Design and analysis of gear testing kit and static structural analysis of spur gear using ANSYS software

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Abstract. Over the years, transmission system plays a vital role in automobile industry in order to improve the performance of vehicle. Since 323 BC to till date gears plays a vital role in building the integral parts of washing machines, gear pumps, gear motors and wrist watches and so on. The lot of gear applications are extended in all sector of transmission mediums, without gear elements most of the transmission become invaluable in modern engineering industry. This paper aims to design and analysis of gear testing kit, for testing the load generation developed on the gear tooth through brake drum. The main objective of this work is to design the components in a gear testing kit and the components involved in the gear testing domain, are modelled using CATIA software. The main components in our paper are 1/2hp motor, coupling; shaft and bearing, mating gears are made of same dimension and of different materials. Further the work is to analyse the mating gears which are made of same material and analysis are carried through ANSYS software. Static Structural Analysis is carried through ANSYS analytical Software. Two types of plots are performed and their corresponding von-Misses stress and Total Deformation at three different torque levels are evaluated.

Keywords: Transmission system, gear testing kit, CATIA, ANSYS, von-Misses.

1. Introduction

Though they are so many tests charpy and Izod are found to be to test the materials for regular material testing use. They belong to later 1800’s and early 1900’s they are valued as a standardised testing method to extract the energy absorption of the materials. However, it is a primitive one, there is no other option except to use it, considering its reliability and validity till now. Another one is Izod impact strength test, which is also a common one now. Both are used in determining the impact resistance of materials and they are both ASTM standard method the consumption, of energy due to bending of the test piece’s or fracturing is derived from the mass of the pendulum, it’s initial swinging angle and the post impact’s rising angle record [1-2]. This exhibits empirically related energy amount to the material’s ductility or brittleness. Generally steel material prefers for heat treatment. There are several standard methods such as ASTM, ISO and SAE are coming now for material testing. Many researchers are doing investigation under a Finite
Element Analysis and experiment for the gear wheels impact which are subjected to force. An analysis is carried out through abaqus software for gear wheel for impact strength. For the rigid body model SIMPACK is used. They found out the similar results when the simulated contact on potential contact points held. In other words, the elastic multibody analysis is very accurate. Then experiments were done to validate the data got through simulation.

The cuboid and spherical impact body are used to validate the strength of the material often. They yield different results on contact conditions while doing the experiments. The results validate both finite element and elastic multi body model. Moreover, to find out the effects on a set of elements intralaminar failure model normally virtual test rig is used. Moreover in order to evaluate, the varied impact energy are compared with the results with experimental values. However in the study, normally two laminates were considered, the effects on the virtual test rig was not analysed. By using an instrument drop bar testing apparatus to do this study. In another way, researchers are often studied the impact behaviour of structural steels. From their investigation, they found out a load displacement curve, instead of charpy impact test, a drop bar was used in order to decrease the oscillations in the load time graph, in which mostly S45C and SM490B specimens are used. The results are shown through data acquisition system. It was known that, the presence of brittleness in lower testing temperature, at the same time, it was noted that, failure of notice at a higher temperature the blunting formation of the specimen seen.

In evaluating the impact fracture behaviour and fracture mechanism of metallic materials, and intended to ascertain its effectiveness and reliability, many experiments have been conducted for simulation of wheel impact test to determine the impact failure of wheel. The same was compared with the finite element results with actual results and they found out a non linear dynamic finite element with a reasonable mesh size and time step can reliably estimate the dynamic response. Similarly related to the above, the experiments have been conducted on test rig to measure transmission error of automobile gearbox. The performance of the gearbox is mainly affected by two factors (i.e) noise and vibration. Transmission error is considered to be a significant excited source. It is noted that while there are so many researchers are keen on the measurement and analysis of single gear drive. Different types of gearboxes are tested by adding different necessary modules. A rear wheel gear box is constructed to test the front engine test rig. The performance of a key component is verified by using static and modal analysis.

