Analysis and Optimization of Drive Shaft in Eccentric Mechanical Press

[1] Rakesh A Oza, [2] S P Patel

[1]Student, M E (CAD/CAM)
Mechanical Engineering Department, Sankalchand Patel College of Engineering, Visnagar.

[2]Assistant Professor,
Mechanical Engineering Department, Sankalchand Patel College of Engineering, Visnagar.

***Abstract*— In current market, drive shaft is most important component to run the application, because of failure of shaft tends to stop the production line or manufacturing of the components. A Drive shaft is a rotating shaft that transmits power from the motor to the gear box. Drive shaft must operate in high and low power transmission of the fluctuating load as required in eccentric mechanical press. Due to this type of fluctuating load drive shaft is fail. Thus it is important to make this shaft as per load requirement to avoid this failure.**

In this paper, first the model is prepare on the pro/Engineer software and after that the analysis work on the ANSYS for comparing the different such as bending stress, shear stress, and deflection of the shaft for existing condition as well as the new design which one developed for this condition. Then weight reduction is check by using E-Glass/Epoxy and HM-Carbon / Epoxy materials.

Keywords: Drive shaft, Torsional buckling, ANSYS software;

I. INTRODUCTION

A driveshaft is a rotating shaft that transmits power from the engine to the differential gear of a rear wheel drive vehicles [1]. Driveshaft must operate through constantly changing angles between the transmission and axle. High quality steel is a common material for construction. In the eccentric press the motor drives the flywheel by means of a belt drive. The energy is transferred from the flywheel via the combined clutch/brake system to the crank shaft. Gear is mounted on the other end of the drive shaft. From that gear mechanism power is transmitted to the eccentric shaft. By the means of the eccentric shaft rotary power is transferred to the linear movement.

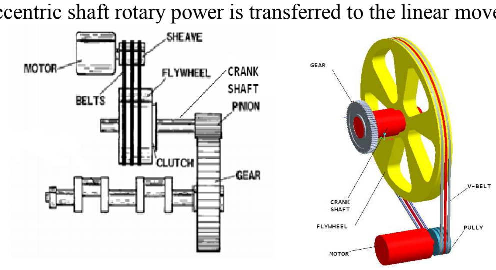


FIG 1: CRANK SHAFT DRIVE MECHANISM

A compressive preload method was developed to reduce the thermal residual stress of a hybrid aluminium/composite[2]. The torsional stability of a composite drive shaft under torsion was studied by Shokrieh et al. [3] using different fibre orientations and stacking sequence. The composite propeller shaft had about 40 per cent weight saving compared with a two-piece steel propeller shaft [4].

shaft in the axial direction during operation, which increased the fatigue characteristics of the hybrid Aluminium/composite shaft In the present work an attempt is made to evaluate the suitability of composite material such as E-Glass/Epoxy and HM-Carbon/Epoxy for the purpose of power transmission applications. Drive shaft is optimally designed and analyzed using ANSYS software for EGlass/Epoxy and HM-Carbon/Epoxy composites with the objective of minimization of weight of the shaft which is subjected to the constraints such as torque transmission, torsional buckling strength capabilities.

II. DESIGN CONSTRAINTS

In eccentric mechanical press, fluctuating type load is occur. So with the day by day use of the press higher stress is generated in the crank shaft and break the crank shaft. Here We take a press 160 ton capacity, stroke per minute 30~60 s.p.m. and drive shaft length 480 mm and major diameter 225mm and minor diameter 185 mm and main motor speed 1400 r.p.m. The primary load carried by the drive shaft is torsion. The shaft must be designed to have enough torsional strength to carry the torque without failure.

The combined shock and fatigue factor for torsion Km : 1.5
The combined shock and fatigue factor for bending Kb : 2.0

Two different types of material are selected for the design:
1. E-Glass/Epoxy and 2. HM-Carbon / Epoxy materials.. For this design, the thickness of each lamina is considered as 0.125 mm.[5]

III.MATHEMATICAL FORMULAE

Torque developed by the motor:

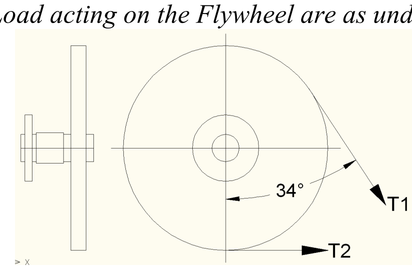
$$P : \frac{2 \times \pi \times N \times T}{60} \text{ kW} \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

$$: 66314.56\ N{\cdot}mm$$

T_f : T $\times$ velocity ratio
:3183098.86 Nmm

T_f is the torque transferred by the motor to the flywheel.

