

Analysis of an Intermediate Rear Axle Shaft Failure

Subrata Kr Mandal*, Palash Kr Maji and S Karmakar

CSIR-Central Mechanical Engineering Research Institute, M. G. Avenue, Durgapur, India

Abstract

This paper describes the failure analysis of an intermediate shaft used in a prototype, which had been failed during the trial run of the prototype. The shaft was found to be bending. The investigation was carried out in order to establish whether the failure was the cause or a consequence of the accident. A study of the bend shaft shows how vulnerable such a rotating component can be to failure by fatigue, even when operating under steady conditions, if basic preventative design actions are not taken. The analysis considers the effects of both transmission torque and weight (thus bending) upon stress levels and assesses their individual affect on the breakage and upon any subsequent modifications needed to improve the design. Results indicate that the axle shaft bends due to fatigue as a result of improper design. The drive shaft arrangement is compared with the feasible alternative of using a driven wheel arrangement rotating on a stationary axle. Findings confirm the importance of recognizing in advance the salient factors leading to fatigue and the necessity in paying adequate attention to detail during design and manufacture if long service life is to be achieved.

Keywords: Axle shaft; Fatigue; Failure analysis

Introduction

Having a wheel shaft which is directly driven from a power source (in this case, the engine of a motorcycle) using a timing belt transmission is a common means for driving small, relatively low cost vehicles. The attraction of a rotating wheel shaft is that it can be made very simple because it can accommodate the driven pulley, be used to mount the bearings and also attach a pair of wheels all on a single component. Such a design has undoubted advantages compared to the alternative of using a driven wheel hub arrangement located on an axle; as occurs, for example, with a chain sprocket on a bicycle wheel. However, there is a fundamental difference in the two designs-with a shaft, as it rotates, bending stresses alternate between tension and compression each revolution, whereas with an axle, since it remains stationary, this is not the case. Even when traveling at constant speed and carrying a steady load across level terrain, the alternating stress in a shaft can lead to fatigue damage, especially when wheel diameters are small and consequently the number of rotations becomes very significant [1].

The design requirement is further complicated by additional factors, many of which are of a variable nature. These include the effect of changing torque transmitted during acceleration of the vehicle from start up to full speed and also abrupt breaking (these effects will result in torsion and thus changing shear stresses), traveling over uneven terrains without suspension will contribute shock loading (further adding to fluctuating bending stresses). Consequently, the vulnerability to fatigue damage is clearly very real and so it is essential to realize this and identify weak locations and take preventative steps at the design and manufacturing stages [2].

In the present study, an intermediate shaft of the rear wheel drive assembly used in an automobile has been examined after failure, which resulted in the vehicle suddenly stopped and failed to conclude the field trial being undertaken. Significant damage was caused to the vehicle. When the vehicle was examined it was found that the intermediate shaft of rear wheel drive assembly had bend.

The Shaft Design

The design of a shaft is generally done based on the application and

uses of the shaft [3]. Here in this case the design was made keeping the fact that it will be used in a rear axle transmission system. The wheel shaft has been made from a stainless steel bar, 40C8 IS:1570-88 and has been turned down to a central diameter of 25 mm along 177 mm length with an overall length of 516 mm. An undercut was given for circlip fitting to arrest the bearings as shown in Figure 1. The ends of the shaft are reduced to 16.5 mm diameter along a 190 mm length (both side) with M16 size threaded in both ends of the shafts for fixing with the frame (Figure 1).

Assessing the Failure

The general appearance of the failed axle shaft is shown in Figure 2. A view of the shaft in assembled condition (before failure) is shown in Figure 3. The failed wheel shaft had bended almost at the end of the step and near the groove, which was unusable as shown in Figure 1. This bend is clearly visible in a naked eye. The remaining section failed through static loading being insufficient to support the loads. From observation of the failed shaft it is not evident whether shaft bending or

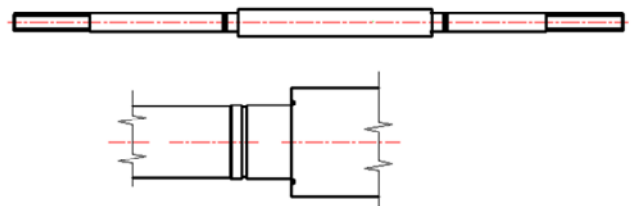


Figure 1: Design Detail of the shaft.

***Corresponding author:** Subrata Kr Mandal, Scientist, CSIR-Central Mechanical Engineering Research Institute, M. G. Avenue, Durgapur, India, E-mail: subrata.mandal72@gmail.com

Received January 03, 2015; **Accepted** February 05, 2015; **Published** February 13, 2015

Citation: Mandal SK, Maji PK, Karmakar S (2015) Analysis of an Intermediate Rear Axle Shaft Failure. Adv Automob Eng 4: 114. doi:10.4172/2167-7670.1000114

Copyright: © 2015 Mandal SK, et al. This is an open-access article distributed under the terms of the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original author and source are credited.

