International Journal of Advance Research in Science and Engineering
Volume No.07, Special Issue No.03, April 2018
www.ijarse.com
IJARSE
ISSN: 2319-8354

Design Methodology of Drive Shaft

Mrs. Vishwal Kawale[1],Mr. Govind Patil[2],Mr.Shyam Pune[3]

[1]Mechanical Design Engineering, TPCT's College of Engineering, Osmanabad, 413501, (India)

[2]Mechanical Design Engineering, TPCT's College of Engineering, Osmanabad, 413501, (India)[3] BE (Mechanical Engineering) GSMCollege of Engineering, Balewadi, Pune, (India)

ABSTRACT

Drive shaft is the most important component to any power transmission application. A drive shaft, also known as a propeller shaft .It is a mechanical part that transmits the torque generated by a vehicle's engine into usable motive force to propel the vehicle. The overall objective of this paper is to analyzedesign methodology of drive shaft for power transmission applications. Design parameters were considered with the objective of minimizing the weight of drive shaft. The design methodology also showed significant potential improvement in the performance of drive shaft. In this paper we are try to explain and analyze the of propeller shaft design with input parameters like type of vehicle,engine,gear box Specifications, tyresize, application of vehicle, desired life expectancy, vehicle aggregate layout etc.

Keywords- *Propeller Shaft, Splined Shaft, Universal Joint, Flange and Needle Roller Bearing*

I. INTRODUCTION

The power from Transmission shaft should be transmitted to the Rear axle of the vehicle. The axisof the Transmission and the connecting member of Rear axle are at an angle, which changes with the variation in load or the road condition. To facilitate the power transmission at a variable angle a Propeller shaft is used. With respect to the geometrical construction the Propeller shafts are categorized into single piece two-piece and three-piece propeller shafts. In case of two or multi stage propeller shaft length of the rear propeller shaft is subjected to variation while the remaining propeller shafts are rigid members. The variation in the length of rear propeller shaft is allowed using a splined shaft. It is assumed that the inclination of cross member bracket (in case of multistage propeller shaft) is also decided based on the requirement criteria such as beta equivalent angle. The maximum and minimum length of the propeller shaft required by finding the slip required for the particular vehicle. The main objective of the paper is to analyse the design of propeller shaft as per design methodology.

Fig1: Coupling Shaft and Drive Shaft

IJARSE
ISSN: 2319-8354

II. INPUT DATA SHEET

Type of Vehicle

- Rated gross weight
- Max axle load front, Rear
- Drive – Front/ rear
- Overload Conditions
- Off road/ On road

Engine

- Max Engine torque at speed (rpm)
- Engine power output at speed (rpm)
- Max& Idling speed

Gear box Specification

- Gear ratio forward
- Gear ratio reverse

Tyre size specification

- Tyre size
- Static Dynamic radius

Vehicle aggregate layout

- Length of propeller shafts
- Joint angle details

III. DESIGN METHODOLOGY

Functional Requirement:

Four basic functions that the driveline is excepted to satisfy are:

1. Torque transmission:
   The driveline in an application must be capable of transmitting the maximum torque developed by the drive train. The torque loads can be applied steadily under static or rotating conditions.

2. Rotation as per the speed range of selected application: Depending on the application the driveline may be required to rotate through a relatively large speed range, which can vary zero to engine maximum speed.
3. Capable of operating at fixed or varying angles:
   The universal joints of the driveline must be capable of operating through the required range of angles at varying torque loads and operating speeds without causing objectionable disturbances.
4. Shaft length changes:
   The driveshaft length changes during operation of a typical automotive vehicle are due to the extreme of suspension travel ranging from full jounce to full rebound positions, as well as rear axle carrier nose wind up and wind down.

IV. DESIGN STEP

5.1 Universal joint design:

1. Joint phasing arrangement
2. Torsional excitation limit
3. Secondary couple excitation limit

5.2 Tube selection:

1. Maximum applied torque& speed
2. Driveshaft length

- Design of cross Trunnion and bearing configuration of joint
- Design of yoke, spline and axial motion devices
- Design of shaft support bearing
- Balancing
- Control of drivetrain disturbances
-

V. DETAIL DESIGN

Please follow the below reference diagram for propeller shaft detail design.

