CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter critically reviews the literature pertinent to Wind Industry; explore the fault diagnosis and failure analysis of WTG gearbox and its components. The basics of WTG gearbox is discussed in section 2.2 wherein the problem encountered by the gearbox is explained. Section 2.3 reviews a failure characteristic of horizontal axis wind turbines and its effect. Section 2.4 thrashes out the literature on WTG components failure and Section 2.5 discusses research theory on critical components in gearbox unit.

2.2 WIND TURBINE GENERATOR GEARBOX AN OVERVIEW

The wind turbine gearbox assembly comprises a set of gears, shafts and bearings that are mounted in an enclosed lubricated housing. Gearbox Unit (GU) assembly in WTG are available over wide range in sizes, capacities with different speed ratios. GU is used for converting the low input speed at higher torque generated by wind force to higher output speed with lower torque. Normally, WTG gearbox comprises of one planetary stage and two helical stages to attain the final speed ratio of 1:25.60 to 1:98.828 (step-up speed drives). Spur gears are used in planetary stage in place of helical gear for sun, planet and ring gear in some gearboxes. Helical gears have been used in parallel or crossed axis helical stage as a power transmitting gear, owing to their relatively smooth and silent operation, large load carrying capacity and
higher operating speed by Zhang and Fang (1999). Figure 2.1 illustrate the kinematic arrangement of gears in WTG gearbox.

![Kinematic View of the Gearbox Assembly in WTG](image)

**Figure 2.1 Kinematic view of the gearbox assembly in WTG**

The WTG gearboxes experience higher loads due to the fluctuation in the wind force, sudden braking because of the frequent grid drops and vibration in the critical parts. Intermediate stage bearing balls and rollers vibrate against the outer and inner race causing the lubricant to squeeze out of the highly loaded contact areas. This phenomenon will cause wear and rubbing effect which will severely damages the bearings. The increased speed of the gearbox induces abnormal noise and vibration during the operation of the gearbox at full load; and both drive and non-drive flank of the high-speed and intermediate pinion experiences more standstill or pressure marks and have scuffing wear and pitting wear which ultimately leads to frequent pinion failure by Errichello et al (2011).
Tae Hyong Chong and Jae Hyong Myong (2001) reported that the major cause of vibration and noise in a gear system is the transmission error because of rotation delay between driving and the driven gear caused due to manufacturing error, an alignment error in assembly and elastic deflection at the time of loading. Negash Alemu and Ing Tamrat Tesfaye (2007) concluded that the gear noise is closely related to the transmission error, if a pinion and a gear have ideal involutes profiles running with no load and hence they should theoretically run with zero transmission error.

2.2.1 Problems Associated with Functioning of WTG Gearbox

i) As discussed in Section 1.3, most of the problems with the current fleet of wind turbine gearboxes are generic in nature i.e. problems are not specific to a single gear manufacturer or turbine model. Over the years, most wind turbine gearbox designs have converged to a similar architecture with only a few exceptions. Therefore, it is an opportunity to take the issues related to wind turbine gearbox and thereby to find the root causes for its failures.

ii) Most of the gearbox failures do not begin as gear failures or gear-tooth design deficiencies. It is observed that failures happens at several specific bearing locations under certain applications, which may later advance into gear teeth failure as bearing debris and excess clearances cause surface wear and misalignments. Field-failure assessments indicated that 10% of the gearbox failures accounts to manufacturing anomalies and quality issues that are gear related; however it is not the primary source of the problem.
iii) The majority of the wind turbine gearbox failures appear to initiate in the bearings. These failures are occurring in spite of the fact that most gearboxes have been designed and developed using the best design practices available.

iv) As such, lessons learned in solving problems on the smaller scale can in turn be applied directly to future wind turbines at a larger scale with minimum cost.

A major factor contributing to the complexity of the problem is that much of the bearing design-life assessment process is proprietary to the bearing manufacturers. Gearbox designers jointly working with the bearing manufacturers initially select the type of bearing for a particular location and determine the specifications for rating. The bearing manufacturer conducts a fatigue life rating analysis to find out whether the correct type of bearing has been selected for the specific application and location. Generally, a high degree of faith is required to accept the outcome of this analysis because it is done with little transparency. Even though bearing manufacturers claim that they have adhered to international bearing-rating standards (ISO 281:2007), each manufacturer have used the internally developed design codes. They have the potential to introduce significant differences that can affect the calculated bearing life without revealing the details to customers. A new code is needed in the public domain that will give the industry a common method for due diligence in bearing design (KISSsoft AG 2008).

