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FAILURE ANALYSIS OF BEARING IN WIND TURBINE GENERATOR GEARBOX

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Abstract- This research paper describes the failure analysis of bearing in Wind Turbine Generator (WTG) gearbox. The two-stage filter element and the gearbox were examined at turbine tower top called nacelle to find out the reason for filter choke alarm in the turbine controller. Drive train alignment was ensured between the asynchronous generator shaft and the gearbox shaft to conclude the mode of bearing failure. The chemical and micro-structure together with hardness measurements were carried out on the failure bearing to check for deviation in the material specifications and heat treatment process. Detailed studies including visual inspection, Scanning Electron Microscope (SEM) and Energy Dispersive Spectrum (EDS) analysis were performed on the damaged bearing surface to determine the root cause(s) for failure. The peak temperature parameter in the gearbox at various places and power of the wind turbine were monitored for any abnormalities in the WTG. The study revealed that an intermediate non-drive end bearing failed in the gearbox due to excessive material removal on the rollers and bearing races due to contact wear (Scoring) followed by contact fatigue (spalling). The oil analysis, EDS, temperature and power analysis confirmed that contamination, presence of bauxite in the lubricant and overloading due to continuous peak power generation in high wind season were the reasons for bearing failure. Further, the SEM study infers that the mode of failure was fatigue fracture due to high cyclic fatigue phenomenon.
Keywords- Bearing failure, Contact wear, Contact fatigue, EDS, Oil analysis, SEM

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Introduction

The rolling-element bearings play a crucial role for the proper functioning of gearbox used in wind turbine generator (WTG). The reliability of bearing in special environments like corrosive, high temperature and power, high speed and high vacuum zone is very important. Many problems are arising in WTG gearbox operation even after conducting scheduled maintenance at stipulated intervals. Oil analysis, drive train alignment and tower torque are linked to bearing faults. In wind industries, bearing failure cannot be tolerated because it leads to catastrophic losses in power production due to down time, cost of repairing, and replacement of parts and so on. Study reveals that defects in bearings and bearings failure are the main reasons for gearbox failure in wind turbine generator.

In general, industrial gearbox works by converting high speed and low torque to high torque and low speed, but wind turbine generator gearbox works in the opposite way by converting low speed and high torque of rotor into low torque and high speed of an electric generator through shrink disc and fluid coupling. This is accomplished by using larger gears and bearings than that used in a typical gearbox. Generally, gearbox is not the most likely component to fail in wind industries. Rolling element bearings are one of the most common components in rotating machinery including wind turbine generator gearbox. As a general rule, machines do not break down or fail without some form of warning, which is indicated by an increased vibration. By repeated measurement and analysis of the vibration of the machine, physical inspection of the gearbox and through oil analysis, it is possible to determine the nature of severity of the defect, and hence predict the machine's useful life or failure point. Harish and Manish [1] have made an attempt to avoid unequal and non-uniform wear of elastomeric bearing used in marine propeller shaft. Hirani [2] has investigated the root cause failure analysis of the outer ring fracture of four-row cylindrical roller bearing used in the back-up roll assembly of cold rolling mills. Sadettin , Nizami and Veli [3] conducted a case study through vibration monitoring and analysis to diagnose the defect in rolling element bearings. Sudhakar and Joel [4] have investigated the defective bearing of a motor for determining the possible mechanism leading to their pre-mature failure during manufacturing. Sudhakar [5] has conducted a failure analysis on an automobile bimetal bearing which failed during storage before final assembly. Lliev [6] investigated the possible cause of bearing failure of a hydro-generator thrust bearing. Sankar and Nataraj [7] have investigated the failure of shear pins in wind turbine generator and proved that the misalignment between the driving and driven element leads to low cyclic fatigue failure. Many papers have been published on the prediction of defects in antifriction bearings using vibration signal analysis [8] and detection and diagnosis of bearing failure in helicopter gearbox [9]; but no attempt has been made to analyze the reasons for the failure of bearing in the field of wind industry, to the best knowledge of the investigator. Therefore, it is imperative and innovative to investigate the problem of failure of bearing in WTG.

