STRUCTURAL AND FATIGUE ANALYSIS OF CARDAN SHAFT

[1]Burak AL, [2]Murat ÖZSOY, [2]Neslihan ÖZSOY

[1]*Turkish Aerospace Headquarter, Ankara, Türkiye, burakal26@gmail.com*

[2]*Sakarya University, Department of Mechanical Engineering, Sakarya, Türkiye, ozsoy@sakarya.edu.tr, nerken@sakarya.edu.tr*

https://doi.org/10.37904/metal.2024.4934

Abstract

Cardan shafts are a system widely used for power transmission in the industrial area, especially in the automotive sector. This study carried out computer-aided design and analysis of the involute profiled spline shaft (cardan shaft) and hub assembly. First, gear profile dimensions of the gears of the cardan shaft and the hub were generated parametrically under the EN-5480 standard using the PTC Creo Parametric software. The solid model was then imported into Ansys software to create the mathematical model. Equivalent stress, total deformation, and damage-life analysis were performed by simulating working conditions. By interpreting the results obtained, it was determined whether the system was compatible with the terms of use, and suggestions were made for improvement.

Keywords: Spline shaft, structural analysis, fatigue analysis

1. INTRODUCTION

The purpose of the cardan shafts is to provide power and moment transmission through the shafts of the movement points, which are mostly axially offset between them. For this reason, many parameters such as the life of the cardan shaft, the load capacity, shaft length, critical speed, vibration, and fatigue are the subject of research. Many researchers have carried out their studies using the finite element method on this subject.

Schäfer and Garkze investigated the behavior of a cardan shaft with an involute tooth profile operating under overload. Their analysis with the Ansys software using the finite element method stated that the tooth root's geometry should be changed to eliminate the primary stress source caused by the overloading on the cardan shafts. It has been stated that increasing the tooth surface pressure angle and the number of teeth is also essential parameters in increasing the radial load carrying capacity [1].

Sandip et al. carried out damage analysis after 13 months of damage to the cardan shaft, which has an average operating life of 15 years. As the first determination in their study, they determined that the stresses in the shaft cause cracking It has been determined that the shaft diameter, whose cause of damage is below the critical limit, is damaged under torsional load [2].

Chase et al. determined that different stresses occur in threads under load, which differ in profile offset and manufacturing tolerances. As a result, it has been understood that the shaft design and tolerance values should be more precise to achieve better performance in cardan shaft designs [3].

Cura et al. investigated the propagation of cracks in teeth in involute profile spline shafts. The effect of shaft length, shaft-gear thickness and speed on the crack propagation path was analyzed with finite element models. In particular, to better understand the effect of crack propagation speed, stress intensity factors in two different models were considered. It is emphasized that, compared to standard gears, tubular gears are subjected to both bending and torsional stresses [4].

There are many studies on spline shaft-hub assembly using experimental and finite element methods [5-9].

Long-term operation of the shafts and the overloads they are exposed to shorten the life of the shaft and gear and dangerous and costly problems such as gear stripping, tooth wear, cracks, and breakages in the shaft and gear. To prevent these damages and costs and, the design and structural analysis of a spline shaft produced by Demireller Kardan company under the DIN 5480 standard was carried out in this paper.

2. MATERIAL AND METHOD

First, mathematical formulas for spline shaft and hub selection are derived. Then, the tooth dimensions on the spline shaft and the hub, which were used as samples, were determined.

Gear calculations according to DIN EN-5480 have been converted to solid model in the computer environment. The modelled geometry was transferred to the Ansys workbench software and mathematical model was created.

The cardan shaft, the subject of this study, is manufactured from DIN 42CrMo4 steel. As a design parameter, the mechanical properties of DIN 42CrMo4 steel, determined according to EN-10083-3:2007-1 standards, are as follows in **Table 1.**

Table 1 Mechanical properties of DIN 42CrMo4 steel [10]

Dimension	Yield strength MPa	Tensile strength MPa (min.)	Density (kg.m^{-3})	Youngs module (GPa)	Poisson's ratio (-)
d ≤ 16 mm	900	1100-1300	7800	210	0.3
16 <d ≤ 40 mm	750	1000-1200			
40 <d ≤ 100 mm	650	900-1100			
100 <d ≤ 160 mm	550	800-950			
160 <d ≤ 250 mm	500	750-900			

2.1. Dimensional calculations and solid model

Cardan shaft and hub designed in this study have been measured under DIN 5480-1 standard. [11]. Dimensioning of the involute threaded shaft and hub assembly is shown in **Figure 1**, and the relations between dimensions are given in **Table 2**.