The mechanical brake is one of the two independent brake systems i.e. active side and passive side in a wind turbine. As a consequence of the gearing in the turbine, the mechanical brake is often placed on the high-speed shaft as this allows the brake system to be as small as opposed to placing it on the low-speed shaft. The gearbox for wind turbine is designed in such a way that it has to sustain huge amount of torque while the brake is applied. Most
brake systems in today’s wind turbines supply the hydraulic brake caliper with maximum pressure when applied. Heege et al (2007) demonstrated that hydraulic brake systems produce backlashes in the gearbox, i.e the end may cause the fatigue load to be underestimated by current gearbox design methods. Schlecht and Gutt (2002) showed that when maximum brake torque is applied, it excites oscillations in the rotor shaft that have amplitude nearly twice as high as the nominal shaft torque. Therefore, it is possible that the mechanical brake is the root cause for large number of the gearbox failures. One such failure had been seen recently at Hornslet in Denmark, where a gearbox suffered a catastrophic failure caused by load from the mechanical brake system (Riso 2008). In recent years the term”soft brake” has been subject of interest from companies such as Svendborg Brakes (2001), General Electric Company (2008), Nordex Energy GmbH (2009). All three companies have filed patents for ”soft brake” systems as described in (Svendborg Brakes 2001, General Electric Company 2008 and Nordex Energy GmbH 2009). All the three patents are concerned with reducing the excessive dynamic load peaks and vibrations during emergency stops. Comparing “hard brake” with the classic approach where full pressure is applied, it is clear that the “soft brake” is superior. However, ”soft brake” still leave room for improvement as relatively large oscillations still occur – especially after the shaft has stopped (Svendborg Brakes 2001).

2.3 REVIEWS ON WIND TURBINE GENERATOR

Wind turbine encompasses multidisciplinary applications of modern renewable energy system. In particular, the operation and maintenance practices are significant and complexity in nature (Alsyouf 2008). The wind energy industry has experienced high rates of gearbox failure from its inception (McNiff et al 1990). Several researchers have focused their study on WTG failure of the gearbox since over the past 15 years. Most of the
failures were identified either through condition monitoring technique or FFT analyzer for the gears and bearing faults. Over the past two decades the wind turbine manufacturers, gear designers, bearing manufacturers, consultants and lubrication engineers all are working together to improve the load prediction, design, fabrication and operation.

Despite reasonable adherence to these accepted design practices, wind turbine gearboxes have yet to achieve their life of twenty years with most systems, requiring significant repair or overhaul well before the intended life is reached (Windpower Monthly 2005, Rasmussen 2004, Tavner and Xiang 2006). Since gearboxes are the most expensive components in the wind turbine system, the higher than the expected failure rates are adding to the cost of wind energy. In addition, the future uncertainty of gearbox life expectancy is contributing to wind turbine price escalation. Turbine manufacturers add large contingencies to the sales price to cover the warranty risk due to the possibility of premature gearbox failures. In addition, owners and operators build contingency funds into the project financing and income expectations for problems that may show up after the warranty expires. To bring down the cost of wind energy back to a decreasing trajectory, a significant increase in long-term gearbox reliability needs to be demonstrated.

Increasing the reliability of wind turbine gearboxes remains a challenge for many in the wind industry (Robb 2005). One of the key issues in wind turbine is misalignment of gears and bearings. Many existing analysis methods fail to take account of the full gearbox system. Instead they rely on individual component analysis. Therefore accurate determination of component interactions is not considered. All these components are connected to each other through gear meshing, bearing mounting and other connections. The housing is normally mounted via the torque arms, which take torsional load only. All these components are interconnected with each other and hence
changing certain parameters in any component will inevitably affect other components. Durability analysis of the bearings can be conducted using the following four different methods:


ii) Adjusted Life: Load zone factor is applied to the ISO Life to account for misalignment, clearances and other effects

iii) Advanced Life: Evaluates the bearing capacity and contact conditions from detailed bearing geometry

iv) DIN ISO 28:1: New standard (DIN ISO 281: 2003) with the extended life theory. This takes account of lubricant cleanliness and temperature effects

During field operation, wind turbine gearboxes are subject to continuously fluctuating loads caused by variations in the wind and control actions. These conditions must be considered in the design process (ISO/IEC 61400-4 DIS). The Gearbox Reliability Collaborative tests were conducted to build an understanding of how normal wind turbine loading conditions and transient events translate into gear and bearing response, including reactions, load distribution, displacements, temperature, stress, and slip (IEC 61400-1 2004 and ISO 281-2010 2010).

To make the wind power cost more competitive, there is a necessity to improve the turbine reliability and availability. The gearbox is the costly drive train component to maintain throughout the expected 20-year design life of a wind turbine. A retest on the damaged gearbox was conducted by the National Renewable Energy Laboratory (NREL) for 2.5 MW using Dynamometer Test Facility (DTF) to conclude the performance and reliability
of the wind turbine drive train prototypes and commercial machines (Musial 2000). Followed by the dynamometer retest, the gearbox was inspected in a rebuild shop and a detailed failure analysis of the gearbox was conducted by dismantling the gearbox unit (Errichello 2012). Complete lists of actual damage found from the failure analysis are shown in Table 2.1.