Table-1- History of Bearing Failure

Date of commissioning of wind turbine	Date of failure	Turbine down time in days for replacement of an intermediate bearing
10/12/2004	20/02/2006	10
31/03/2006	06/06/2007	7
31/03/2004	09/01/2009	5
05/06/2007	05/07/2009	14
02/03/2007	07/09/2009	8
30/03/2006	09/09/2009	13
04/04/2006	16/08/2010	9

Problem Investigation

A study was carried out to investigate the frequency of failure of an intermediate stage non-drive end bearing of gearbox used in wind turbine generator. Table 1 gives the history of bearing failure in the WTG model under consideration. It is observed from Table 1 that the bearing fails within 2 years of usage against their recommended design life. In the event of such pre-mature failure, it demands immediate replacement of bearing for getting uninterrupted power supply from the wind turbine. Fig.1 shows the general arrangement of drive train components in wind turbine generator. The wind turbine under consideration consist of a speed increasing gearbox comprising of one planetary stage and two helical stages (Fig.2) to achieve the final speed ratio of 1: 74.917. The position and description of the bearings at the helical stage of the gearbox is given in Table 2.

The overloading of the wind turbine due to fluctuating wind force, non -synchronizing of pitching, sudden braking, sudden and frequent grid drops induces undesirable forces on the system. These forces not only damage and / or destroy bearings, seals and couplings but also the gearbox or generator eventually. Apart from this, the flanks of both drive and non drive end pinion have pressure marks due to excess axial movement of the shaft, scuffing wear and pitting wear when wind loads are high. Besides, any bearing in the gearbox fail while the wind turbine generator is in operational mode, it necessitates huge man hours and machine stoppage to set right the turbine.

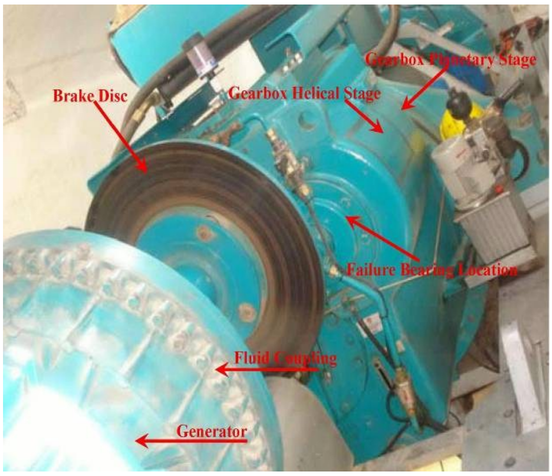

Fig. 1 - Arrangement of Drive Train Components in Wind Turbine Generator

Table-2- Bearing Details

Position of the gear box	Intermediate stage (IMS) Drive End	Intermediate stage (IMS) Non Drive End	High speed stage (HSS) Drive End	High speed stage (HSS) Non Drive End
1st Helical stage	Cylindrical roller bearing (230 NJEC / C3 D5	Cylindrical roller bearing (INA LSL 19 2330 A 0993 / 72 / VII)	Cylindrical roller bearing (328 NU EC / C3 D5)	Cylindrical roller bearing (228 NU EC C3J / MID) Four point angular contact ball bearing (328 QJ N2MA / C3 D)

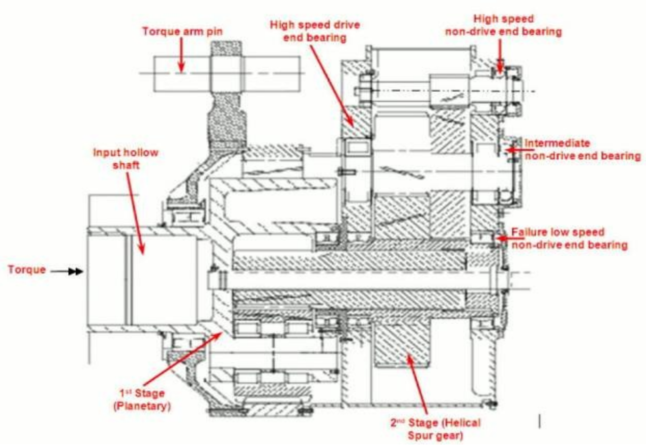

Fig. 2- Sectional view of the gearbox

Moreover in operation of turbine during high wind season leads to enormous generation losses, customer dissatisfaction and so on. Failure of bearing will harm the smooth functioning of gear trains. If any pinion fails in an intermediate stage, then either replacement of that pinion or overhauling the gearbox itself is difficult due to the tower height and weight of the gearbox. Moreover, de-erection of nacelle which is located at the top of the tower demands a huge crane (200 Ton to 400 Ton capacity) at wind turbine site to swap the gearbox in the turbine nacelle.