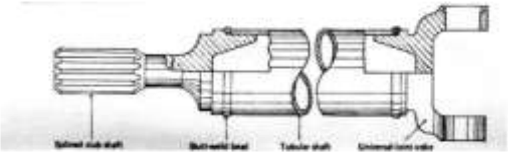
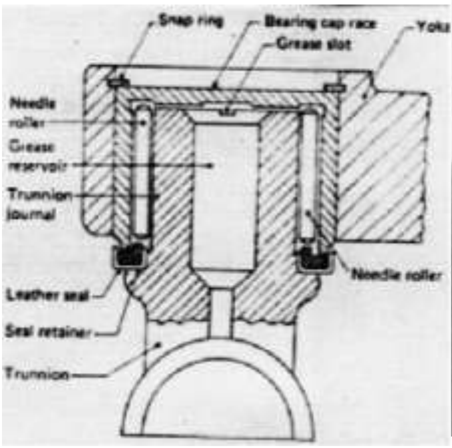


Fig-:2 Configuration of Propeller Shaft



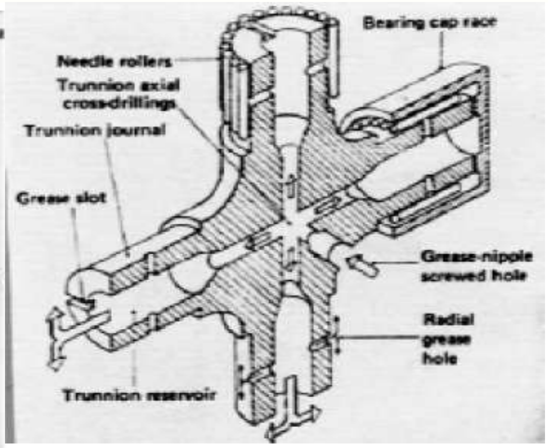


Fig-3 Universal Joint TrunnionFig-4Universal joint needle bearing lubrication

- **Cross & Needle bearing Design:**

1) Universal Joint cross trunnion:

Torque couple force on the cross

F = T/2R

T = Maximum torque-mm

R = Torque radius, mm

2) Nominal bending stress

   6b = FL/Z

   Z = πD^3/32

L = Bending moment arm

Z = Area section modulus

D = trunnion diameter = 18.7

*In case of stress raiser taken into consideration for trunnion, the maximum effective stress 6b affected by geometric or theoretical stress concentration factor Kt.

6b = KtFL/Z

= 32KtFL/πD^3

3) Needle Roller Bearing Design:

International Journal of Advance Research in Science and Engineering
Volume No.07, Special Issue No.03, April 2018
www.ijarse.com
IJARSE
ISSN: 2319-8354

3.1 The Length to diameter ratio I/d = 3.50 – 8.50

Min Permissible dia of roller = d =3.175mm

Min diameter clearance = Cc = 0.3048mm

3.2 No. of needle roller n=π (D+d)/D

D=dia of trunnion

d=Roller Diameter

3.2 Highest loaded roller face

Po = 11F /N

F = Torque Couple Force

N = No. of Needle rollers

11=Constant for needle roller bearing. With radial ply or clearance.

4) Hertz Contact stress between Roller & Bearing Cup Bore:

$$\sigma_m = 65.4\sqrt{\left(\frac{P_0}{L_1}\right)\left(\frac{1}{d}-\frac{1}{B}\right)} \quad \text{Kg/mm}^2$$

L1 = Effective needle roller length on bearing cup bore (mm)

D= Needle roller dia (mm)

Po= Force on highest roller

B=Bearing cup bore dia (mm)

4.1 Contact Stress between Roller & Trunnion (approx. Equation)

$$\sigma_m = 58.86\sqrt{\frac{T}{BF}} = 58.86\sqrt{\frac{T}{DL_1R}} \quad \text{Kg/mm}^2$$

T = Torque (N.mm)

L1= Effective Needle Roller Length (mm)

D= Dia of Trunnion (mm)

R= Torque Radius (mm)

BF= Bearing Factor (mm^3)

*For cardan joints, the contact stress between needle rollers & trunnion can be 6m=4, 00,000 psi= 280 kg/mm^2 at max applied torque.

