \times z \qquad \text{... (69)}$$

$$= \sigma_{bR} \times \frac{1}{6} w \cdot t^2 \qquad \text{... } (\because z = 1/6 \times w \times t^2)$$

$\sigma_{bR} = 74.85 \text{ N/mm}^2$
Bending moment due to tangential component F_T,

$$M_T = F_T\left(r - \frac{d_{s1}}{2}\right) = 44.24 \times 10^3 \text{ N} \cdot \text{ mm} \qquad \text{... (70)}$$

Also, Bending Moment

$$M_R = \sigma_{bT} \times z \qquad \text{... (71)}$$

$$= \sigma_{bT} \times \frac{1}{6} t \cdot w^2 \qquad \text{... } (\because z = 1/6 \times t \times w^2)$$

$\sigma_{bT} = 1.44 \text{ N/mm}^2$
Direct Compressive Stress,

$$\sigma_d = \frac{F_R}{2w.t} = 1.68 \text{ N/mm}^2 \qquad \text{... (72)}$$

Total Compressive stress, $\sigma_c = \sigma_{bR} + \sigma_{bT} + \sigma_d = 77.97 \text{ N/mm}^2$ ... (73)
Twisting Moment on the arm,

$$T = \left(H_{T_2} + R_{\mathrm{gh}}\right)\left(b_1 - \frac{l_{\mathrm{cp}}}{2}\right) = 142.94 \times 10^3 \text{ N} \cdot \text{ mm} \qquad \text{... (74)}$$

Shear Stress on the arm, $\tau = \frac{T}{Z_p} = \frac{4.5T}{w \cdot t^2} = 31.89 \text{M/mm}^2$ ... (75)

Maximum combined stress,

$$(\sigma_c)_{\max} = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2} = 89.36 \text{ N/mm}^2 \qquad \text{... (76)}$$

Since the maximum combined stress is within safe limits, the dimension $w = 119.37$ mm is accepted.
Design of left-hand crankweb:
The dimensions for the left-hand crankweb are the same as for the right-hand crankweb
These calculated stress implications should normally be within the limits for the alloy steel material it was designed for. The allowable bending and shear stress for the forged carbon steel according to Khurmi and Gupta are 75N/mm^2 and 42N/mm^2 respectively, implying a factor of safety of 3 for a crankshaft.

4. Data Collection

Data for this study were collected through a combination of reverse engineering and analytical calculations tailored to Nigerian road conditions as depicted in Table 1.

5. Results and Discussion

The numerical and graphical results are presented in this section.

5.1 CAD Design and SolidWorks Modelling

The 2D design of the newly optimized crankshaft design was created using AutoCAD (Figure 2)

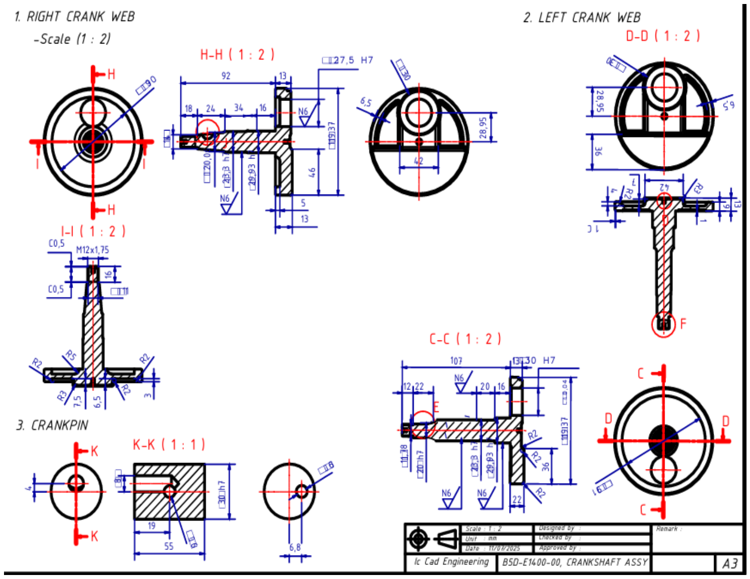

Figure 2. CAD Design of the Crankshaft

I. The 3D design of the newly designed crankshaft was modeled in SolidWorks as a sldprt file after which it was then converted to a STEP file so it could be uploaded into ANSYS Workbench. Then the material SG 700/2 was assigned to it, and the weight was measured. The crankshaft after applying the material was measured to be 0.7 kilograms less than the originally measured crankshaft.