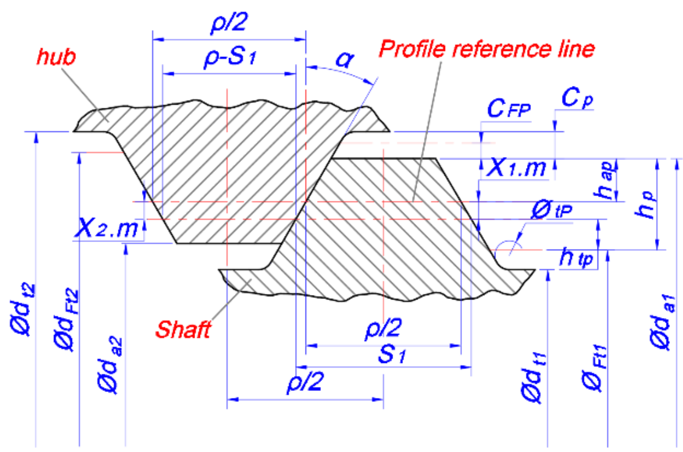

Figure 1 Spline shaft and hub geometrical dimensions [11]

In this study, six different samples will be analyzed. First, a basic spline shaft and hub design for the samples to be analyzed were created with Creo parametric software. Then, the geometric values of the samples were changed over the parametric equations, and the solid model of each sample was obtained.

Table 2 Cardan shaft and hub geometric and profile formulas [11]

Parameter	Symbol	Calculation Formula
Module	m	0.5-10
Pressure angle	α	30°
Pitch	p	$m\pi$
Number of teeth (shaft)	z_1	z_1
Number of teeth (hub)	z_2	z_2
Addendum Modification (Shaft)	$x_1.m$	$-0.05m\ to + 0.45m$
Nominal value (hub)	$x_2.m$	$+0.05m\ to - 0.45m$
Addendum of basic rack profile	h_{aP}	$+0.45m$
Dedendum of basic rack profile	h_{fP}	$+0.55m\ (broaching)$
Tooth depth of basic rack profile	h_p	$h_{aP} + h_{fP}$
Bottom clearance of basic rack profile	c_p	$h_{fP} - h_{aP}$
Root fillet radius of basic rack profile	ρ_{fP}	$0.16m{\sim}0.54m$
Reference diameter	d	mz
Base diameter	d_b	$mzcos\alpha$
Reference diameter	d_B	$mz_1 + 2x_1m + 1.1m$
Tip circle diameter of hub	d_{a2}	$mz_2 + 2x_2m + 0.9m$
Root circle diameter of hub	d_{f2}	$mz_2 + 2x_2m - 2h_{fP}$
Root form circle diameter of hub	d_{Ff2}	$m \leq -(d_{a1} + 2C_{Fmin})$
Tip circle diameter of shaft	d_{a1}	$mz_1 + 2x_1m + 0.9m$
Root circle diameter of shaft	d_{f1}	$mz_1 + 2x_1m - 2h_{fP}$
Base form circle diameter of shaft	d_{Ff1}	$\leq \lvert d_{a2} \rvert - 2C_{Fmin}$
Nominal space width of hub	e_2	$e_2 = S_1$
Nominal tooth thickness of shaft	S_1	$\frac{m\pi}{2} + 2x_1mtan\alpha$

In the technical documentation of the spline shaft produced by Demireller Kardan, the diameter over the shaft tooth diameter is 130 mm. Accordingly, for the shaft and hub profiles of the cardan shaft, the number of modules and gears was selected from DIN 5480. For the shaft and hub, the module was chosen as 3, the number of teeth was 42 and the pressure angle was 30 degrees.