Fig. 3- Choked gear oil filter

There was filter choke alarm in the controller of wind turbine generator on 16th August 2010. On thorough scrutinization of the 2-stage (50 μm and 10 μm) filter element (Fig. 3), enormous metal particles (Fig.4) were traced out in the retainer plate (Fig. 3) located at the bottom of the filter. Immediately, the turbine was stopped for vibration analysis and thorough gearbox inspection to pin point the failure location. From the study, it was concluded that the failed component is an intermediate stage non-drive end bearing (Fig. 5). The persistent running of the gearbox with defective bearing caused damage on intermediate pinion (Fig. 6) due to tooth loading. It was observed from the site feedback that most of the wind turbines require significant repair and even complete overhauls within five to seven years or even before that benchmark as wind turbine downtime is attributed to gearbox related issues. The replacement of gearbox and lubrication attribute to 38% of the cost of turbine parts. Considering the cost of the gearbox, crane rental, labour and revenue loss, replacing a gearbox of 1.0mW to 2.0 mW turbines can cost more than 60 lakhs INR. The bearing failure occurs very often at the non-drive end of the intermediate stage. The main geometric dimensions of the failed bearing (Fig. 7) are listed in Table 3. A team comprising of operation, maintenance and product development engineers has been trying to explore the possibility of designing heavier bearing and employing new bearing manufacturing process for trouble free operation of the wind turbine.

Fig. 4- Metal particles at retainer plate

This research study is aimed to predict how and why the bearing fails in the wind turbine generator gearbox and arrive at the remedial measure / to be undertaken to minimize the occurrence of failure. Maintaining the cleanliness level of the gearbox oil and monitoring of the condition of the lubricant are vital for an optimum service life of the gearbox and performance of the wind turbine.

Fig. 5- Failed intermediate non-drive end bearing INA LSL 19 2330 A0993

Fig. 6- Damaged intermediate pinion

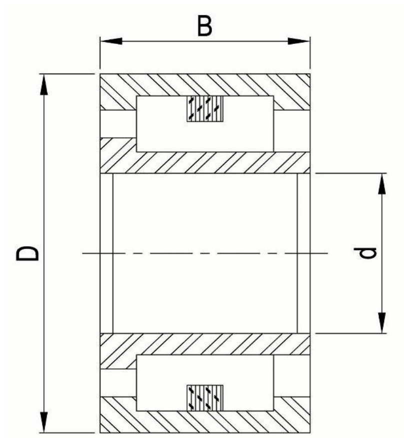

Fig. 7- Cylindrical roller bearing - INA LSL19 2330

Table-3- Geometric dimension of the failed bearing

Bearing location	Intermediate pinion shaft non-drive end
Bearing number	INA LSL19 2330 A 0993; 1 row semi fixed cylindrical rolling element bearing
Bearing inner diameter (d)	150 mm
Bearing outer diameter (D)	320mm
Width of the bearing (B)	108 mm
Weight of the bearing (Wt)	40.7 Kg
Speed limit (rpm)	4200

Failure Analysis

The gearbox has been designed for a power of 1250 kW and output speed of 1500 rpm to achieve the final step up speed ratio of 1:74.917. The high speed shaft of the gearbox is coupled with the fluid coupling with the help of brake disc, an axial damper and the 'H' spacer. The coupling in turn is connected with the generator by means of 'D' flange, a key and a locking screw. The technical team inspected the damaged bearing and intermediate pinion (Figs. 5 and 6) and predicted that the failures may be due to over-load by wind force or misalignment between the asynchronous generator shaft and the gearbox high speed shaft.