Load acting on the Flywheel are as under:



Diameter of the flywheel Df : 1350mm

$$T_f : (T1 - T2)\ R_f \quad \ldots\ldots (2)$$

$$\frac{T1}{T2} : e^{\mu\theta} \quad \ldots\ldots (3)$$

T1 : 6535.94 N
T2 : 1820.24 N

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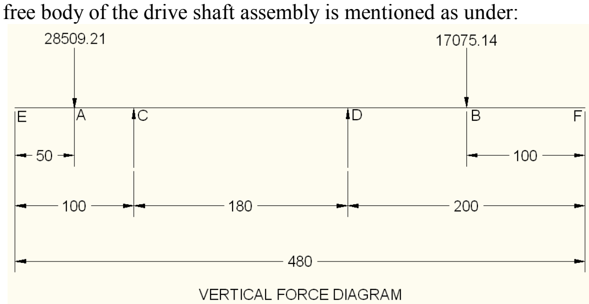
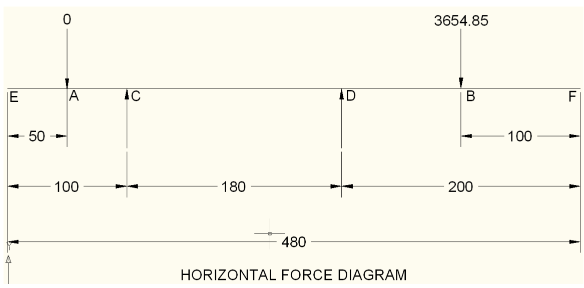
Load acting on the Gear are as under:

Diameter of the Gear Dg : 225mm
Torque transferred by the gear:

$$T_g : F \times R_g$$

The free body of the drive shaft assembly is mentioned as under:





Equivalent twisting moment Te :

$$T_e : \sqrt{(Km \times M)^2 + (Kt \times T)^2} \quad \text{…….. (4)}$$

$$T_e : 34490189.86 \text{ Nmm}$$

Equivalent bending moment Me :

$$M_e : \frac{1}{2}[(Km \times M) + Te] \quad \text{…….. (5)}$$

$$M_e : 34324146 \text{Nmm}$$

Design with EN 8 Material:

Hardness: 280 BHN
Ultimate tensile stress

σut : $3.41 \times$ BHN [6]
: 954.80 N/mm^2

Yield stress

σ_y : 555 N/mm^2 [7]

Endurance limit

σ_e : $0.80 \times \sigma_y$
: 444 N/mm^2

Design stress

σ : $0.30 \times \sigma_y$
: 166.50 N/mm^2

Factor of safety

FOS : σ_e / σ
: 3

Diameter of the shaft is calculated on the base of basically two types which are as under :

1. According to the maximum shear stress theory
2. According to the principle stress theory

Diameter of the shaft based on the maximum shear stress theory

Maximum shear stress

$$\tau_{max} : \frac{\sigma_y}{2 \times FOS}$$

$$: 92.5\ N/mm^2$$

$$T_e : \frac{\pi}{16} \times d^3 \times \tau_{max} \quad \text{…….. (6)}$$

d : 125 mm

Diameter of the shaft based on the principle stress theory.

Maximum stress

$$\sigma_{max} : \frac{\sigma_y}{FOS}$$

$$: 185\ N/mm^2$$

$$M_e : \frac{\pi}{32} \times d^3 \times \sigma_{max} \quad \text{…….. (7)}$$

d : 125.0 mm

By comparing the both results, according to the maximum shear stress theory is higher than the principle stress theory. So here the diameter of the shaft is 125.00mm.