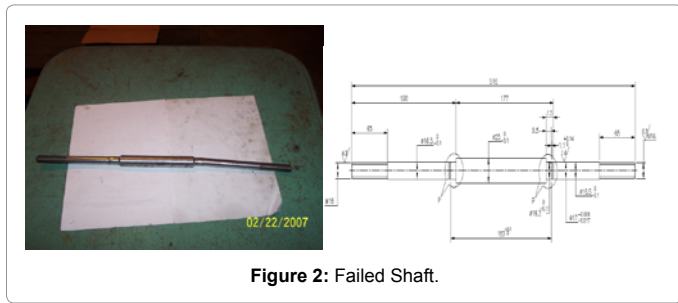


Figure 2: Failed Shaft.

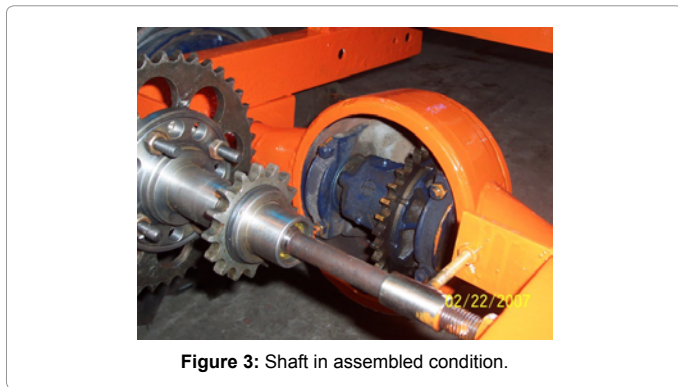


Figure 3: Shaft in assembled condition.

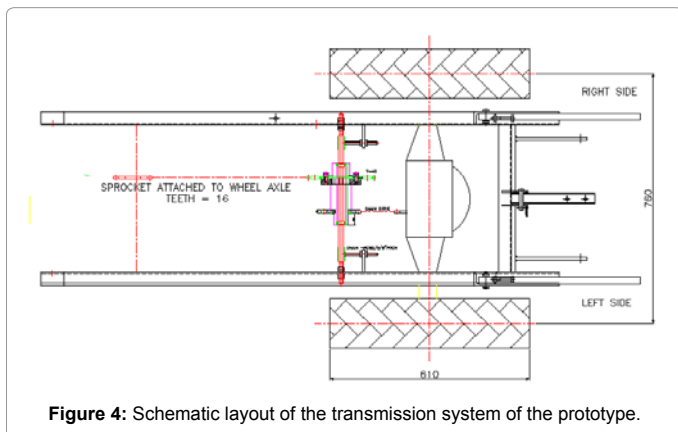


Figure 4: Schematic layout of the transmission system of the prototype.

torsion is the primary cause or whether it is a combination of the two and so analysis is necessary [4] (Figures 2 and 3).

Shaft Analysis

Because it is not immediately obvious whether bending or torsion has been the major cause of failure both effects are considered independently [5,6].

Bending under static loading

The total weight of the rear axle system is $W=50 \text{ kg}$ [0.9 kN] and is assumed to be evenly shared by the two rear wheels. The bending moment occurring along the shaft may simply be determined from taking the product of the wheel reaction force and the moment arm (from wheel centre to location of interest) (Figure 4).

$$M = W/2 \times (760-516) = 61 \text{ Nm}$$

As the shaft wheel hub diameter, $d = 12.5 \text{ mm}$, so bending stress,

$$\sigma = 32M / \Pi d^3 = 32 \times 61 / \Pi \times 0.0125^3 = 318 \text{ MN} / \text{m}^2$$

Torsion due to drive transmission

In this case the main primer mover is the engine of the Motorcycle. The power rating, $P = 13.42 \text{ kW}$ with an output speed, $N=5000 \text{ rpm}$ (Figure 5).

The maximum shaft driven speed = $5000 \times 16/40$ (sprocket teeth ratio) = 2000 rpm .

The maximum applied torque in the drive shaft, T

$$T = P/60 \div 2\pi N = 13420 \times 60 / (2000 \times 2\pi) = 64 \text{ Nm}$$

Shear stress due to drive, τ

$$\tau = 16T / \Pi d^3 = 16 \times 64 / \Pi \times 0.0125^3 = 167 \text{ MN} / \text{m}^2$$

Results of Fatigue Tests of Steel Specimens Subjected to Combined Bending and Torsion [6].

Material fatigue strength

Grade 40C8 IS: 1570-88 stainless steel grade is used for the shaft and the following material data is obtained from Appendix B of Ref 3 and stress concentration factor data also from Appendix C of Ref 3.

