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# Fatigue failure of an idle gear shaft of a gearbox

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**Abstract** This paper reports the results of failure analysis of an idle gear shaft of a gearbox in a hot re-rolling steel mill in Thailand. The shaft failed prematurely after only about 6,000 hours of service which was very much lower than the expected working life of 40,000-50,000 hours. The results showed that the shaft failed by fatigue fracture. Beach marks on the fracture surface were clearly visible. Fatigue cracks were initiated at two corners of the keyway of the shaft. Relatively large final fracture area of the fracture surface indicated that the shaft was under a very high stress at the time of failure. The root cause of the fracture is poor design and poor machining of the keyway. Too small fillet radii at the bottom of the keyway gave rise to stress concentration resulting in excessive stresses at the keyway corners. These high stresses led to fatigue crack initiation at the keyway corners followed by crack growth, and final fracture.

## 1. Introduction

Shafts are extensively used in machines and numerous engineering components including gearboxes. Failures of shafts not only result in replacement cost, but also in process downtime. This could have a drastic effect on productivity and, more importantly, late delivery. In the case being investigated, for example, the downtime was four days and 2,500 metric tonnes of steels were lost before the failed shaft could be replaced.

A gear shaft is usually subject to a high torsion and a bending moment as well as cyclic stresses which, when combined, may cause fatigue in the shaft. These factors may be influenced by stress concentrations which could drastically decrease the fatigue life and lead to the fracture of gear shaft [1]. Fatigue fracture is one of the most common causes of shaft failure. Fatigue failures are insidious and are therefore important considerations in mechanical designs [2]. Despite preventive measures taken during the design stage, fatigue failure can still occur due to either defects introduced during fabrication and/or degradation of shafts during service [3]. The most common failure modes of gear shafts were found to be, in decreasing order of frequency, as follows: fatigue, impact fracture, wear, and stress rupture [4]. Stress concentrations resulting from various causes such as protrusions, undercuts, poor machining, inadequate fillet radii were found to play key roles in shaft fracture [5,6]. Protrusions and troughs on the surface of the relief groove on shaft due to poor machining gave rise to high stress concentration leading to crack initiation [7]. In general, fatigue strength of engineering components increases with decrease in surface roughness. Manufacturing errors such inclusions, porosities, quench cracks, retained austenite, soft ferrite are potential sites for fatigue cracks to initiate internally [8, 9, 10]. Fatigue damage often starts at the outer surface of component due to surface integrity problems resulting



from manufacturing, and the presence of stress concentrations originating from surface topography [11]. The failed pinion shaft used in bowl mills was analyzing the cause for failure. Results revealed the initiation of cracks coming from the keyway corner. A small overload fracture zone was also observed in interior of the shaft suggesting low stress but high stress concentration torsional failure [12].

This paper aims at identifying the causes of failure of an idle gear shaft in a gearbox so that the reoccurrence of similar failure can be minimized or avoided in the future.

## 2. Background

The failed shaft was used in a gearbox in a continuous hot rolling steel re-bars mill in Thailand. The steel mill produces re-bars 12 mm to 20 mm diameter with a capacity of 35 metric tonnes per hour. The mill was designed for rolling steel billets with cross-sectional area of 120x120 mm to 130x130 mm square and six meters long. The gearbox was installed between pass number two and three, and was driven directly by a 900 kW AC electric motor. The rotational speed of the failed shaft was 20.8 rpm. The shaft failed prematurely by sudden fracture after approximately 6,000 hours in service. Normally, a gearbox has an expected working life of around 40,000-50,000 hours [13].

Details of the gearbox assembly showing the failed shaft and the location of the fracture are as shown in Figure 1a, and key dimensions of the failed shaft are shown in Figure 1b.

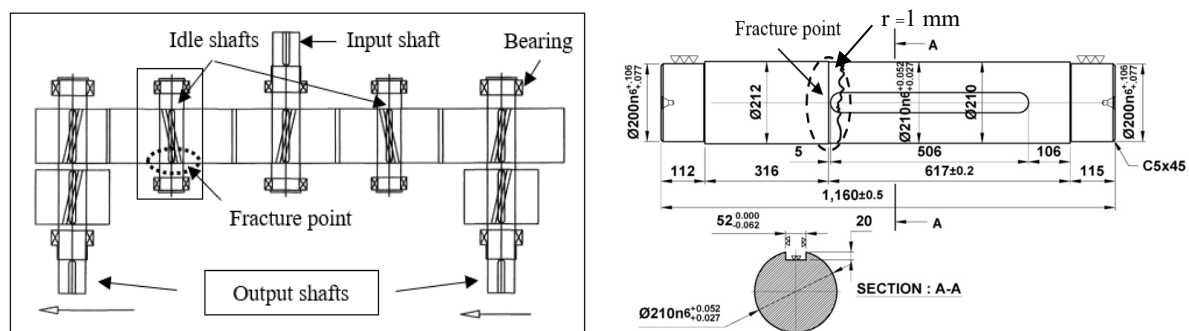


Fig. 1. (a) The gearbox layout showing failed shaft and fracture position (b) Key dimensions of the failed shaft