IV.ANSYS SOFTWARE RESULT DATA

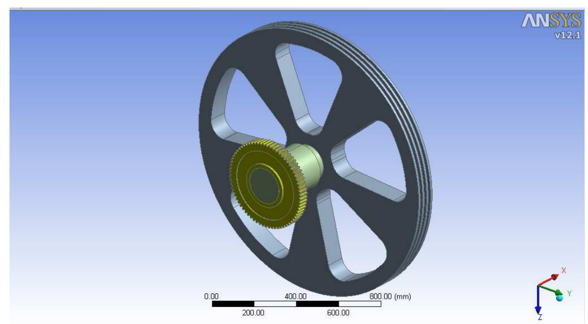


FIG 2: MODEL OF DRIVE SHAFT ASSEMBLY

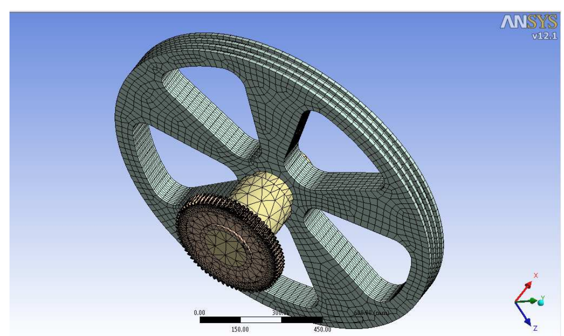


FIG 3: MESHING MODEL

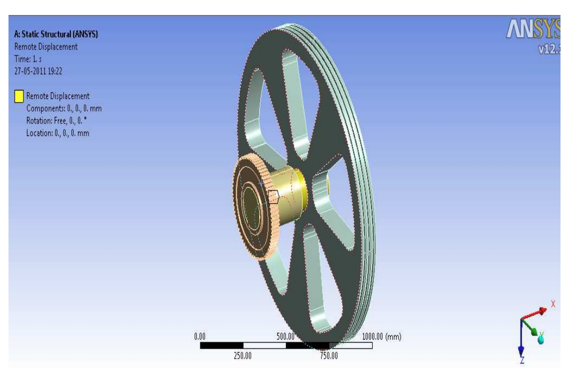


FIG 4: CONSTRAIN OF SHAFT

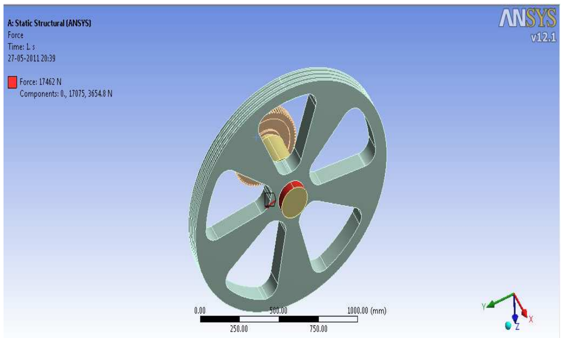


FIG 5: APPLY FORCE ON FLYWHEEL

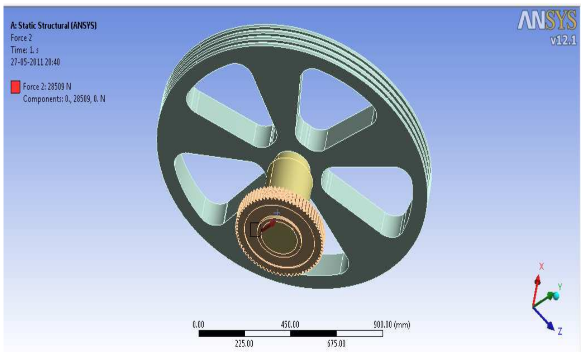


FIG 6: APPLY FORCE ON GEAR

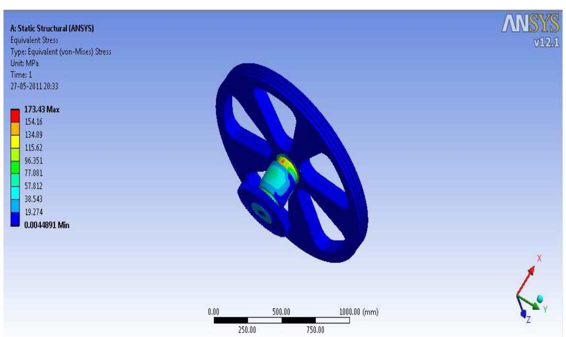


FIG 7: EQUIVALENT (VON-MISES) STRESS

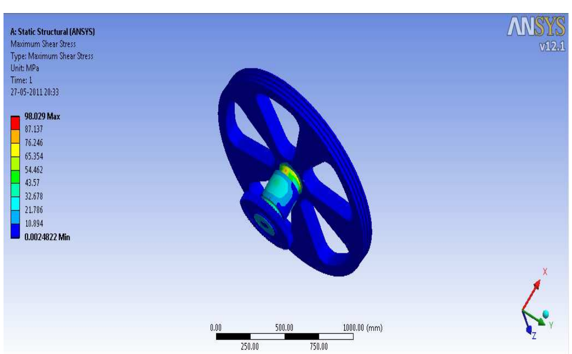


FIG 8: MAXIMUM SHEAR STRESS

V.SHAFT DESIGN USING COMPOSITE MATERIAL

A. Design 1 (E-Glass / Epoxy Material)

Properties of the E-Glass/Epoxy Material[8]

E11 : 50 GPa (Modulus of elasticity in xx direction)
E22 : 12 GPa Modulus of elasticity in yy direction)
E22 : 12 GPa (Modulus of elasticity in yy direction)
G12 : 5.6 GPa (Modulus of rigidity in xy direction)
v12 : 0.30 (Poisions ratio)
σ1 : 800 MPa (working stress in x direction)
σ2 : 40 MPa (working stress in y direction)
τ12 : 72 Mpa (working shear stress in xy direction)
ρ : 2000 Kg/m3 (density of the material)

Normal stress for this material :

$$\sigma_{max} : \frac{\sigma_1+\sigma_2}{2} + \sqrt{\left(\frac{\sigma_1-\sigma_2}{2}\right)^2 - \tau^2}$$

σ_{max} : 793.12 N/mm2

From the above calculation of the steel shaft the design stress for the tensile is twice the shear stress.

σ : 2 × τ12
σ : 144 N/mm2

Diameter of the shaft based on the maximum shear stress theory from eq. 6
d : 135 mm

Diameter of the shaft based on the principle stress theory from eq. 7
d : 135.00 mm