2.2. Static analysis

According to the product catalogue, the shaft-hub assembly to be analyzed is based on 50 kNm static loading and 25 kNm fatigue loading.

The mechanical properties of the DIN 42CrMo4 material used in the analysis are shown in **Table 1**, and the fatigue properties are shown in **Table 3**. **Figure 2** shows spline shaft and hub solid model

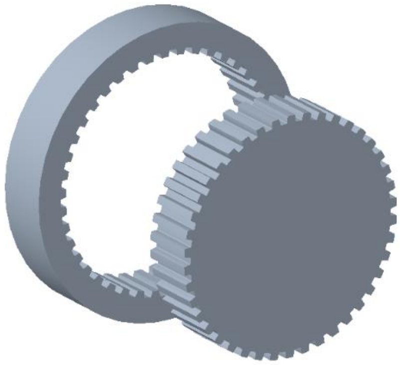

Figure 2 Spline shaft and hub solid model

Table 3 Fatigue properties of 42CrMo4 [12]

Cycle	Stress (MPa)
4,000	550
10,000	480
20,000	420
50,000	340
100,000	300
200,000	270
1,000,000	240

The tooth surfaces of the spline shaft are selected as "Frictional" surfaces from the "Connections-Contact" section. The friction coefficient is taken as 0.2 for ductile steel materials. A torque of 50 and 25 kNm was applied to the shaft in two different analyzes.

To fix the outer circumference of the hub structure as the boundary condition, its circumference was chosen as "Fixed Support". For the torque to be applied on the shaft to be equally distributed, the entire gear surface of the shaft structure was selected, and the "Remote Point" was created for moment loading. Boundary conditions and loading details can be seen in **Figure 3a** and **Figure 3b** respectively.

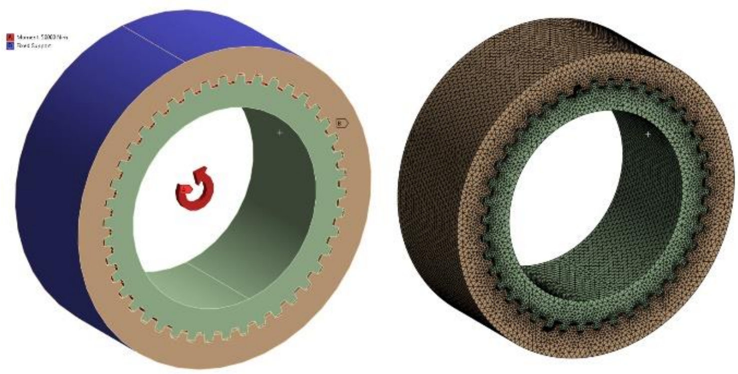

Figure 3 Cardan shaft-hub assembly mathematical model

3. RESULTS AND DISCUSSION

A total of 6 different analyzes were performed in the study, including 3 different tooth lengths and 2 different root radii. As a result of the analysis, equivalent stress values and total deformation amounts were examined. A torque loading of 50 kNm was applied for structural analysis. The stress and total displacement results of the static analysis are shown in **Table 4** and **Table 5**, respectively.

Table 4 Von-Misses stress results

Analysis No	Tooth length (mm)	Root fillet radius (mm)	Hub stress (MPa)	Shaft stress (MPa)
1	30	0.2xm	522	539
2	30	0.4xm	427	473
3	45	0.2xm	327	376
4	45	0.4xm	281	342
5	60	0.2xm	232	269
6	60	0.4xm	203	221

Table 5 Total deformation results

Analysis No	Tooth length (mm)	Root fillet radius (mm)	Hub deformation (mm)	Shaft deformation (mm)
1	30	0.2xm	0.015	0.019
2	30	0.4xm	0.014	0.018
3	45	0.2xm	0.01	0.013
4	45	0.4xm	0.009	0.013
5	60	0.2xm	0.007	0.01
6	60	0.4xm	0.007	0.009

After static structural analysis a fatigue test with 25 kNm was performed. “Stress Life” was chosen as the Analysis Type, and “Fully Reversed”, fully variable loading, was chosen as the loading type. The analysis was carried out in the mean stress theory by choosing the “Goodman” theory. **Table 6** gives the fatigue life and safety factor values for the spline shaft and hub component.