**Table 2.1 Actual damage on the test gearbox**

<table>
<thead>
<tr>
<th>Damage</th>
<th>Instances</th>
<th>Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>HSS gear set</td>
<td>Scuffing</td>
</tr>
<tr>
<td>2</td>
<td>HSS downwind bearings</td>
<td>Overheating</td>
</tr>
<tr>
<td>3</td>
<td>ISS gear set</td>
<td>Fretting corrosion, Scuffing and Polishing wear</td>
</tr>
<tr>
<td>4</td>
<td>ISS upwind bearing</td>
<td>Assembly damage, Plastic deformation, Scuffing, False brinelling, Debris dents and Contact corrosion</td>
</tr>
<tr>
<td>5</td>
<td>ISS downwind bearings</td>
<td>Assembly damage, Plastic deformation and dents</td>
</tr>
<tr>
<td>6</td>
<td>Annulus/ring gear and sun pinion</td>
<td>Scuffing and polishing and Fretting corrosion</td>
</tr>
<tr>
<td>7</td>
<td>Planet carrier upwind bearing</td>
<td>Fretting corrosion</td>
</tr>
<tr>
<td>8</td>
<td>Sun pinion thrust rings</td>
<td>Fretting corrosion and Adhesive wear</td>
</tr>
<tr>
<td>9</td>
<td>Oil transfer ring for planet carrier</td>
<td>Polishing</td>
</tr>
<tr>
<td>10</td>
<td>LSS seal plate</td>
<td>Scuffing</td>
</tr>
<tr>
<td>11</td>
<td>LSS downwind bearings</td>
<td>Abrasion</td>
</tr>
<tr>
<td>12</td>
<td>HSS shaft</td>
<td>Misalignment</td>
</tr>
</tbody>
</table>

Figure 2.2 shows clear scuffing marks on the high speed stage gear. The root cause of the fault was due to the assembly damage and oil starvation resulting from the two oil loss events in the field test. Among the 12 damaged items listed in Table 2.1, the consensus reached among the
sixteen round robin partners was that the first seven should be detectable by vibration analysis. Misalignment is one of the cause of bearing overloading, leading to premature bearing failure. Whittle (2011) investigated the effect of misalignment between the gearbox high speed shaft and the generator shaft on high speed bearing life. Typical drive train design includes a flexible coupling to protect the bearings from misalignment forces. Wind turbine couplings are custom designed to transmit torque well (i.e., to have high torsional stiffness) while tolerating high parallel or angular misalignment (i.e., to have low parallel and angular stiffness).

![Gear damage in high speed stage](image)

**Figure 2.2 Gear damage in high speed stage**
Courtesy: NREL/PIX 19599

Piotrowski (2007) suggested that “special alignment considerations must be taken in to account for an equipment that is started and stopped frequently like the wind turbine generator or where loads may vary considerably while running.” The unique aspects of wind turbine drive trains should be kept in mind, including the frequency of starts and stops, the high
power density, the flexibility of the bedplate, and varying loads. Erichello (1994) developed a wind turbine gearbox health management practices for wind project asset managers to minimize the risk of premature wind turbine gearbox failure. When oil is added to a gearbox, there is a possibility for the contaminants to enter the gearbox. ISO 81400-4 recommends maintaining oil cleanliness levels of -/-14/-11 (codes defined in ISO 4406:1999) or cleaner for oil added to a gearbox. In practice, typical oil filling methods may cause oil to be far dirtier than this level. Oil manufacturers, filtration system manufacturers, gearbox manufacturers, and turbine manufacturers all have oil filling protocol. Hydac has formulated a wind turbine gearbox lube oil change procedure (Santos 2010). Clean oil is essential for the gearbox longevity. Bearing life has been shown to decrease as coarser as filtration levels are used (Zaretsky 1992). Figure 2.3 shows the relationship between filter rating and bearing life.

![Figure 2.3](image)

**Figure 2.3** Relationship between filter rating and bearing life

*Courtesy: NREL*
Bearing life calculations include consideration of lubricant cleanliness, temperature, and other factors. Caution should be taken when extrapolating these specific results from small to large bearings. There is broad data indicating that oil containing excessive contaminants can cause premature wear of bearings and gear meshes (Nixon et al 2009). In order to maintain adequate cleanliness level and pump oil to required capacity of oil, oil should be filtered through two circuits; i) an online and ii) an offline circuit (Skriver 2000). According to Lauer (2011), Fitch (2008) moisture in oil may contribute to the following problems:

i) **Corrosion**: Water gives acids increased potential for corrosion.

ii) **Additive depletion**: May deplete antioxidants in the oil, and diminish the performance of other additives in the oil.

iii) **Flow restrictions**: Water is polar, and attracts impurities that are also polar (oxides, dead additives, particles, carbon fines and resin, for instance) to form sludge balls and emulsions. These amorphous suspensions can enter oil ways, glands and orifices that feed bearings with lubricating oil, impeding flow. In subfreezing conditions, free water can form ice crystals which can interfere with oil flow as well. Filters can also be clogged.

iv) **Aeration and Foam**: Water lowers the interfacial tension of oil, increasing the risk of foaming.

v) **Impaired film strength**: Water’s limited film strength limits its capability to bear loads at high pressure.
vi) **Microbial contamination:** Water promotes microorganisms such as fungi and bacteria. Over time, these can form thick biomass suspensions that can plug filters and interfere with oil flow. Microbial contamination is also corrosive.

vii) **Hydrogen-induced fracture:** There is a current hypothesis stating that risk of hydrogen embrittlement is posed by both soluble and free water. Sulphur from additives, mineral oils, and environmental hydrogen sulfide may accelerate the progress of the fracture.

viii) **Micropitting:** The mechanism of water promoting micropitting is unproven (Erichello 2012), but water is suspected as a factor in micropitting of gears and bearings.