## 3. Investigation procedure

The failed shaft was first inspected visually and macroscopically. Relevant dimensions were measured and details of operating conditions noted. Fracture surface were examined visually as well as by using optical microscopy and scanning electron microscopy (SEM) techniques.

Hardness measurement at various distances from the crack origin of the keyway corner to the interior were carried out on a polished specimen using a Mitutoyo hardness tester model ARK 600 (steel ball diameter 1/16") with a load of 981 N. Chemical composition of the failed shaft material was analysed using a spectrolab spectrophotometer (Model: M8, Type: LAVWA 18A). Microstructure of the failed shaft material was examined under an optical microscope (LECO: IA32-Image analysis system).

Forces acting on the shaft and stress analysis were performed using bearinX software of Schaeffler Group. Applied stress was determined based on the data from actual operating conditions and relevant dimensions. The geometry of the model in the analysis reflected the actual dimensions of the shaft. The bending moment calculated using bending moment and shear calculator online civil engineer software.

## 4. Results

### 4.1 Visual examinations

#### 4.1.1 Fracture surface examination

The fracture surface of the failed shaft has two distinct portions both of which are rather flat as shown in Figure 2. Beach marks are clearly visible in one zone of the fracture surface. Such fracture surface appearance indicates that the shaft most likely failed by fatigue fracture. The final fracture surface area is approximately 80% of total fracture surface indicating that the stress in the shaft, hence the loads, must have been very high. Fracture surface appearance, as shown in Figure 2b, also indicates the crack most likely initiated at two corners of the keyway of the shaft. Ratchet marks in the fracture surface suggest further that the fracture is fatigue fracture.

The overall fracture surface was rather flat indicating that the final fracture was brittle by nature and that the shaft was subjected to very high triaxial stress before failure.

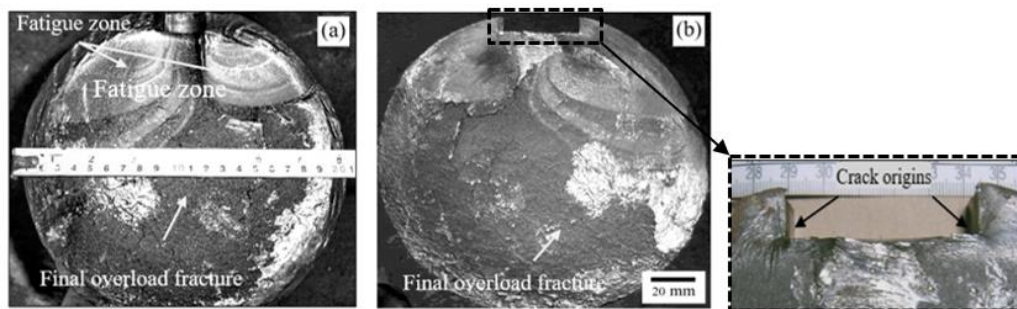


Fig. 2. The failed shaft the fracture surfaces

#### 4.1.2 Keyway examination

The keyway of the failed shaft was examined in more detail to identify characteristics and features of the keyway that could lead to crack initiation. It was observed that the keyway surface had distinctive rough machining marks as shown in Figure 3a. The keyway corners are quite sharp as shown in Figure 3b. Rough surface and sharp corners of the keyway are possible causes for the failure of the shaft as such features could lead to stress concentration and crack initiation.