A. Examination of Drive Train Alignment and Oil Analysis

Initially, the position of the asynchronous generator shaft relative to the gearbox shaft was examined through laser-optical alignment technique (Fig. 8) to conclude the mode of failure, as most of the bearing failures in rotating machinery are attributed to misalignment in the drive train components. A vertical offset misalignment of 0.14mm and vertical angular misalignment of 0.12 mm /100 mm were found while inspecting the shaft alignment using easy laser measurement and alignment system. The generator foot positions were 0.07 mm at the front side and -0.16mm at the rear end. Similarly, the horizontal offset misalignment of 0.30mm and horizontal angular misalignment of 0.08 mm/100 mm were noticed. The generator foot positions were 0.22 mm at the front side and -0.39mm at the rear end. According to the maintenance manager, the acceptable limit of horizontal and vertical offset misalignment is 0.10 mm and the angular misalignment is 0.08mm / 100mm and this slight misalignment will not cause any catastrophic effect on the bearing life and on operation of the gearbox. So, it was confirmed from the alignment study that the failure of intermediate non drive end bearing and the pinion is not because of the mis-alignment in the system. Further, the gearbox under consideration is lubricated by mineral oil having a viscosity index of 460 for bearing and gear.

Fig. 8- Drive train alignment in WTG

The outcome of the previous oil analysis from the last bearing failure location, which was conducted in March -2010 is presented in Table 4.

The objective of this study is to know the maintained cleanliness level of the lubricant before the bearing failure and to know for any abnormality in the lubricant. During the examination, sodium (8 ppm), Silicon (1 ppm) and aluminium (less than one ppm) which is the agent for formulation of oxides were found in acceptable level. It is obvious from the Table 4 that the obtained cleanliness level as per ISO 4406-1999(e) indicated by the particle count 4µm, 6 µm and 14 µm was 21/19/15 which denote that the oil is started contaminating

Table-4- Oil Analysis Report

Test	Protocol	Unit	Result
TAN	ASTM D664-01	mg.KOH,/gm.	0.96
Colour	ASTM D1500:2007	ASTM	7.5
Flash Point (COC)	ASTM D92-05a	°C.	268
Kinematic Viscosity (at 40 Deg. C.)	ASTM D445-2006	mm^2/s.	456
Kinematic Viscosity (at 100 Deg. C.)	ASTM D445-2006	mm^2/s.	29.79
Viscosity Index	ASTM D2270:2004		97
Pentane Insolubles	ASTM D893-05a	% wt.	< 0.01
Water Content	ASTM D6304-04a-Procedure A	% wt.	0.016
Particle Count	ISO 4406-1999(E)		21/19/15
P Q Index			3
Elemental analysis	ASTM D5185-05		
Aluminium, Barium, Boron, Cadmium, Calcium, Chromium, Lead, Magnesium, Manganese, Molybdenum, Nickel, Tin, Titanium & Vanadium		ppm	< 1
Copper		ppm	3
Iron		ppm	12
Phosphorous		ppm	192
Silicon		ppm	1
Sodium		ppm	8
Zinc		ppm	74

B. Material and Geometry

The preliminary step in failure analysis is material identification. The bearing under consideration is made of EN31 steel that has metallurgical and processing characteristics. The material properties of EN31 are given below :

- Modulus of elasticity: 203396 N/mm^2
- Tensile strength: 2240 N/mm^2
- Yield strength: 2034 N/mm^2
- Density: 7833 kg/m^3
- Hardness: 58 - 63 HRC

The hardness of the inner race, outer race and roller of the bearing was measured in Rockwell Hardness Tester in 'C' scale and found to be in the range of 55-57 HRC which is the recommended hardness range for the bearing materials.

C. Compositional Analysis

The chemical analysis of the roller and races was carried out using ARL Spark Analyzer as per OES method to confirm the material specification. The result obtained from the laboratory is presented in Table 5. It is observed from Table 5 that the composition of failed bearing confirms to the specification of EN31material.

Table-5- Chemical Analysis of the Bearing Material

Element	Specification of EN31 (Wt%)	Composition of failed bearing
Carbon (C)	0.9 - 1.1	1.02
Silicon (Si)	0.15 - 0.30	0.31
Manganese (Mn)	0.25 - 0.45	0.67
Phosphorus (P)	<= 0.025	0.027
Sulphur (S)	<= 0.025	0.008
Nickel (Ni)	0.25 max	0.15
Chromium (Cr)	1.3 - 1.6	1.60
Molybdenum (Mo)	0.10 max	0.13
Vanadium (V)	---	0.012
Copper (Cu)	0.25 max	0.18
Lead (Pb)	0.002 max	0.016
Iron (Fe)	96.5 - 97.32	95.54

D. Visual Inspection

The visual inspection of the roller revealed that scoring lines representing contact wear were observed at the middle and drive end portion of the roller.