Unless moisture quantities are high enough to result in direct visual evidence of the presence of free or emulsified water, detecting moisture in oil requires testing oil samples or online sensing. For oil sample testing, the Krackle test is not valid for moisture content below 1000 parts per million (ppm) and is not recommended for wind turbine gearbox applications. Karl Fischer Method C (ASTM D6304) is the recommended test method. The typical moisture levels in oil samples taken from new gearboxes are as low as 300 ppm and as high as 1500 ppm according to Karl Fischer Method C. Cantley (1977) developed a relationship between moisture content and bearing life (Figure 2.4).
Majumdar and Farooq (2011) recommended that membrane dehydrators are an emerging technology that shows promise for online moisture removal. The moisture content may vary with season; a 6-month sampling frequency may not capture this variation. The draft AWEA O&M practices recommends increasing sampling frequency so that seasonal moisture data are captured.

In addition to moisture in the oil, moisture in the air within the gearbox headspace can cause corrosion of internal components or migrate into the oil. Herguth Laboratories (2001) suggested for operating the gearbox with filtered and regenerable desiccant breathers or a sealed breather, to avoid ingestion of airborne particles in to the gearbox and also to reduce airborne moisture content. Breather clogging has been a possible cause of seal failure and oil leakage. Oil analysis is used to determine the condition of gearbox
lubricant. Poley and Kittiwake (2011) has proposed the following three-tier approach for oil analysis.

i) **On-line oil analysis:** On-line oil analysis is deployment of sensing systems on the turbine as a first method of detecting oil degradation. This includes direct sensory examination of a sample (Johnson 2010).

ii) **On-site oil analysis:** On-site analysis is conducted with a testing kit, and this level of analysis allows for checking anomalies detected by the on-line system.

iii) **Laboratory oil analysis:** This level of analysis provides the most complete information.

For all tiers of this approach, data should be collected and tracked in order to identify trends in oil condition which may indicate the problem with the gearbox. For determination of oil cleanliness in accordance with ISO 4406, two primary methods of laboratory oil particle counting namely i) optical methods ii) pore blockage methods are commonly used. Pore blockage however requires more time and labour resulting in higher cost. Samples with high water content should be tested with pore blockage methods in order to maximize the accuracy of the measurements. Some failure modes such as subsurface fatigue or gear cracking may occur without generating debris. Debris monitoring however is one of the source of information that can provide an indication of gearbox health, and it can be installed on the gearbox and provide output that in turn is integrated in to the turbine monitoring system or SCADA systems. The location of an on-line particle counter (debris monitor) will affect the results. For better result, it should be placed prior to the filter unit (i.e. upstream).
2.4 COMMON FAILURE AND ITS EFFECT IN WTG.

2.4.1 Gear Failure

Gear teeth damage at the intermediate and high speed pinion are the most common failures in WTG gearbox. As discussed in section 1.2, if the pinion fails in an intermediate stage of the gearbox, it necessitates either replacement of the entire intermediate stage or the gearbox itself due to the complication in the gearbox assembly. The constraint for either replacement of the entire intermediate stage or the gearbox demands for huge capacity (400 Tonne or 800 Tonne) crane at the wind turbine site for swapping. The broken gear tooth has to be collected from gear oil sump before draining the entire oil from the gearbox. Also, the gearbox has to be flushed with same grade flushing oil and new oil has to be maintained for further operation of the gearbox. Further, the gear oil filter element has to be changed with new one and thorough inspection of the gear oil pump is also required for smooth functioning of the gearbox. All the above said expenditure are unavoidable cost which will lead to increase in operation and maintenance cost of the wind turbine.

Helical gears are extensively used in numerous engineering applications including wind turbine generator gearbox to carry huge dynamic loads. As stated in section 1.5.1, mostly gearbox works for converting high speed and low torque to high torque and low speed (step down) and vice versa for step up. Wind turbine gearbox works by converting low rotor speed and high torque to low torque and high generator speed (step up gearbox) with the help of drive train components such as shrink disc, brake disc, coupling assembly, axial dampers etc.
2.4.2 Bearing Failure

In wind industry bearing fault accounts for over 41% of all gearbox failure (Allbrecht 1986). The bearing temperature measurement shall provide useful information about the motor condition, gearbox health and the bearing health (Randy 1995). Continued stress causes fragments of the material to get break loose, producing localized fatigue phenomenon known as flaking or spalling (Riddle 1985). Contamination and corrosion are the key factors of bearing failure of the harsh environment present in most industrial settings. The lubricants are contaminated by dirt and other foreign matter that are commonly often present in the industrial environment. Under and over-lubrication are also some of the other causes of bearing failure. In either case, the rolling elements are not allowed to rotate on the designed oil film causing increased levels of heating. The excessive heating causes the lubricant to break down which reduces its ability to lubricate the bearing elements and accelerates the failure process. In addition, installation problems are often caused by forcing the bearing onto the shaft or in the housing. This produces physical damage in the form of brinelling or false brinelling of the raceways which gives way to premature failure.