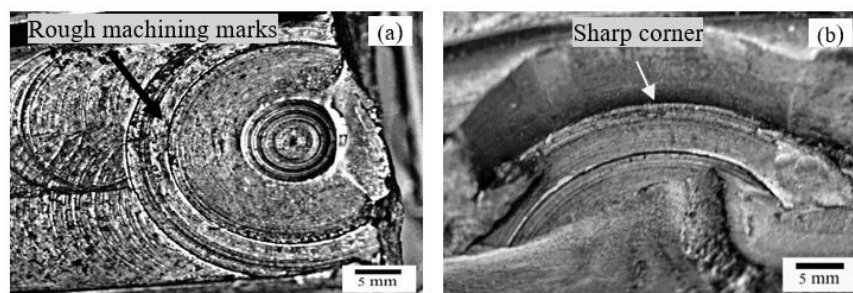


Fig. 3. Keyway (a) machining marks on surface (b) sharp corners

### 4.2 Keyway corners examinations

A sample from the material in vicinity of the keyway corner was cut, polished and examined to look for any abnormalities. The fillet radius of the keyway edge was found to be 0.228 mm as shown in Figure 4a which is much smaller than the recommended value of between 0.7-1.0 mm [14]. Two cracks at a

keyway corners were found as shown in Figure 4b. This demonstrates clearly that the crack did indeed start at the keyway corners.

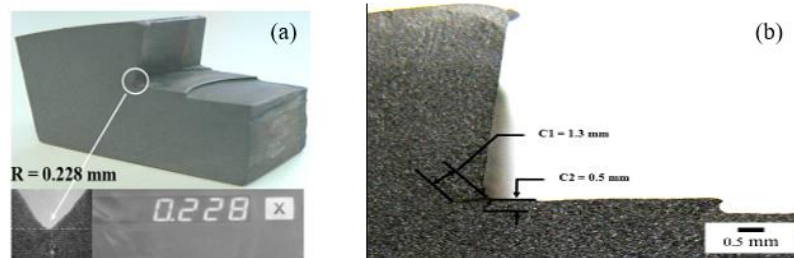


Fig. 4. Section the keyway showing (a) fillet radius (b) cracks at a keyway corner

#### 4.3 Hardness measurements

The hardness values measured at various distances from the crack region of the corner of the keyway are as shown in Figure 5. The hardness values were found to be between 92.7-94.9 HRB (195-210 HB) which were within specified limits of commercial machinery steels [15]. The hardness values indicated shaft material was in annealed condition to improve machinability. The result indicated that correct material was selected.

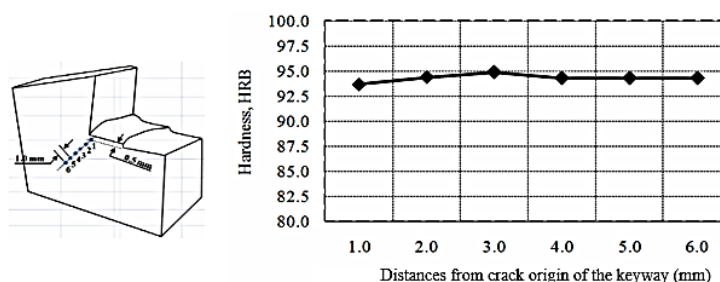


Fig. 5. Hardness values of the failed shaft

#### 4.4 Chemical composition analysis

The average values of the chemical compositions of shaft are shown in Table 1. The compositions are that of low alloy Cr-Mo steel to JIS- SCM440 standard [16]. The SCM440 belongs to a class of high strength structural steels and is commonly and widely used in making shafts [17]. The result shows that proper material was used for making the shaft.

**Table 1** Chemical compositions of the failed shaft and JIS-SCM440 (%wt)

Material	C	Si	Mn	Ni	Cr	Mo
Failed shaft	0.426	0.276	0.845	0.014	1.092	0.208
SCM440	0.38-0.43	0.15-0.35	0.60-0.85	0.25 (Max)	0.85-1.25	0.15-0.30

#### 4.5 Microstructural examination

Examination of the polished and etched example from the shaft material in the vicinity of a keyway corner revealed that there was a crack which extended from the keyway corner through the sample as shown in Figure 6a. This indicated that the crack started at the keyway corner then propagated towards the center of the shaft.

The microstructure of the failed shaft was a mixture of pearlite and ferrite as shown in Figure 6b. The structure of the shaft is typical structure for low alloy medium carbon steel in an annealed condition [18].

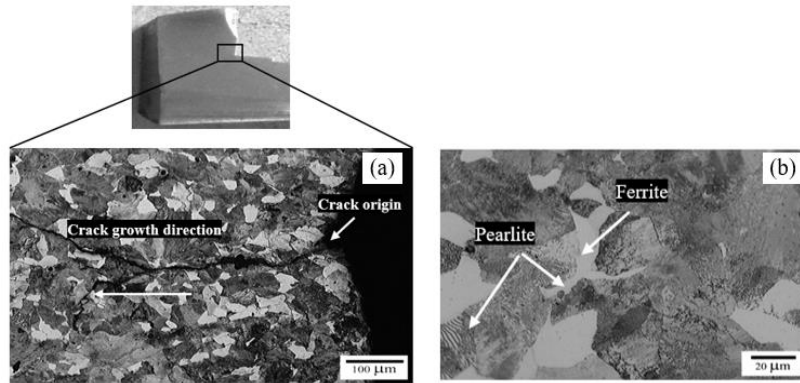


Fig. 6. The microstructure (a) crack growth direction and crack origin (b) (high magnitude)