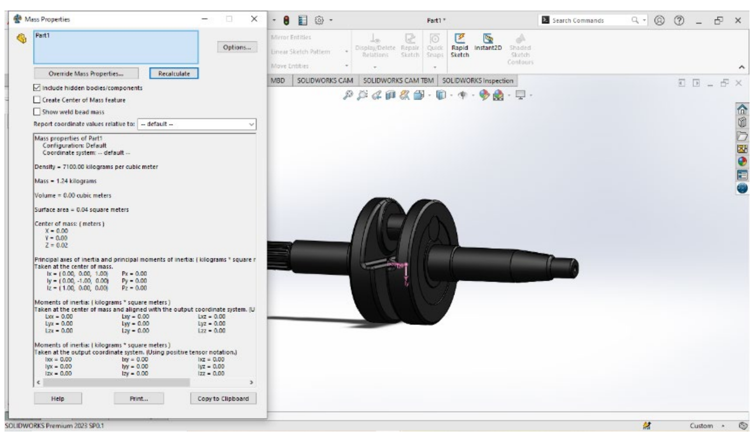

Figure 3. 3D Model of the Crankshaft in SolidWorks

II. Calculated loads to be applied on the crankshaft

Table 8. Loads to be Applied to Crankshaft

S/N	Load	Magnitude (N)
1	Piston gas load, F_p	25,970
2	Flywheel weight, W_f	25.51
3	Clutch weight, W_c	21.59
4	Radial gear force, F_{rg}	1902.25
5	Tangential gear force, F_{tg}	5226.4
5	Resultant Reaction at bearing 1, R_1	3,080
6	Resultant Reaction at bearing 2, R_2	8,180

7	Reactions at bearings 1&2 due to piston gas load, H_1&H_2	12,990

5.2 Material Analysis and Performance Validation

The finite element analysis was carried out after importing the SolidWorks step file into Ansys Workbench. After importing the model, meshing was then carried out on it using and element size of 0.005m to capture all the elements of the crankshaft.