- **Spline**

Design of Spline

1) Shear stress in the spline teeth: At the pitch line

$$\tau = \frac{2T}{DtLN}$$

Assuming 100% contact

2) Shear stress at the core diameter of spline: At the minor diameter of the internal spline with 100% teeth in contact,

$$\tau = \frac{2T}{D_i(t_x)LN}$$

Where tx = tooth thickness

Di = internal minor diameter

3) Compressive / contact stress in the spline teeth:

$$\sigma_c = \frac{4T}{D(D_o - D_i)LN}$$

The Max. Allowable compressive stress is 5000psi/3.51Kgmm2 from table1

Ka = Spline application factor

Lw = Life factor due to wear

The compressive stress calculated considering the life and application factors must be less than the maximum allowable compressive stress. This should be satisfied to meet the wear criteria in spline.

4) Hoop or Bursting stress in the hub

$$\sigma_c = \frac{2T \qquad \tan\phi}{\pi D(D_{re} - D_{ri})L}$$

Where Φ = pressure angle of the spline teeth

D re =external major diameter

D ri =internal minor diameter

Table -1 Allowable compressive stress for spline:

Material	MAX. ALLOWABLE COMPRESSIVE STRESS	
	PSI	Kg/mm2
Steel	1500	1.05
Steel	2000	1.4
Steel	3000	2.1
S/F hardened steel	4000	2.81
Case hardened steel	5000	3.51

- **Flange**

The flange is standardized as per SAE and DIN standards; these are having either plane connecting faces or serrated teeth. The serrated flanges have advantages of reduced loading on connecting bolts and also reduced risk of bolt loosening.

The design check on the selected bolt can be done as follows:

P= pretension by virtue of tightening torque, Kg (Ref-TS-12805)

R= pitch circle radius of the flange

N= No of bolts

U=coefficient of friction between the tightened flanged.

Tr = the resisting torque after tighten the flange joint

$$Tr = n\mu \Pr$$

This torque is required to be more than the starting torque to avoid loosening. However this may not meet the higher torque during jerky operation caused due to improper starting of the vehicle. In such case, serrated flange joints are reliable against loosening.

- **Balancing**

Balancing can be defined as the processes of altering the distribution of mass in a rotating body to eliminate or minimize vibration at the support bearing. Balancing is to compensate for the weight of eccentrically rotating masses in the prop shaft so as to contain smooth running and a reduction in joint loads and bearing forces connected unit.

Displacement of center of gravity

$$\varepsilon = \frac{ur}{G}$$ G-mm/kg

Where, u= uncompensated individual mass at radius r

G= weight of the part to be balanced.

Balancing Parameters.

1. Run out of driveshaft/over a wide speed range especially at high speed, run out is the most important single factor in dynamic balancing
2. Operating clearances of universal joints, splines and must be maintained as close as possible.
3. Uniform wall thickness to be maintained around the diameter of the tube.
4. Any dynamic balancing operation on a driveshaft should be performed with the shaft's mass center supported very close to its rotational axis.

VI. CONCLUSION

1) Detail design methodology for propeller shaft is analysing in this paper.

2) Design of propeller shaft is carried with input data sheet. As the input is changes, output of the design will also get changed.

3) Design will also get optimise though analysis of calculation study.

REFERENCES

Books:

1] E. R. Wagner. "Driveline and Driveshaft Arrangements and Constructions, Universal Joint and Driveshaft Design Manual",Chapter I, SAEAE-7. 1947, P.440.

2] P. J. Mazziotti, "Universal Joint and PropellerShaft", Dana Corporation Bulletin J-1371, October 15, 1954.

3] B. R. Reimer, "Design and ApplicationConsiderations for Agricultural PTO Drivelines", SAE Paper 650680, 1965.

4] Spring Design Manual. AE-21.

5] Thomas. D. Gillespie. 1994. "Fundamentals of Vehicle Dynamics", P. 24

Journal:

6] "Wheel lift-off &ride comfort of three wheeledVehicle over bump", IE (I) Journal-MC.PP 78-87.

7] L. J. Francesco," Better Needle Bearings forUniversal Joints" SAE Paper 660159, 1966.

8] ''Failure Analysis and Design modification of propeller shaft of BUS'',Volume 4 Issue X, Oct. 2016ISSN: 2321-9653