#### 4.6 Force analysis

The transmitted torque ( $T$ ) was calculated using equation 1 [19].

$$T = \left( \frac{9,550 P_o}{N} \right) \quad (1)$$

where  $T$  is transmitted torque (Nm).  $P_o$  is power of electric motor (kW) and  $N$  is rotational speed of shaft in rpm.

In the present case the power of electric motor is 900 kW and the rotational speed of input shaft is 25.2 rpm. The transmitted torque carried by each shaft is 50% of the value in equation 1 as there are two output shafts in the gearbox. The forces on the shafts were calculated using bearinX software of Schaeffer (FAG) Company. The calculated values of  $F_{1x}$ ,  $F_{2y}$  and  $F_{1z}$  are -79.75 kN, 165.30 kN, 446.50 kN, respectively, as shown in Figure 7.

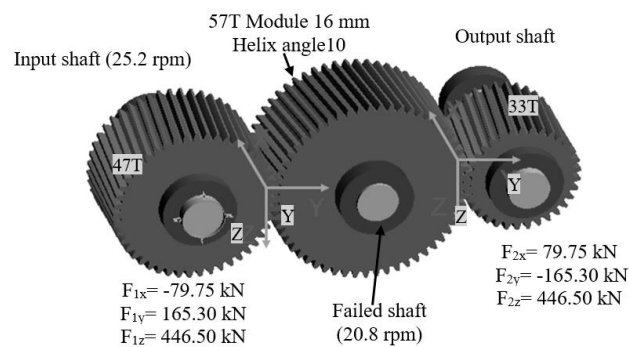


Fig. 7. Force analysis on the gear

#### 4.7 Stress analysis

The shaft under consideration is being subjected to bending force. So as the shaft rotates there is a fluctuation of stress, the point which is subjected to bending moment value is found and shown in Figure 8. Bending stress ( $\sigma_b$ ) at the keyway corner of failed shaft is found and shows in Table 2.

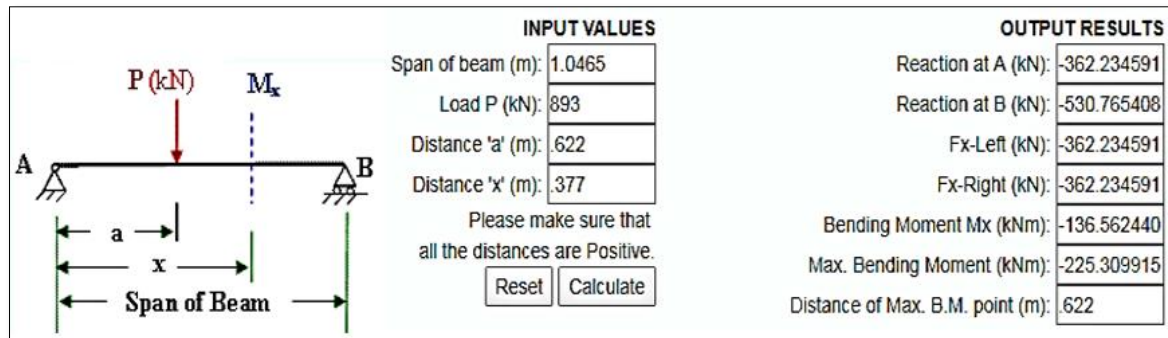


Fig. 8. Bending moment of the shaft

**Table 2** Calculation values of bending stress

Description	Values	Unit
Geometric moment of inertia (I) = $\pi(d^4/64)$	$9.541724 \times 10^{-5}$	$m^4$
c=dimeter of shaft (d)/2	0.105	m
Bending moment (M) at keyway corner	136.5	kNm
Bending stress ( $\sigma_b$ )=Mc/I	150.2	MPa

The maximum stress ( $\sigma_{\max}$ ) was calculated using equation 2 [20] and the stress concentration ( $k_t$ ) was selected using chart of theoretical stress concentration factor [20]. Fatigue stress concentration factor ( $k_f$ ) can be calculated using equation 3 [20].