Misalignment of the bearing is also a common result of defective bearing installation. Regardless of the failure mechanism, defective rolling elements in the bearings generate mechanical vibrations at the rotational speeds of each component. Bearing failure is one of the major head ache in the wind turbine generator (SKF Product Information-401 1994). Following are the types of bearing used normally in the WTG gearboxes;

i) Cylindrical roller bearing

ii) Paired taper roller bearings

iii) Four point angular contact ball bearings

iv) Spherical roller bearings.
The expected design life of the WTG gearbox is 20 years, but the gearbox fails within 5 years because of shock loads due to fluctuating wind force, sudden grid drops, non-synchronization of pitching and sudden braking. However overloading of the gearbox and contamination in the lubricant make the bearing to fail within 50,000 hours of the turbine operation. From the field survey conducted over the past 6 years it was proved that the most common types of bearing failure in the wind turbines gearbox are; i) Fatigue failure and ii) Pitting (surface fatigue) and Spalling (sub-surface fatigue). The effects of bearing failure in the wind turbine generator are as follows;

i) The minimum time required for replacing the bearings at both drive end and non-drive end of the high speed and non-drive end of the intermediate stage is 15-20 hours. But preprocessing activities such as draining of entire gearbox oil, removal of the brake disc, coupling assembly and post processing activities such as assembly of coupling, brake disc, drive train alignment flushing of the gearbox and filling up of new oil will take minimum 2-3 days depending upon the wind condition, availability of power at site and etc.

ii) Replacement of bearing at drive end of an intermediate stage is not at all possible at the tower top (nacelle) due to the complication in the gearbox assembly. The entire intermediate stage has to be changed as a complete set with the help of huge capacity crane (400 tonne to 800 tonne).

iii) If any bearing fails at slow speed in the helical stage then the only one available solution is to go for gearbox exchange due to service constraint.
2.4.3 Shear Pin Failure

The shear pins are used in the coupling assembly of WTG to transmit the drive from the gearbox to the electric generator. The shear pin is a mechanical sacrificial component like an electric fuse designed to break itself as and when the mechanical overload arises to prevent the severe damage of the expensive components in the gearbox assembly and also in the electric generator. The failure of shear pin is an unavoidable occurrence in the WTG but frequent failure due to design mistake is a nuisance. The effect of shear pin failure in the wind turbine leads to stoppage of the turbine for 4-12 hours for replacing the failed shear pins. The frequent change of the shear pin during high wind season and also during present power crises time will lead to power generation loss and customer dissatisfaction.

2.4.4 Yaw and Pitch Drive Failure

Yaw drive fault, pitch drive fault and associated electrical faults also have an effect on system efficiency and even on breakdown. It was observed from field services that the major failures are due to uneven gap setting between the clutch and the brake pad in yaw and pitch drives. In general, the turbine will be equipped with 3-4 yaw drives for yaw the nacelle towards wind direction (maximum 360 degree) and 3 pitch drives for micro-pitching (maximum 3 degree) the blades during power production and to stop the turbine (maximum 90 degree) during air brake. If something goes wrong in one of the drive due to improper gap setting and mechanical overloading then it will directly reflect on the other drives and also to the motors resulting in either drive or motor failure. By monitoring the yaw and pitch motor current and pitch angle of the blades the drive failures can be avoided. At any case, the first and foremost failure component will be the motor due to excess current followed by heating. FFT spectrum analysis and
Machine Current Signature Analysis (MCSA) are used as fault detection systems in the wind industry.

2.4.5 Rotor Failure

Wind turbine rotors are prone to acquire creep and corrosion fatigue which can be observed as cracks and delaminations in the blades. Moreover dirt, ice, bird collisions, dampness manufacturing defects, assembly problem during erection and error during blade calibration may also make imbalance in the rotor blades and cause asymmetric aerodynamic. The roughness in the blades also leads to the loss in energy efficiency which could be caused by erosion, ice, dirt etc. Shearography, radiography, eddy current, strain gauges and thermography are the techniques that are employed by the manufacturer in different situation. In this research study gear failure, bearing and shear pin failure alone is considered as these parts are subjected to frequent failures during functioning of the WTG.

2.5 REVIEWS ON RESEARCH THEORIES

As described in section 2.2 the WTG gearbox is subjected to shock loads due to fluctuating wind force, sudden grid drops, non-synchronization of pitching and sudden braking. The helical gears used in the wind turbine gearboxes are manufactured with special care even in selection of the material, manufacturing, heat treatment and finishing process to minimize the noise level and other defects during the operation of the WTG. Though the WTG gearbox is designed for 20 years, because of the incorrect assembly and misalignment of the gears, overloads, inadvertent stress raisers or subsurface defects in critical areas of the helical gear, use of incorrect materials and heat treatment process, the gearbox fails within 5 years of service (Fernandes 1996). In general, each type of failure leaves characteristic clues on gear teeth, and detailed examination of the gear teeth often yields
enough information to establish the cause of failure. The general types of gear failure modes shall be i) Fatigue ii) Impact Fracture iii) Wear and Stress Rupture (Failure analysis and prevention 1986). Fatigue is the most common failure in gearing. Tooth bending fatigue and surface contact fatigue are the two types of the common modes of fatigue failure noticed in gears. Surface contact fatigue in gear teeth is due to excessive local hertzian. Surface contact fatigue stress plays an important role in the wind industry.