The gearbox is a critical component in automobile which is largely used to transfer power from engine, which has self excited vibration. This vibration totally affects the performance of the vehicle. There is no different opinion about this. The gearbox has gears, shafts, bearings, synchronizers and housing. Though it has many parts, gear plays a vital role in power transmission. For noise and vibration in gear system, gear transmission error is a main source. The current study is focussed on the measurement of transmission system for single gear drive. Besides to be noted that transmission error measurement in a complete automobile gearbox under actual operating conditions very rarely. A kind of electrically closed test rig is formulated to get the accurate measurements of transmission error. It is found that the use of energy recycling technique has made the maintenance more economic. Moreover the system works flexibility. Subsequently the test rig overall structure and operating principles are described.

2. Methodology
The methodology explains the step by step procedure for gear test rig, as shown in the figure 1. The modeling of gear test rig is modeled using CATIA software. The analysis of gears is carried out through ANSYS software for determining tooth deformation and bending strength by both numerically and theoretically. The flowchart describes step by step procedure as follows.
2.1. **Schematic layout of Gear test rig.**

The figure 2 illustrates the schematic layout of the gear test rig. Initially the motor supplies the power supply to the entire rotating medium of the test rig. From the motor as shown in the figure 2, the coupling gets rotating power, again from there through bearing the power is transmitted to the gears.

![Schematic layout of Gear Test Rig](image)

2.2. **Modeling and assembled view of gear test rig.**

The figure 3 shows the assembled view of gear testing kit is done by modelling software. The gear test rig has been designed for fatigue testing of gears when the load on the brake drum has applied. The main components involved in a gear testing kit are bed, single phase induction motor, drive shaft, load shaft, coupling, speed reducers, loading component, column and base plate, standard and test gear. Our main aim is to find out the torque capacity through tangential load generation by applying load on the brake drum. It consists of two shaft (i.e) driver and driven shaft. The standard gear is mounted on the driver shaft; on the other hand test gear is mounted on the driven shaft. The principles of the gear test rig are as follows. The initial weight of the test gear is measured by using electronic weighing machine. The test gear is mounted on the shaft is aligned with cast iron gear. With the help of load stand which is fixed on the brake drum, initially one kg of load is applied in the load stand that how much amount of fatigue takes place in order to withstand the gear.
3. Methodology

3.1. Selection of motor
The motor selected for gear testing are single phase induction motor. It acts as a prime mover to run the whole unit. The specifications of motors are as follows.

1. Motor type – single phase induction motor.
2. Voltage - 230v.
3. Frequency - 50Hz.
4. Speed - 1440RPM.

3.2. Design of lower shaft
(a) Material selection
Since there are no special requirements, the conventional material steel is chosen as shaft material.

\[
\text{Power (P)} = 2 \times 3.14 \times 1440 \times \frac{M_t}{60} = 9047.77868M_t
\]
\[
22380 = 9047.768M_t
\]
\[
M_t = 2.47N-M
\]

Forces on the gear
Tangential force on the gear,
\[
F_t = \frac{F}{\cos\theta} = \frac{123.5}{\cos20} = 131.425N
\]
Normal load acting on gear tooth,
\[
W_g = F_t = 123.5\cos20
\]
\[
W_{gv} = W_g \times \cos20 = 131.425 \times \cos20 = 123.499N
\]
Horizontal component, \[ W_{gh} = W_g \times \sin 20 \]
\[ = 44.94 \text{ N} \]

**Figure 4.** Shear force and bending moment layout

(b) **Design of gear**
- Type of gear = spur gear
- No. Of. Teeth on pinion = 20 teeth
Maximum safe stress = 220N/mm².
Maximum shear stress = 40N/mm².