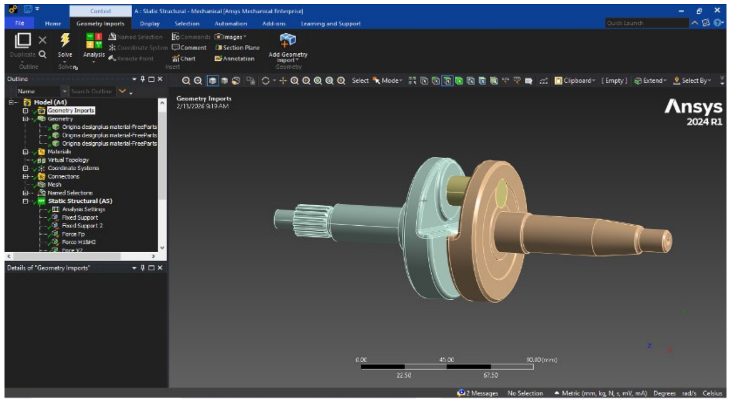

Figure 4. Unmeshed crankshaft model in Ansys Mechanical

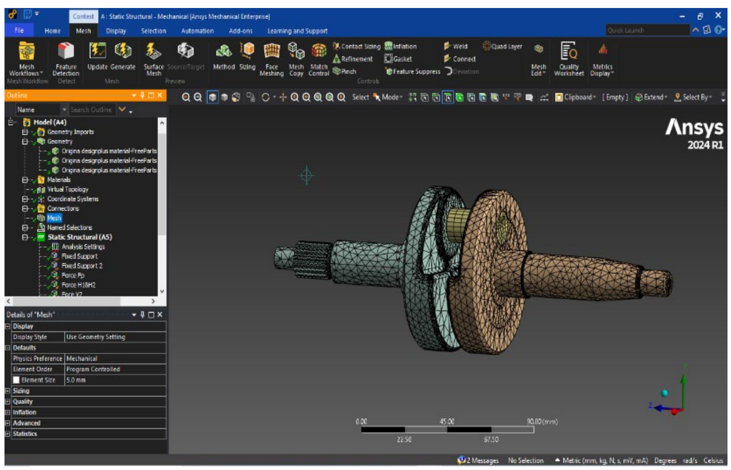

Figure 5. Meshed crankshaft model in Ansys Mechanical

The next step was applying fixed supports on the bearing on the drive end and the fillet at the shaft at the juncture of flywheel end. The forces Fp (piston gas load), W_f (flywheel weight), W_c (clutch weight), F_{tg} and F_{rg} (radial and tangential gear force), R_1 and R_2 (resultant reactions at bearings 1 & 2) and H_1 and H_2 (reaction at bearings 1 and 2 due to piston gas load), were then added to the crankpin, right arm shaft, left arm shaft, bearing 2, bearing 1 & 2, and fillet at bearing 2 respectively. The forces were applied in the direction in which they're acting.

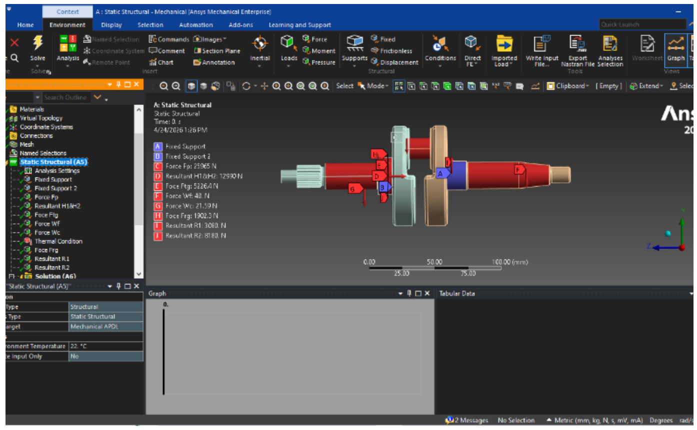

Figure 6. Applied forces on the crankshaft model in Ansys Mechanical

A thermal condition of 40°c was also applied to entire crankshaft to allow for thermal strain analysis.

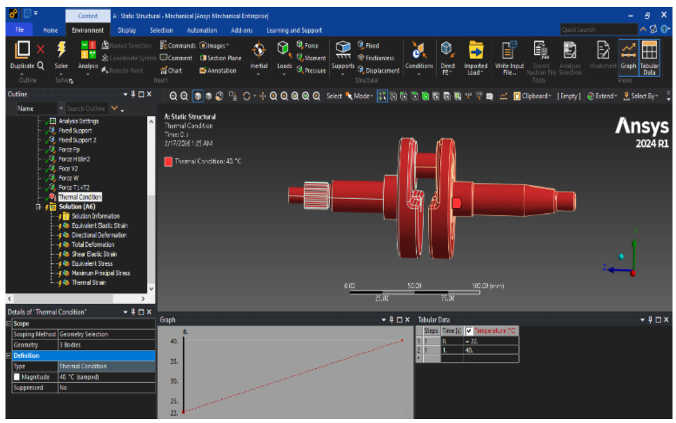

Figure 7. Applied thermal condition of 40⁰ on crankshaft model in Ansys Mechanical

Total deformation, thermal strain, von-mises, and maximum principal stress analysis were then carried out on the model using AISI 1045 and SG 700/2 as the two test materials. The two test materials were experimented on during analysis to determine which would deliver a better fatigue life.

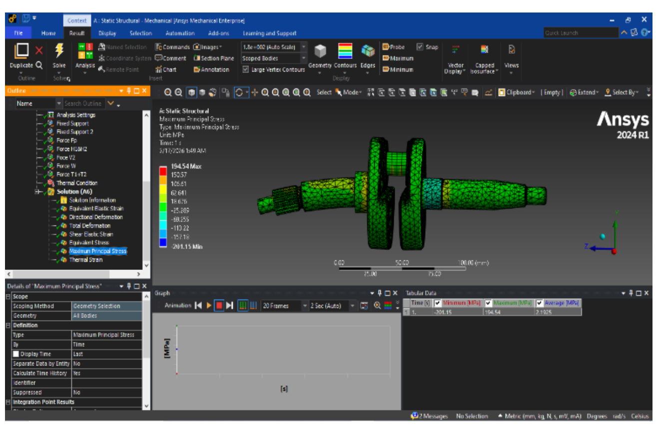

Figure 8. Maximum principal stress analysis for AISI 1045 with maximum stress at the juncture of left arm