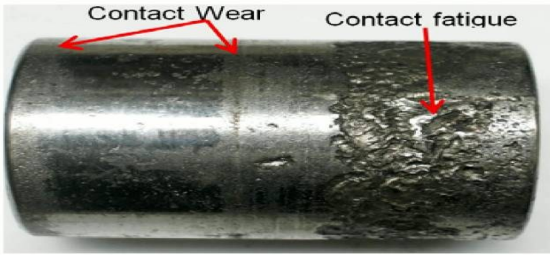

Fig. 9- Wear on roller

Surface and sub-surface fatigue wear that is contact fatigue (spalling) at the non drive end of the roller (Fig. 9) on several rollers (Fig. 10) due to overloading and excessive material removal. The same could be seen on both inner races (Fig. 11). The damaged outer race of the bearing is shown in Fig. 12.

Fig. 10- Damaged roller

Fig. 11- Wear on inner race

Fig. 12- Damaged outer race of bearing

E. Analysis of Temperature and Power

In order to identify the reason for excessive material removal on the rollers and races, the peak temperature and power at which the turbine (last bearing failure location) operated before bearing failure were downloaded from the turbine controller for evaluation. It is obvious from the temperature and power curve (Figs. 13 and 14) that the instant peak power generation of the turbine has gone up to 2024 kW continuously for the last two month (15/07/2010 to 15/08/2010) which is higher than that of the rated power of the gearbox (1250kW). This excess load on the gearbox and frequent change in wind speed caused overloading on the bearing resulting in severe damage.

The maximum peak power of the turbine and the temperature observed at various locations in the gearbox for last two month is presented in Table 6. It is evident from Table 6 that the temperature of gear oil sump, both drive and non drive of high speed bearing and an intermediate drive end bearing were found on maximum side but within the limit of 80° C.

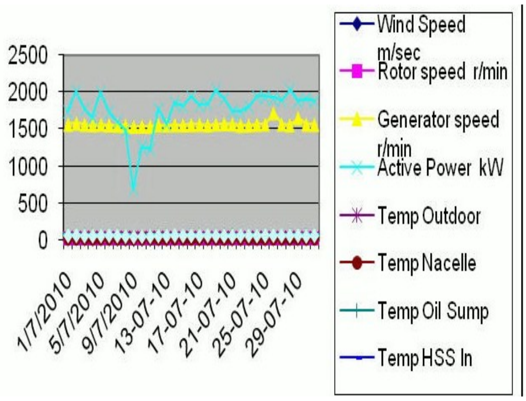

Fig. 13- Comparison of temperature and power for July 2010

Table-6- Peak Power and Temperature Details

Month	Active power (kW)	Gear oil sump temp (°C)	HSS drive end bearing (°C)	HSS non drive end bearing (° C)	IMS drive end bearing (°C)
Jul-10	2024	69	76	77	66
Aug-10	1853	71	78	79	69

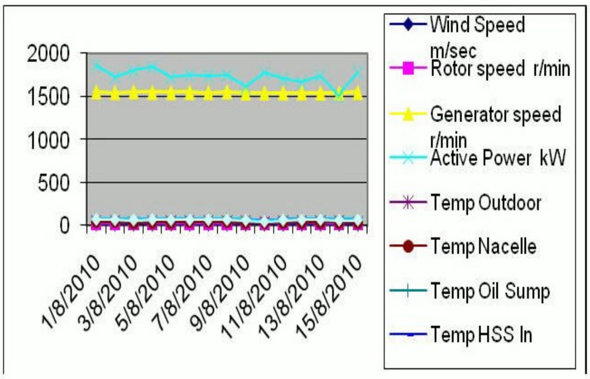

Fig. 14- Comparison of temperature and power for August 2010

F. Metallography

The specimen of failed bearing was prepared by fine polishing preceded by rough and intermediate and then etched with 2% Nital solution for examination. The microstructure of the damaged bearing was analyzed under optical microscope as well as Scanning Electron Microscope (SEM). The photographic view of the microstructure of bearing based on the experimental study is shown in Fig. 15. Tempered martensite structure with fine carbides (Fig.15) is seen on the bearing surface during microstructure examination. The specification of the material includes tempering treatment which was clearly evident from the microstructure examination.