Generally, pitting and spalling are the two types of surface contact fatigue. The pitting of gear is characterized by occurrence of small pits on the contact surface. Pitting originates from small surface or subsurface as initial cracks, which grow under repeated contact loading. Pitting is a three-dimensional phenomenon and strongly depends on the contact surface finish of the gear profile, material microstructure and operating condition such as type of contact, loading, misalignment, lubrication problem and temperature. Spalling is not considered as an initial mode of failure but rather a continuous propagation of pitting and rolling contact fatigue. Although pitting appears as shallow craters at contact surfaces (Fatigue and fracture 1996), spalling appears as deeper cavities at contact surfaces (Ding 2003). Psman Asi (2005) made an attempt to investigate the root cause for the failure of the helical gear used in automobile gearbox and he concluded that the gear failed because of the fatigue with a fatigue crack initiation from destructive pitting and spalling region at one end of the tooth in the vicinity of the pitch line due to misalignment. Abhay and Diwakar (2002) investigated the failure of bronze gear used in mechanical transmission system through standard metallurgical technique. The study discovered that the corrosive wear at the root fillet caused pitting, where as intense localized plastic strain and folds leading to crack formation. Advancement of the crack took place under the successive stress repetitions to which the gear was subjected, causing the tooth to fail by fatigue. Holmes Edward and Schmidt Frederick (2006)
conducted a metallurgical analysis of the failure in the main landing gear of an aircraft. Kader et al (1998) has made an attempt for in-depth study of bending, pitting and scoring mode of failure of 20° and 25° pressure angle full depth spur gear.

Samroeng Netpu and Panya Srichandr (2011) have analysed the failure of herringbone gear used in a gearbox and concluded that the manufacturing defect is the root cause for the failure. The gear designer while designing a gear must consider different factors such as ratio, torque, speed, mounting dimensions, efficiency, noise and strength. The strength of the gear depends on the basic gear data say micro-geometry of the tooth like flank correction such as lead and profile crowning, tip relief and etc (Thompson et al 2000). According to Alban (1993) if the gear is kept in service despite the occurrence of spalling, a secondary crack may start from a spalling crater and propagate through the tooth thickness. As a consequence a part of the tooth may fall off and the failure surface appeared at a mid-height position on the tooth. Also, it was stated that the cracking can be subdivided into fatigue cracks and non-fatigue cracks. Non-fatigue cracks are typically hardened cracks, grinding crack, rim cracks and etc.

### 2.5.1 Literature on Spur Gear Root Modification

Gear drive is to transmit power with comparatively smaller dimensions of the driving system and to run reasonably free of noise and vibration with least manufacturing and maintenance cost (Derek Smith 2005). There is a growing need for higher load carrying capacity and increased fatigue life in the field of gear transmissions.

Spitas et al (2005) proved that the circular root fillet design is particularly suitable for gears with small number of teeth (pinion). Fredette and Brown (1997) have discussed the possibility of reducing
gear tooth root stresses by adding internal stress relief features. Michele Ciavarella and Giuseppe Demelio (1999) reported that the fatigue life is lower on gears with a lesser number of teeth. Hebbal et al (2009) have reduced the root fillet stresses in the spur gear using internal stress relieving feature of different shapes. Senthilvelan and Gnanamoorthy (2005) studied the effect of gear tooth fillet radius on the performance of injection moulded nylon 6/6 gears. Tae Hyong Chong and Jae Hyong Myong (2001) conducted a study to calculate simultaneously the optimum amounts of tooth profile modification for minimization of vibration and noise. Beghini et al (2004) proposed a simple method to reduce the transmission error for a given spur gear at the nominal torque by means of the profile modification parameters.

More recently numbers of authors have analyzed failure of gears for the applied bending forces. Moriwaki et al (1993) developed a technique named Global Local Finite Element Analysis (GLFEA) and applied it to a gear tooth for its stress analysis. In GLFEA no fine subdivision is required for the analysis and it is an easy method to find the critical section. In fact, GLFEA is a numerical analysis technique that combines finite element solutions and the classical analytical method on the basis of the energy principle.

Spitas et al (2005) suggested the idea of spur gear teeth with circular root fillet instead of the standard trochoidal root fillet and it was investigated numerically using BEM. The strength of the modified teeth were studied in comparison to the standard design by discretizing the tooth boundary using isoparametric boundary elements. In order to facilitate the analysis, the teeth were treated as non-dimensional assuming unitary loading normal to the profile at their Highest Point of Single Tooth Contact (HPSTC). It was demonstrated that the novel circular design surpasses the existing trochoidal design of the spur gear tooth fillet in terms of fatigue endurance.
without affecting the pitting resistance. The proposed geometry does not produce undercut teeth even for small number of teeth.

Beghini et al (2004) proposed a simple method to minimize the Peak to Peak Transmission Error (PPTE) for a spur gear set through parametric analysis using advanced software. The study was carried out to develop a simple method for profile optimization in terms of PPTE, which was the main cause of whining noise in spur gears. Costopoulos Th and Spitas (2005) have introduced one-sided involute asymmetric spur gear teeth to increase the load carrying capacity and to combine the meshing properties. Tesfahunegn and Rosa (2010) have investigated the influence of the shape of profile modifications on transmission error, root stress, and contact pressure through non-linear finite element approach. To improve the gear teeth strength, a lot of work has been carried out, but all mostly employed positive profile shifting. These contribution exhibits lower pitting and scoring resistance and lower contact ratio resulting in more noise and vibration during the power transmission (Niemann 1995).