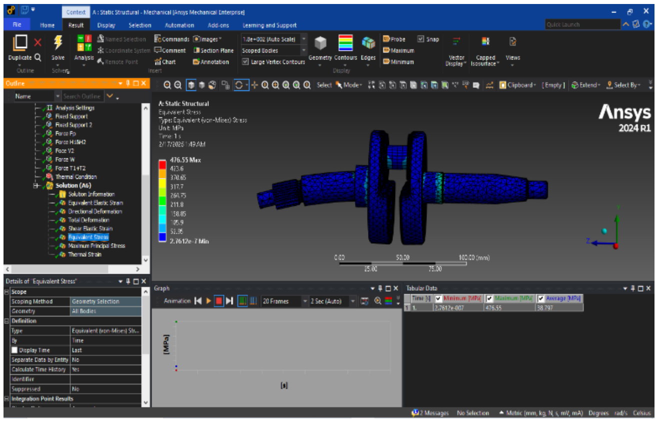

Figure 9. Von-mises stress analysis for AISI 1045 with maximum deformation at the bearings

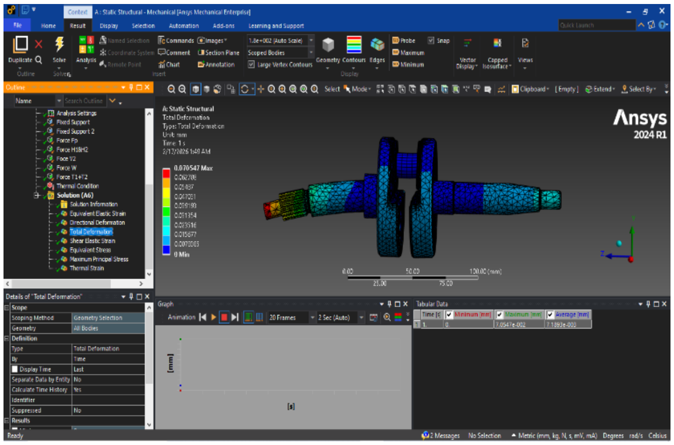

Figure 10. Total deformation analysis for AISI 1045 with maximum deformation at the clutch side snout

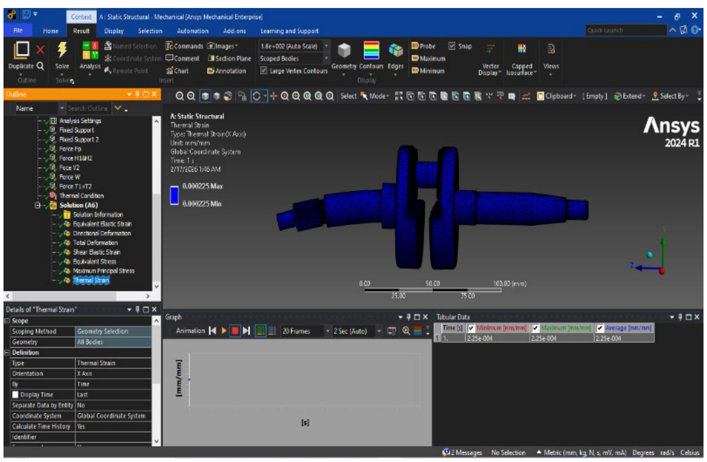

Figure 11. Thermal Strain Analysis for AISI 1045

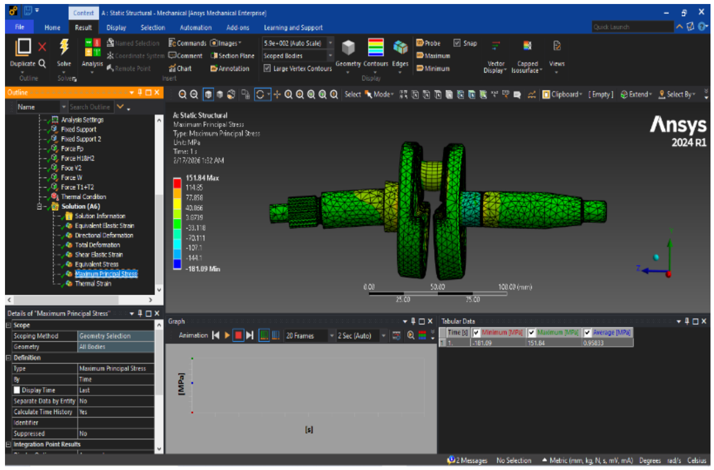

Figure 12. Maximum principal stress analysis for SG 700/2 with maximum stresses at the juncture of left arm