$$\sigma_{\max} = k_f \times \sigma_b \quad (2)$$

$$k_f = 1 + q(k_t - 1) \quad (3)$$

$$q = 1 / (1 + \frac{\sqrt{a}}{\sqrt{r}}) \quad (4)$$

where  $\sigma_{\max}$  is maximum stress (MPa),  $\sigma_b$  is stress (MPa),  $k_t$  is stress concentration factor,  $r$  is fillet radius of the keyway corner,  $d$  is diameter of the failed shaft (mm),  $k_f$  is fatigue stress concentration and  $q$  is the notch sensitivity it can be defined from the Kunn-Hardarth formula in terms of Neuber's constant ( $a$ ) and the notch radius ( $r$ ). In this case, the measured notch radius at corner of keyway is 0.228 mm. The stepped change of diameters ( $D$ ), diameter ( $d$ ) of failed shaft is 212 and 210 mm respectively. Approximate fillet radius of stepped shaft is 1.00 mm as shown in Figure 1b. Stress concentration factor at change of diameter,  $k_t=3$  (approximate) (stepped diameter with fillet  $r/d=1/210$ ).

$$\sqrt{a} = \frac{104}{\sqrt{S_{ut}}} = 104/655 = 0.15877$$

$$\sqrt{r} = \sqrt{0.228} = 0.47740$$

$$q = 1 / (1 + \frac{0.15877}{0.47740}) = 0.7504$$

The calculated value  $k_f$  was 2.501. The calculated maximum stress included fatigue stress concentration factor ( $\sigma_{\max}$ ) is 375.6 MPa.



#### 4.8 Material fatigue strength

Fatigue strength or endurance limit for steels can be approximated from tensile strength data using equation 5 [20].

$$S'_e = 0.50S_{ut} \quad (5)$$

where  $S'_e$  is endurance limit of material,  $S_{ut}$  is tensile strength in MPa. From the chemical composition analysis and hardness measurement of shaft material, it was clear that shaft material was SCM440 steel in an annealed condition. The tensile strength of shaft material is assumed to be around 655 MPa [21]. The endurance limit of the steel is therefore approximately 327.5 MPa. The endurance limit of the shaft is expected to be lower than the above figure due to geometrical and other factors. The value can be calculated using equation 6 [19].

$$S_e = K_a K_b K_c K_d K_e K_f S'_e \quad (6)$$

where  $S_e$  is endurance limit of the shaft under investigation (MPa),  $K_a$  is surface factor,  $K_b$  is size factor,  $K_c$  is reliability factor,  $K_d$  is temperature factor,  $K_e$  is modifying factor for stress concentration, and  $K_f$  is miscellaneous effects factor. The values of various factors are as follows;  $K_a = 0.75$ ,  $K_b = 0.652$ ,  $K_c$ ,  $K_d$ ,  $K_e$ ,  $K_f = 1$  [20]. The endurance limit of this particular shaft, calculated using the above data, is approximately 160 MPa. The maximum stress value is high, exceed the fatigue strength of the material. It is no wonder that cracks initiated at the keyway corners.

#### 5. Discussion

It can be seen in Figure 2 that the final fracture area is relatively large covering about 80% of the fracture surface. The large final fracture area indicated that the failed shaft was heavily stressed (workpiece of steel bar too low temperature) at the time of final fracture. The fatigue cracks initiated at both corners of the keyway and ratchet marks between two cracks can be readily observed. The results indicated the stresses at the corners of the keyway were very high which was confirmed by the calculation in the previous section. The result is agreement with that of other investigators [22].

A detail drawing of fillet radii of the keyway in drawing as shown in Figure 1 did not specify the value of fillet radii of the keyway. The shaft manufacturer cannot be blamed for machining the keyway with sharp corners. Inadequate fillet radii of the keyway resulted from poor design in detail design phase.

The keyway surface contained distinctive very rough machining marks as shown in Figure 3. This is the result of poor milling of the keyway. Poor design and poor milling, resulting in low values of fillet radii and rough surface, are potential stress raisers that could lead to high stress concentration being created at the keyway corners which eventually lead to the initiation of the fatigue crack.

#### 6. Conclusions

1. The idle gear shaft of a gearbox under this investigation failed by fatigue fracture.
2. The fatigue cracks initiated at both corners of the keyway and propagated inwards until final fracture occurred.
3. Excessive stress combined with poor machining and inadequate fillet radii at the keyway corner, which gave rise to high stress concentration, are the main causes of the fracture.
4. It is recommended that extreme care must be taken in detail design phase of critical component such as detail specification of fillet radii of keyway corners and generous fillet radii should be provided. Machining must also be done with great care avoiding sharp corners and too rough surfaces.

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