**Selection of material**
Assume that pinion and gear are made of cast iron.

**Calculation of Tangential load (Fₜ):**
\[
\text{Tangential load (Fₜ)} = \frac{p}{v} \times k₀
\]

\[
\text{Pitch line velocity (v)} = \pi d₁n₁/60 = 1.50796 \text{ m/s}
\]

Assume \( k₀ = 1.25 \) for medium shock conditions

\[
\text{Tangential load (Fₜ)} = 309.1925 \text{ N/m}.
\]

**Calculation of initial dynamic load (Fₙ):**
\[
\text{Initial dynam load (Fₙ)} = \frac{Fₜ}{C_v}
\]

\[
C_v = \frac{6}{6+V}
\]

\[
\text{Initial dynamic load (Fₙ)} = 928.506 \text{ N/m}
\]

**Calculation of beam strength (Fₛ):**
\[
\text{Beam strength (Fₛ)} = \pi \times m \times 10 \times 220 \times 0.1084
\]

\[
= 749.207 \text{ m}^².
\]

**Calculation of module**
\[
F_S \geq F_d
\]

\[
749.207 \text{ m}^² > 928.506 \text{ m/m}
\]

\[
749.207 \times \text{ m}^³ = 928.506
\]

\[
m = 2 \text{ mm}
\]

**Calculation of b, d and v**

Face width (b) = 10*2=20 mm
Pitch circle diameter = 2*20=40 mm
Pitch line velocity = 3.015 m/s

**Recalculation of Beam Strength:**

Beam Strength (Fₛ) = 2996.828N

**Recalculation of accurate dynamic load**

Dynamic load (Fₙ) = 2996.825N.

**Check for beam strength**
Dynamic load of gear (Fₙ) is lesser than beam strength (Fₛ). Hence design is safe.

(c) **Design of KEY**

Diameter of the shaft (d) = 20 mm
Torque to be transmitted = 7.75*10³N-mm
Mild steel(t) = 50 N/mm²
Crushing stress = 120 N/mm²

Based on the shaft diameter (d) = 20 mm
Width of the key, b = d/4
= 5 mm
Height of the key (h) = d/6
= 3.33 mm
Length of the key (l) = 1.5d = 30 mm

Check for shear strength of the key
\[
M_k = tlb\times d/2
\]

\[
= 7.75 \times 10³ \times 2 / 30 \times 5 \times 20
\]
Check for crushing $M_t = 5.016 \text{N/mm}^2$.

(d) **Design of KEY**

Based on the shaft diameter ($d$) = 20 mm

Speed = 1440 rpm,

A single row deep groove ball bearing is selected.

Dynamic load rating, $c$ = 1000 kgf = 10000 N

Equivalent load ($p$) = $(X_F \times Y_F)\cdot S$

Loading ratio ($C/P$) = 9.11

$C = 70768.95$ N.

(e) **Dimensions of BEARING:**

Diameter of bearing ($d$) = 130 mm

Outer diameter ($d$) = 200 mm

Width ($B$) = 33 mm

Outer corner radius ($r_2$) = 3 mm

Inner corner radius ($r_1$) = 2 mm

Static capacity ($C_o$) = 8300 kgf

Dynamic capacity ($C_o$) = 8300 kgf

Permissible speed ($N$) = 3000 rpm

(f) **Design of BRAKE DRUM:**

$\varphi = \frac{4\pi \sin \theta / 2 \varphi + \sin 2 \theta}{2 \theta = 90, \ p=10 \text{N}, \ \varphi=0.35}$

$\varphi' = 4 \cdot 0.35 \cdot \sin 45/\pi/2 + \sin 90$

$= 0.81$

$R_0 = \text{normal force pressing the block to brake drum}$

$10(60+80) + F_t \cdot 50 = F_t / 0.381 \cdot 60$

$1400 + 50F_t = 157.48 F_t$

$F_t = 13.025 \text{N}$

Diameter of Brake drum ($D$) = 50 mm

Radius of brake drum ($R$) = 25 mm

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Figure 5. Schematic layout of single band brake

Diameter of brake drum ($D$) = 50 mm

Radius of brake drum ($R$) = 25 mm

$2 \theta = 90, \ p=10 \text{N}, \ \varphi=0.35$
2\theta=90, \quad p=10N, \quad \varphi=0.35