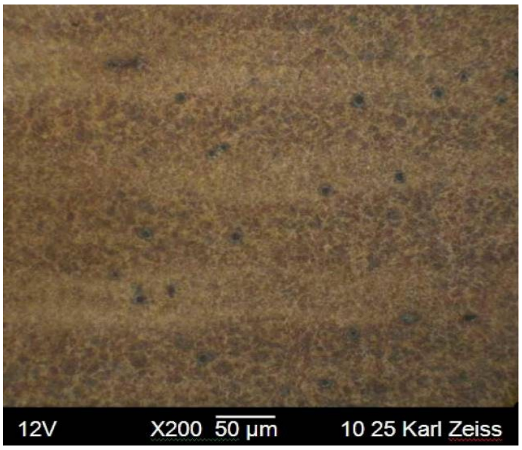

Fig. 15- Micro structure of bearing material. It consists of tempered martensite structure (200x)

G. Scanning Electron Fractography

The fractured bearing was cleaned ultrasonically using acetone, and examined under SEM and the outcome of the investigation are shown in Figs. 16, 17, 18 and 19. No surface preparation and coating were done on the surface of the bearing components during an examination. Furthermore, an accelerating voltage of 25 kV was applied during the SEM analysis. Wear debris and corrosion products (Fig. 16) were seen on the surface of the rollers as well as on the races of the bearing due to the presence of some kind of corrosive and wear elements in the gearbox oil. The mode of failure of bearing is fatigue and cleavage fracture (Figs. 17 and 18) because of high cyclic fatigue phenomenon. Fig. 19 shows the SEM image of the bearing with high cyclic fatigue and brittle fracture. Further, there was no evidence of cracks and micro-cracks on the bearing surfaces during the study.