Parker et al (2000) investigated the dynamic response of a spur gear pair using a finite element/contact mechanics model. The gear pair was analyzed across a wide range of operating speeds. The non-linearity source was the contact loss of the meshing teeth which occurred for large torques despite the use of high precision gears. The study used detailed contact analysis at each step as the gears rolled through the mesh to find the dynamic mesh forces. Hiremagalur et al (2004) studied the effect of backup ratio in spur gear root stresses analysis and design. Backup ratio was considered important in understanding the rim failures that start at the tooth root. An analytical approach, based on the theory of elasticity was used to provide a computational formulation for root stress calculations in spur gears.
According to Gitin Maitra (1998) if a gear is given undercut for one reason or another, it is necessary to know the magnitude of the undercutting radius. Ming-Jong Wang (2003) have investigated the Maximum Tensile Bending Stress (MTBS) and the critical point in the root fillet of spur gear tooth during transmission, by digital photoelastic system involving real time imaging. Math and Chand (2004) presented an approach for the determination of geometry of spur gear tooth root fillet. He has developed an equation to find the point of tangency of involute profile and root fillet on the base circle for a spur gear without undercutting and the point of intersection of root fillet and involute profile above the base circle for an undercut gear.

2.5.2 Literature on Helical Gear Profile Modification

Involute gears have been widely used because of their advantages such as ease of manufacturing and the fact that the gear speed is unchanged even if the gear centre distances changes (Henry Merritt 1979). Florin Tutulan et al (2004) formulated an optimum design method to determine the tooth root stress and the tooth contact stress of the involute cycloid composite tooth profile helical gear, since the involute gears have several disadvantages such as sliding on the teeth is greater, surface durability is lower, and undercut occurs more frequently in gears having a smaller number of teeth compared to other tooth profile gears. Tae Hyong Chong and Jae Hyong Myong (2001) developed a method to evaluate the optimum amount of tooth profile modification, end relief, and crowning by minimizing the vibration exciting force of helical gears as the meshing vibration of gear occurs not only due to the meshing impact and the change of mesh stiffness of gear tooth but also due to the manufacturing error and assembly error of the gear train.

The absolute and relative maximum transmission error variation increases with an increase in pressure angle. Thus, a higher-pressure angle gear tends to be more sensitive to pitting damage (Lin Liu 2002). The contact
stress and bending stress are the sources of failure in helical gear. Tooth surface wear mainly occurs near the dedendum and the amount of wear increases as the number of teeth in meshing increases (Yong Chen et al 2007). Andrzej and Jerzy (2006) made a comparative study to evaluate the root strength using International Organization for Standardization (ISO) and American Gear Manufacturing Association (AGMA) standards, and the results are verified using the finite-element technique with model development and simulations. Simon (1989) proposed a design method to find out the optimal tooth tip relief and crowning for spur and helical gears. Mao (2006) has done gear microgeometry modifications mathematically for the power train gear transmission using Python script interfaced with finite-element models.

Shan Chang Donald et al (2005) introduced tip relief and root relief to reduce the high contact stresses occurring at the root corners, as well as in the entering and exiting regions. Alexander et al (2003) presented a novel method for bending stress balance suggesting an approach to the tooth parameters tolerance and tooth profile definition. Increase in module results in reduction of tooth deflection and root stresses. Tip relief is provided for minimizing the contact stress and to enable smooth running of the gear pair. Composite profile design reduces bending stress, tooth deflection, and contact stresses in the helical teeth. According to Gitin Maitra (1998) the tooth strength is improved by proper selection of parameters such as pressure angle and rolling circle in the gear.

Yi-Cheng Chen and Chung-Biau Tsay (2002) investigated the contact stress and bending stress of a helical gear set with localized bearing contact by means of Finite Element Analysis (FEA). They proposed a helical gear set comprised of an involute pinion and double crowned gear.
Mathematical models of the complete tooth geometry of the pinion and the gear had been derived based on the theory of gearing.

### 2.5.3 Literature on Bearing Failure in WTG Gearbox

An unsatisfactory condition, out of specification or catastrophic is called failure and when a bearing surface spall reaches 10 square milli inches then it is called bearing failure (TIMKEN bearing manual). The rolling-element bearings play a crucial role for the proper functioning of the gearbox used in the wind turbine generator (WTG). As discussed in section 2.3 increasing reliability of wind turbine gearboxes remains a challenging task for many in the industry (Robb 2005). One of the key issues is the misalignment of bearings and gears (Alex Tylee 2010). The consistency of performance by bearing in special environments like corrosive, high temperature and power (load), high speed and high vacuum zone is very important. In wind industries, the 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. Many studies reveal that defects in bearings and bearings failure are the main reasons for gearbox failure in the wind turbine generator.

Harish Harani and Manish Verma (2009) have made an attempt to avoid unequal and non-uniform wear of elastomeric bearing used in marine propeller shaft. Hirani (2009) 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 Orhan et al (2006) conducted a case study through vibration monitoring and analysis to diagnose the defect in rolling element bearings. Sudhakar and Joel Cruz Paredes (2005) investigated defective bearings of a motor for possible mechanisms leading to their premature failure during manufacturing. It was determined that the
motor bearing failed mainly due to improper sintering process and the variation in chemical composition.