By comparing the both results, according to the maximum shear stress theory is higher than the principle stress theory. So here the diameter of the shaft is 135.00mm.

B. Design 2 (HM-Carbon / Epoxy Material)

Properties of the HM-Carbon / Epoxy Material[8]

E11 : 190 GPa (Modulus of elasticity in xx direction)
E22 : 7.7 GPa (Modulus of elasticity in yy direction)
G12 : 4.2 GPa (Modulus of rigidity in xy direction)
v12 : 0.30 (Poisions ratio)
σ1 : 870 MPa (working stress in x direction)
σ2 : 54 MPa (working stress in y direction)
τ12 : 30 Mpa (working shear stress in xy direction)
ρ : 1600 Kg/m3 (density of the material)

Normal stress for this material :

$$\sigma_{max} : \frac{\sigma_1+\sigma_2}{2} + \sqrt{\left(\frac{\sigma_1-\sigma_2}{2}\right)^2 - \tau^2}$$

σ_{max} : 871.10 N/mm2

From the above calculation of the steel shaft the design stress for the tensile is twice the shear stress.

σ : 2 × τ12
σ : 60 N/mm2

Diameter of the shaft based on the maximum shear stress theory from eq. 6
d : 180.00 mm

Diameter of the shaft based on the principle stress theory from eq. 7
d : 180.00 mm

III. Conclusion

By comparing the both results, according to the maximum shear stress theory is higher than the principle stress theory. So here the diameter of the shaft is 180.00mm.

Sr.No.	Material	Diameter	Weight	Saving
1	EN-8	125.00	45.95 Kg.	0.00
2	E-Glass / Epoxy	135.00	13.74 Kg.	70.09 %
3	HM Carbon / Epoxy	180.00	19.54 Kg.	57.46 %

From the above table, the E-Glass / Epoxy is most beneficial then the available three material.

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