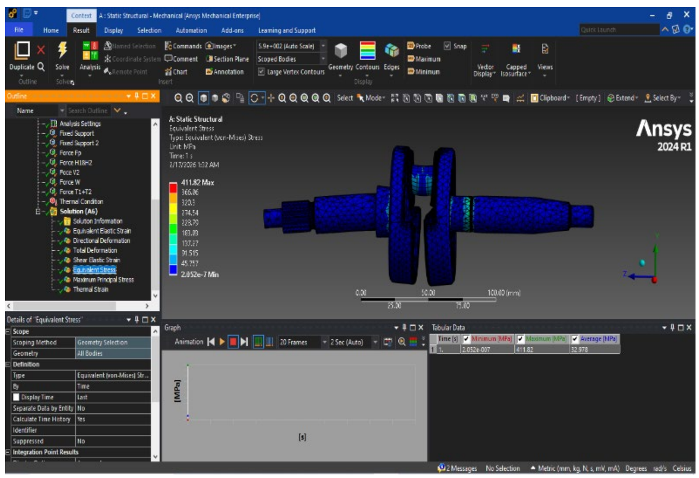

Figure 13. Von-mises stress analysis for SG 700/2 with maximum stresses at the bearings and crankpin

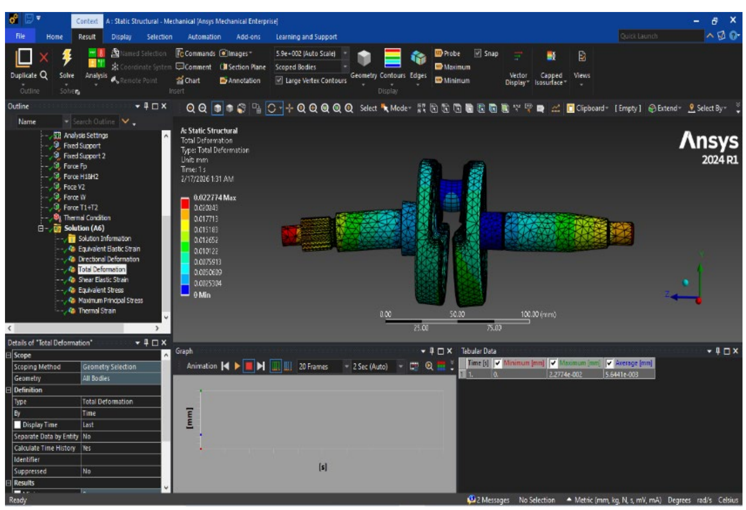

Figure 14. Total deformation analysis for SG 700/2 with maximum deformation at snouts and fillet regions