\varphi' = 4\pi \sin \theta / 2 \theta + \sin 2 \theta

Apply moment on both sides

\begin{align*}
20(60+80) + F_t \times 50 &= F_t / 0.381 \times 60 \\
2800 + 50F_t &= 107.48F_t \\
F_t &= 26.005 N.
\end{align*}

From those comparison of two loads; tangential load of brake drum is lesser than tangential load of spur gear. Hence design is safe. Hence 20 N of load applied in brake drum produced 123.71N of force produced in the spur gear as shown in the figure 4. Hence design is safe.

4. Finite Element Analysis

4.1. Modeling of gears

Parametric modelling of spur gear allows the design engineer to let characteristics parameters of the product drive the design of that product. The parameters to model the spur gear are pressure angle, module, numbers of teeth of gears are taken as input parameters, model as shown in figure 5.

4.2. Meshing of gears

The spur gear is modelled in CATIA software. The next step after modelling is to save the part in IGES format and then exported to ANSYS software. Meshing is the second step in ansys software. The purpose of meshing is done to get the accurate results of the part. The sizing of the mesh is fine mesh. The number of elements and nodes in the mesh is 42777 and 209205. The meshing of gears is shown in figure 6.

Figure 6. Modelling of spur gear using CATIA Software

Figure 7. Meshing of gears by ANSYS Software
4.3. Boundary conditions
The next step after meshing is to apply boundary conditions. Frictionless support is applied at the left end of the gear to allow its tangential rotation but restrict from radial translation. Boundary condition plays an important role in finite element calculation. Here we have remote displacements for bearing supports are fixed as shown in the figure 7. Here are two boundary conditions at torque (a) \( t = 1000 \) N-m (b) 2000 N-m.

![Figure 8. Boundary conditions of gears at torque (a) t = 1000 N-m (b) T = 2000 N-m.](image)

5. Results and discussion

![Figure 9. Von misses stress at torque (a) t = 1000 N-m (b) t = 2000 N-m](image)

Figure 8 shows that von misses stress of cast iron gear with pinion gear at two different torques \( t = 1000 \) N-m and torque \( t = 2000 \) N-m. Von Misses stress shows that how much stress during engagement of the gear tooth. A load acting at the tip of the gear tooth. Maximum von misses stress developed on the gear tooth is 681.06 MPa. Minimum stress developed on the gear tooth is 38.17 MPa.

5.1. Total deformation
Figure 9 shows that total deformation at different torque levels at torque \( T = 1000 \) N-m and \( T = 2000 \) N-m. Total deformation shows that displacements along X-direction in the tooth flank. The minimum displacements occurred in right side and maximum displacements occurred in the left side. The minimum displacements occurred in the gear tooth flank is 0.00038m and the maximum displacement occurred on the gear tooth flank is at torque \( T = 2000 \) N-m is 0.87786 m.

![Figure 10. Total deformation at torque T=1000 N-m and Torque T = 2000 N-m](image)
6. Conclusion
In this paper, the design procedures of various parts of gear testing kit are designed using standard design procedures. Required dimensions of each part have been derived and based on that a 3D model has been created using CATIA software. Further the critical component of the kit is identified and stress analysis has been created using ANSYS software. The results are validated by theoretically and numerically. By theoretical, 20 N of load applied in the brake drum via load stand produced 123.71 N. Hence a design rig has the capacity to withstand 20 N load. From numerically, two stresses are developed von miss stress and total deformation, maximum stress developed on the gear tooth is 681.06 MPa and minimum stress developed on the gear tooth is 38.17 MPa. This is due to the reason by increasing the load more stress will be developed. On the other hand, total deformation takes place on the gear flank on the mating gears, the minimum displacements occurred in the gear tooth flank is 0.00038 m and the maximum displacements occurred in the gear tooth flank is 0.87786 m. This is because of due to rotation at lower torque at low deformation. Higher the torque high deformation is developed.

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