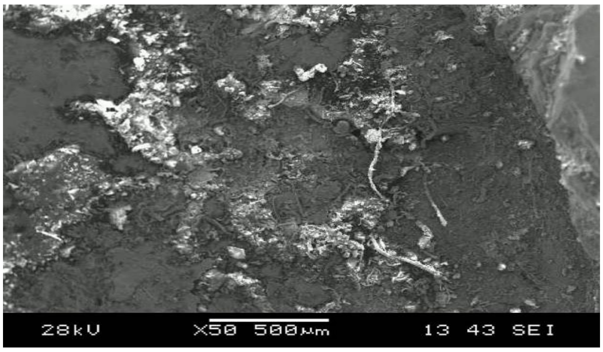

Fig. 16- SEM image of the bearing. Wear debris and corrosion products are seen

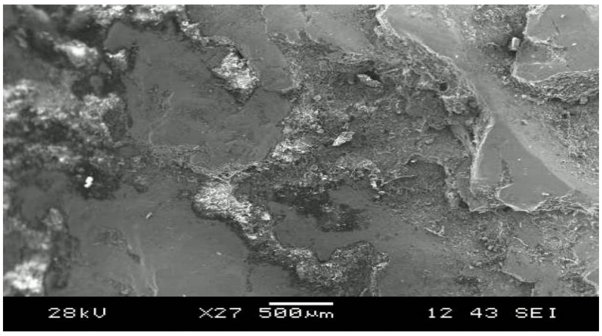

Fig. 17- SEM view of the bearing. Fatigue and cleavage fracture observed

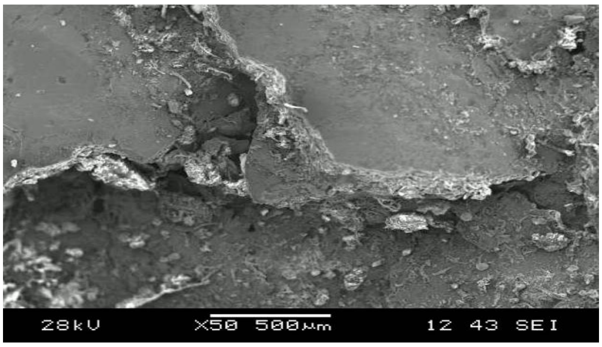

Fig. 18- SEM view of the bearing. Cleavage fracture observed

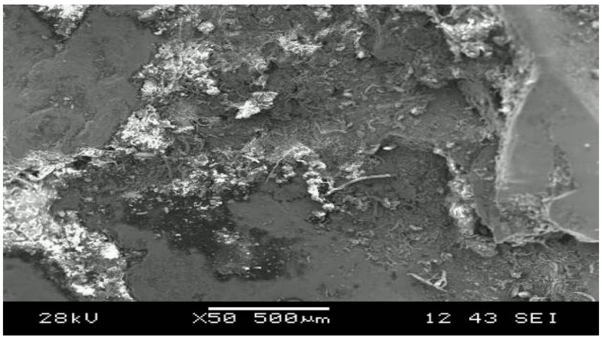

Fig. 19- SEM view of the bearing. High cycle fatigue and brittle fracture observed

Energy Dispersive Fractography

The damaged bearing was examined under Energy Dispersive X-ray Spectrometer (EDS) and the outcome of the investigation is illustrated in Figs. 20 and 21. Further, an accelerating voltage of 28 kV and probe current of 1.00000 nA were applied during the examination. In the EDS report, the horizontal and vertical axes represent the energy range (0 to 20 keV) and the counting rate (max 3250 cps) respectively. The bauxite elements observed in the first sample (Fig. 20) were sodium (Na), aluminium (Al) and silicon (Si) . The percentage of mass of silica was prominent (16.02 %) at 1.739 keV followed by Alumina (8.17 %) at 1.486 keV and sodium 2,22 % at 1.041 keV. Similarly, there were three bauxite elements viz, sodium, alumina and silicon were observed in the second sampling (Fig.21). In this case also, the percentage of mass of silicon was on the higher side (3.20 %) at 1.739 keV energy range. From the EDS study, it was evident that the presence of bauxite particles in the gearbox oil is due to the wear of gear and bearing. The bauxite will lead to excessive material removal during peak power generation on the rollers as well as on the races of the bearing.