Ultimate Tensile strength $\sigma_t = 560-670 \text{ MN/m}^2$

Yield stress = 320 MN/m^2

Stress concentration factor at end of keyway, $K_T = 1.9$

[shaft with radial hole chart $d/D=4/12$]

Stress concentration factor at change of diameter, $K_T = 1.4$

[stepped diameter with fillet $r/D=3/12$]

Discussion of Findings

According to American Society of Mechanical Engineers code for the Design of transmission of shafts, the maximum permissible working stress in tension or compression may be taken as:

- 112 M Pa for shafts without allowance for keyway
- 84 M Pa for shafts with allowance for keyway

The maximum permissible shear stress may be taken as:

- 56 M Pa for shafts without allowance for keyway

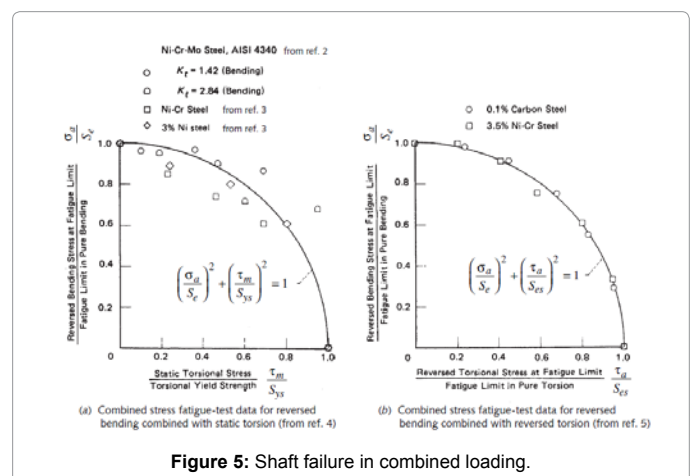


Figure 5: Shaft failure in combined loading.



Figure 6: Shaft showing the line of bending.

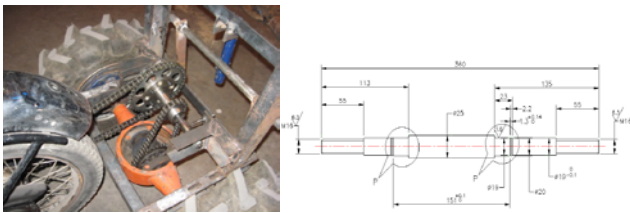


Figure 7: Modified design of the shaft.

- b) 42 M Pa for shafts with allowance for keyway

Calculations show that the bending stress magnitude is much greater than the shear stress caused by torsion due to accelerating and braking. And braking and this combined with much lower frequency of occurrence of fluctuation of shear stresses effectively eliminates torsion as a significant contribution to fatigue damage.

From the Figure 6 it appears that the shaft starts bending from the stepped portion and the maximum bending occurs at the end of the stepped portion of the shafts (Figure 6).

The Design Modifications

As the current shaft assembly is prone to bending failure after relatively a short operational time, modifications to the detail design are essential if longer life is to be obtained. A number of relatively simple

changes to the detail design would substantially improve the resulting safe life of future drive shafts in service. These can be achieved in a number of ways and some of the most effective and simple alterations are as follows:

- Shortening the total length of the shaft. This will ultimately reduce the overall length of the different portion of the shaft.
- Increasing the diameter of the shaft. This will also increase the diameter of the stepped portion at a ratio
- Increase the fillet radius. The shaft fails at the change of diameter and hence an increase from R3 to R5 would nevertheless be desirable in lowering the stress concentration (Figure 7).

Conclusions

The detail design of the wheel shaft assembly is found wanting. Even when operating under ideal circumstances, the predicted life is relatively short and leaves no margin for vehicle misuse, a distinct possibility with such an application. Recognition of the vulnerability to fatigue of a rotating component subjected to bending and torsion loading should lead automatically to taking fatigue preventative measures at the detail design stage. It is essential to avoiding having high stress concentrations at locations of greatest nominal stress if at all possible. Even taking relatively simple measures, such as those described will greatly improve component reliability without affecting manufacturing costs and prolong the life of components.

References

- Shigley JE (1986) Mechanical Engineering Design. McGraw Hill.
- Deutschman AD, Michels WI, Wilson CE (1975) Machine Design-Theory and Practice. Collier McMillan, NewYork.
- Juvinal RC, Marshek KM (1991) Fundamentals of Machine Component Design. European Journal of Engineering Education 16:283-288.
- Ryder GH (1958) Strength of Materials. Cleaver-Hume Press Ltd.
- Wulpi DJ (1986) Failures of shafts, Failure analysis and prevention. ASM metals handbook 11: 459-82.
- ASME Standard B106.1M-1985 Design of Transmission Shafting. American Society of Mechanical Engineers, New York.