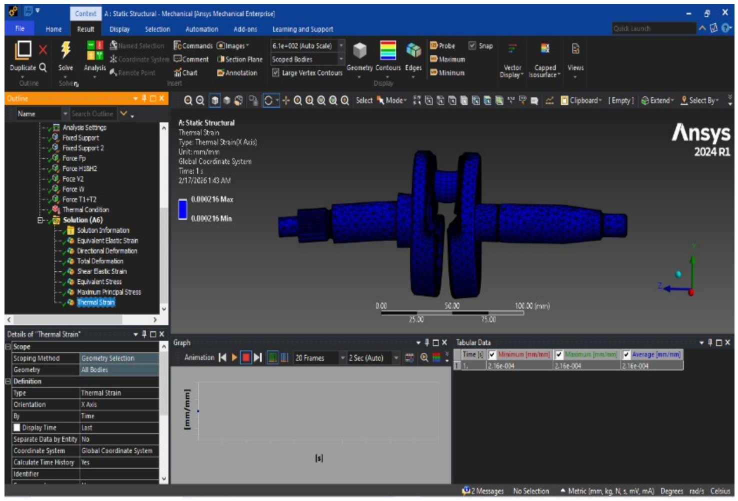

Figure 15. Thermal strain analysis for SG 700/2

From the analysis, it was found that the maximum principal stress and von-mises stress for the SG 700/2, 151.84 and 411.82, respectively, are lower than those of the AISI 1045, 194.54 and 476.55, respectively, due to the material's strength combined with slight ductility, together with its vibration-damping effect, which indicates a higher fatigue life. A design where the maximum principal stress is lower than the yield strength is less likely to fail compared to a material where the opposite is the case. The total deformation, also for the SG 700/2, 0.022774, is also lower than that of the AISI 1045, 0.070547, and the maximum thermal strain for SG 700/2, 0.000216, is also lower than that of AISI 1045, 0.000225. Also, the maximum principal stress experienced in the analysis for SG 700/2, 151.84, is less than its yield strength, 420Mpa, which means the design is safe, hence supporting the decision of SG 700/2 as a better material choice for Nigerian roads.

6. Conclusion

This study successfully designed and analyzed a single-cylinder, four-stroke engine crankshaft for a three-wheeler vehicle (TVS King Deluxe Plus) using Finite Element Analysis (FEA), AutoCAD, and SolidWorks CAD tools. The goal was to produce a crankshaft optimized for Nigerian road conditions, characterized by rough terrain, poor lubrication, and high thermal and vibrational loads. Based on the analysis, it was found that crankshaft performance in such challenging environments is related to several criteria, such as the choice of material, geometric optimization, tolerance of fits, and surface treatments. The materials that were analyzed and compared included SG 700/2 and AISI 1045 in terms of fatigue strength, corrosion resistance, cost, and manufacturability. Among them, SG 700/2 was found to have better mechanical characteristics particularly in vibration damping and wear resistance as well as less costly and reduced bulkiness.

The resulting design was optimized in terms of the reduction of stress concentration regions (e.g., fillets), a lightweight crankshaft geometry which was 0.7kg less than the weight of the original, and a carefully designed oil passage design, resulting in increased fatigue life and reliability. Simulations with FEA were performed at the conditions of the specific loading of the country of Nigeria and it was demonstrated that the crankshaft could work under the normal stresses of shocks and thermal conditions without yielding.

References

Akıncı, T. Ç., Şeker, S., & Gökmen, G., Vibration analysis of a single-cylinder engine crankshaft, *Scientific Research and Essays*, 7(2), 241–248, 2012.

Balamurugan, G., & Krishnaraj, V., Computer aided design and analysis of crankshaft for four stroke single cylinder engine, *Procedia Engineering*, 38, 1893–1902, 2012.

Chang, T. C., Wysk, R. A., & Wang, H. P., *Computer-aided manufacturing (3rd ed.),* Pearson Prentice Hall, 2005.

Farouki, R. T., The Bernstein polynomial basis: A centennial retrospective, *Computer Aided Geometric Design*, 8(1), 1–24, 1991.

Giri, N. K., *Automobile Mechanics (9th ed.),* Khanna Publishers, 2013.

Heywood, J. B., *Internal combustion engine fundamentals (2nd ed.),* McGraw-Hill Education, 2018.

Kane, T. R., *Dynamics: Theory and applications (Reprint ed.), Cambridge University Press, 2020.* (Original work published 1985).

Kumar, R., Singh, D., & Sharma, A. K., Fatigue analysis of forged steel crankshaft with geometry optimization, *Materials Today: Proceedings*, 47(Part 15), 4567–4574, 2021.

Mantyla, M., *An introduction to solid modeling,* Computer Science Press. 1988.

O'Connor, P. D. T., & Kleyner, A., *Practical reliability engineering (5th ed.),* John Wiley & Sons, 2012.

Patel, N., Shah, H., & Patel, D., Analysis of crankshaft for single cylinder 4-stroke diesel engine using FEA, *International Journal of Engineering Research and Applications*, 4(4), 91–94, 2014.

Peters, J., & Timings, R., *Engineering materials and processes,* Elsevier, 2003.

Rezaei, A., Behzad, M., & Haddadpour, H., Nonlinear vibration analysis of a crankshaft system, *Journal of Mechanical Science and Technology*, 24(12), 2415–2423, 2010.

Shah, J. J., & Mäntylä, M., *Parametric and feature-based CAD/CAM,* Wiley, 1995.

Solano, L., Failure analysis of a damaged diesel motor crankshaft, *Engineering Failure Analysis*, 102, 1–6, 2019.

Zienkiewicz, O. C., & Taylor, R. L., *The finite element method for solid and structural mechanics (6th ed.),* Elsevier, 2005.

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