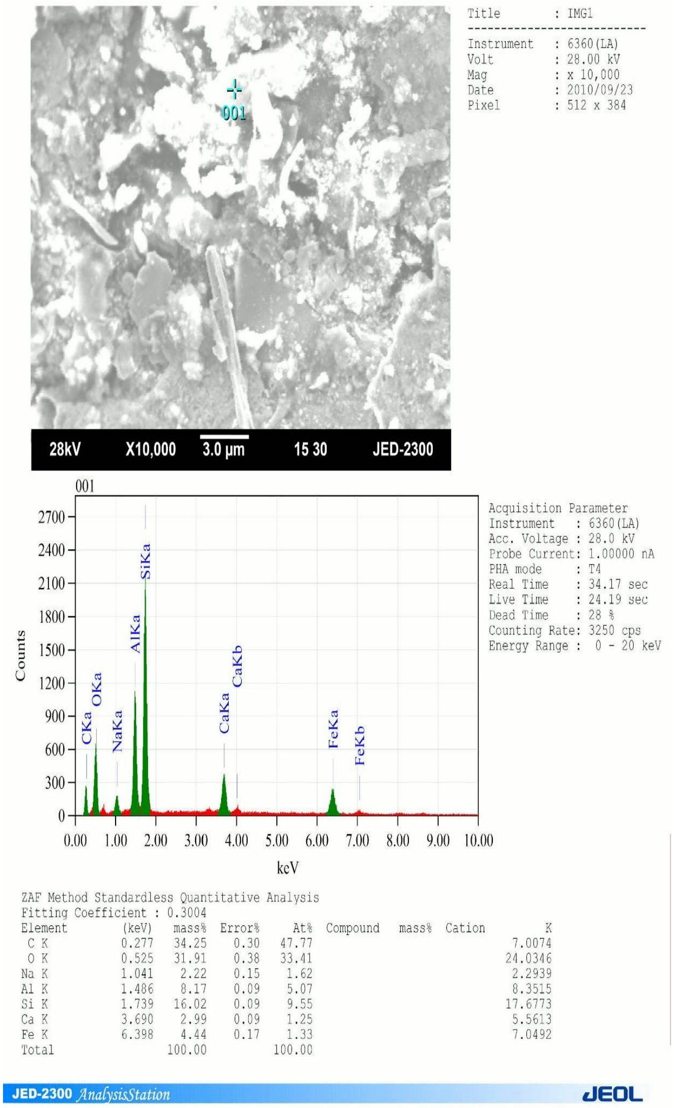

Fig. 20- EDS Report of the bearing for first sample

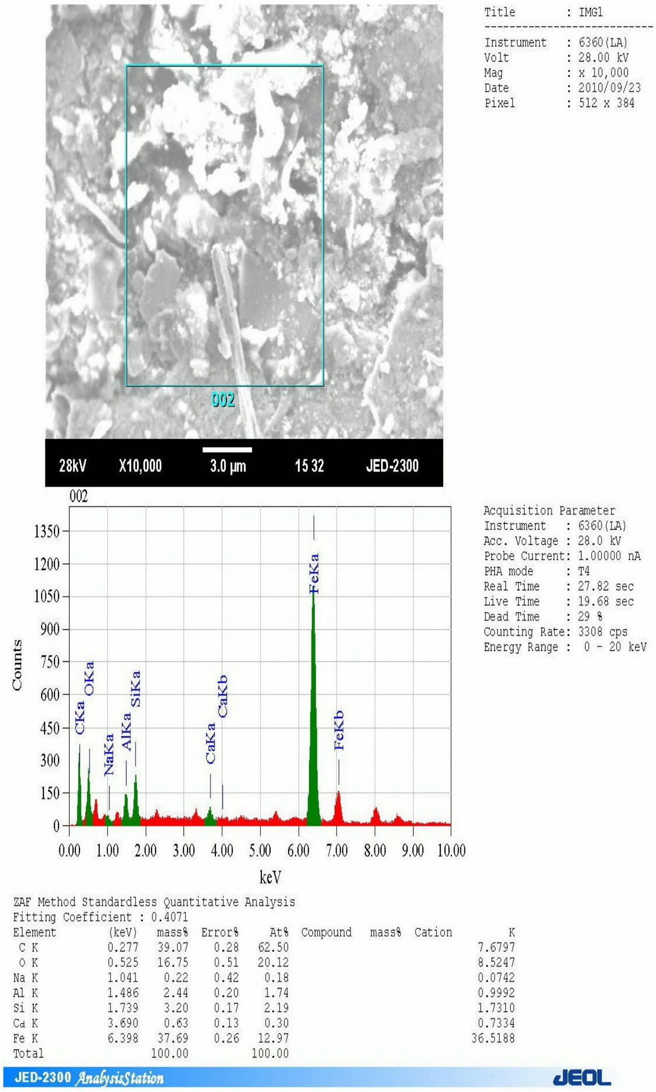

Fig. 21- EDS Report of the bearing for second sample

Discussion

The visual observation and the microstructure examination confirmed that an excessive damage of the roller as well as the surface of the race is primarily due to contact wear resulting in scoring line followed by spalling due to surface and sub-surface fatigue wear. The results of the chemical analysis (Table 5) revealed that the bearing material is in accordance with EN31 specifications. It is obvious from failure analysis that the failure of bearing is due to high cyclic fatigue fracture (Fig. 19). It is clear that the failure of the bearing has happened due to continuous peak power generation of the WTG during high wind season and the presence of bauxite element such as aluminum oxide, calcium oxides and silicon oxides in the gearbox oil.

Conclusion

The major conclusions of the present work are as follows:

- The chemical and metallographic examination has revealed that the root cause of failure of bearing is not because of defect in material and heat treatment process.

- The drive train alignment study confirmed that the bearing and an intermediate pinion failure is not due to the misalignment in the system.
- The debris collected at the filter unit resulted from contact wear (Scoring) followed by surface and sub-surface wear (Spalling) on rollers and races of an intermediate non-drive end bearing.
- The data collected at the wind turbine site and the experimental investigation has revealed that that the failure of the bearing is attributed to presence of bauxite elements in the gearbox oil which had triggered an intermediate stage bearing and the pinion failure.
- The pre-mature failure of the bearing is also due to overloading during continuous peak power generation of wind turbine in high wind season.
- Visual inspection and the SEM study have confirmed that the nature of bearing failure is fatigue fracture due to high cyclic fatigue phenomenon.

Acknowledgements
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