Table 6 Fatigue analysis results

Analysis No	Tooth length (mm)	Root fillet radius (mm)	Hub Life	Hub Safety Factor	Shaft Life	Shaft Safety Factor
1	30	0.2 m	156,000	0.9	93,000	0.89
2	30	0.4 m	564,000	1.14	432,000	1.01
3	45	0.2 m	1,000,000	1.45	1,000,000	1.31
4	45	0.4 m	1,000,000	1.71	1,000,000	1.67
5	60	0.2 m	1,000,000	1.98	1,000,000	1.78
6	60	0.4 m	1,000,000	2.35	1,000,000	2.31

4. CONCLUSIONS

As a result of the static structural analysis, it was observed that the maximum equivalent stress values in the shaft and hub for each analysis were very close to each other, and the stresses decreased by increasing the tooth length and root radius. It is understood that the stress values, except for analysis number one, are sufficiently below the yield limit of the material, 550 MPa. For the total deformation results, it is seen that increasing the tooth length and root radius reduces the deformation.

When fatigue analyses were examined, increasing the tooth length or root radius increased the fatigue life and safety factor for all shaft and hub samples. Analysis one and two results were below 10^6 cycles according to the fatigue life results. Other analysis results showed success by reaching 10^6 lifespans.

The next step of this study can be validation these analysis by experimental methods.

REFERENCES

[1] SCHAFER, G., GARZKE, M. Increasing Load Capacity of Splines Due to Design. In: *International Design Conference*. Dubrovnik: SCIRP, 2002, pp. 695-700.

[2] SANDIP, B., BANERJEE, A., CHAKRABARTI, I., BHAUMIK, B.K. Failure analysis of an input shaft of skip drive gearbox. *Engineering failure analysis*. 2008, vol.15, no. 4, pp. 411–419.

[3] CHASE, K., SORENSEN, D., DECAIRES, B. Variation analysis of tooth engagement and load sharing in involute splines. *Gear Technology*. 2010, Vol.5, pp.54-62.

[4] CURÀ, F., MURA, A., Adamo F. Investigation of crack propagation path in tube gears. *Procedia Structural Integrity*. 2017, vol. 7, pp. 476-483.

[5] FEROZ, A.A., CHAWLA, H.D., KUMAR, R. To study and analyze the design of drive shafts for automobiles using composite material through empirical review on literature. *Materials Today: Proceedings*. 2022, vol. 56, no. 6, pp. 3820-3822.

[6] CARDOSO, A.S.M., PARDAL, J.M., CHALES, R., MARTINS, C.H., SILVA, M.M., TAVARES, S.S.M., PEDROZA, B.C., BARBOSA, C. Fatigue resistance performance of universal cardan joint for automotive application. *Engineering Failure Analysis*. 2022, vol 135, pp. 1350-6307.

[7] SOLAZZI, L., BERTOLI, D., GHIDINI ,L. Static and dynamic study of the industrial vehicle transmission adopting composite materials. *Composite Structures.* 2023, vol. 316, 117042.

[8] KOHLI, A., MAHESH, H., PATIL, A.Y., KUMARA, P.C.A. Analysis of universal joint using virtual simulation method. *Materials Today: Proceedings.* 2022, vol. 59, no. 19, pp. 858-866.

[9] KOUEJOU, B.X.T., ALFAYO, A.A. Dynamic Analysis of a Rotating Cardan Shaft under the Influence of a Breathing Crack and Dense Fluid Force. *Shock and Vibration.* 2022, vol. 2022, 5021488, pp.1-20.

[10] DIN EN 10083-3. *Steels for quenching and tempering- Part 3: Technical delivery conditions for alloy steels.* Berlin, Germany: Deutsches Institut für Normung, 2007.

[11] DIN 5480-1. *Splined connections with involute splines based on reference diameters.* Berlin, Germany: Deutsches Institut für Normung, 2006

[12] STRZELECKI, P., SEMPRUCH, J. Verification of analytical models of the s-n Curve within limited fatigue life. *Journal of Theoretical and Applied Mechanics*. 2016, vol. 54, no.1, pp.63-73.