Sudhakar (2002) presented failure analysis of automobile bimetal bearings which failed during storage before final assembly. Lliev (1999) analyzed the failure of hydro-generator thrust bearing and concluded that the cause for the failure is misalignment of bearing to the axis of rotation. It is noticed from the field study since 2008 that the major bearing failure in wind industry are contact fatigue due to excessive load and lubrication, wear due to contamination in the oil, vibration and looseness. The same have been discussed in chapter 4.5 in detail. The typical bearing failure modes and the probable causes described by John Judd (2006) are presented in Table 2.2.

**Table 2.2  Bearing failure modes and probable causes**

<table>
<thead>
<tr>
<th>FATIQUE</th>
<th>PROBABLE CAUSE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spalling – subsurface fatigue</td>
<td>Excessive load</td>
</tr>
<tr>
<td>Pitting – surface fatigue</td>
<td>Lubrication</td>
</tr>
<tr>
<td><strong>WEAR</strong></td>
<td></td>
</tr>
<tr>
<td>Fretting / Surface corrosion</td>
<td>Vibration / Looseness</td>
</tr>
<tr>
<td>Abrasion</td>
<td>Contamination</td>
</tr>
<tr>
<td>Scoring / Abrasion</td>
<td>Defective seals</td>
</tr>
<tr>
<td>Corrosion</td>
<td>Defective seals</td>
</tr>
<tr>
<td>Brinelling</td>
<td>Excessive vibration</td>
</tr>
<tr>
<td>Localized fretting / hardening</td>
<td>Excessive vibration</td>
</tr>
<tr>
<td>Smearing / Pitting / Fluting</td>
<td>Electric discharge</td>
</tr>
<tr>
<td><strong>PLASTIC FLOW / FRACTURE</strong></td>
<td></td>
</tr>
<tr>
<td>Brinnelling / Denting</td>
<td>Excessive or Point load</td>
</tr>
<tr>
<td>Material failure</td>
<td>Hot / Cold working / Latent Defect</td>
</tr>
</tbody>
</table>
The rolling bearing cannot rotate forever. Unless or otherwise operating conditions are ideal and the fatigue load limit is not reached, sooner or later material fatigue will occur. The period until the first sign of fatigue appears is a function of the number of revolutions performed by the bearing and the magnitude of the load. Fatigue is the result of shear stresses cyclically appearing immediately below the load carrying surface. After some time these stresses cause cracks which gradually extend up to the surface. As the rolling elements pass over the cracks, fragments of material break away and this is known as flaking or spalling. The flaking progressively increases to an extent and eventually makes the bearing unserviceable. The life of a rolling bearing is defined as the number of revolutions the bearing can perform before incipient flaking occurs. This does not mean to say that the bearing cannot be used after that. Flaking is a relatively long, drawn-out process and makes its presence known by increasing noise and vibration levels in the bearing. Therefore, as a rule there is plenty of time to prepare for a change of bearing (SKF Product Information 401). Further, the durability analysis of the bearings can be conducted using 4 different methods. These methods are already summarized in section 2.3 briefly.

2.5.4 Literature on Shear Pin Failure

Many researchers in the field of wind energy have focused their studies on the development of advanced materials, new heat treatment methods, designing larger size couplings to tackle the problem of shear pins failure due to mechanical overload. Abhay Jha et al (2008) studied the role of inclusion in the fatigue performance of a shear pin used in chemical mixer fabricated using EN 19 material. Azevedo et al (2009) carried out investigation and reported that the stable crack propagation due to fatigue caused by bi-directional bending was nucleated at machining marks under
normal load. Smith et al (2007) suggested that misalignment during shear pin assembly associated with vibration might have promoted the pre-mature failure of the shear pin by fatigue. Goksenli and Eryurek (2009) have investigated the reason for elevator drive shaft failure by performing stress analysis using finite element method and also determined the permissible force and torque acting on the shaft.

Bhaumik et al (2002) analyzed the failure of a hollow power transmission shaft and recommended for failure prevention. Many research papers have been published on failure analysis of Kaplan turbine (Luo Yongyao et al 2010), Steam generators (Cicero et al 2010), Elevator drive shaft and air compressor (Siva Prasad et al 2010); but no one have made an attempt to analyze the reasons for the failure of shear pins employed in the wind turbine generator to the best knowledge of the research scholar. Muhammad et al (2009) investigated failure analysis of high speed pinion shaft through microstructure analysis, electron micrography method and proved that the failure was because of fatigue. Stevenson (2005) investigated an increase in service stresses after bearing replacement along with the presence of significant nonmetallic inclusion in the pinion shaft are the root cause for failure of a high speed pinion shaft used in marine diesel engine. Darryl and Tony (2005) investigated the failure of generator shaft keyway crack via metallurgical failure analysis, material testing, stress and fracture mechanics analysis. Zhong Su (2001) performed a comprehensive investigation including dynamic analyses of the piston rod to identify the design problem. Poursaeidi et al (2009) has recommended while analyzing the performance of a lock-pin in 3D finite element model that inadequate design and long service reduced the performance of lock-pin for sustaining a severe hot condition.
2.6 SUMMARY

This literature review covered a variety of topics, techniques, methods, and approaches. The literature was basically categorized into three major themes; basic concept of the WTG gearbox, types of faults encountered in the wind industry, and the practical use of various methods for investigating the wind turbine gearbox and its associated components failure. The review presented in this chapter indicates that previously proposed methods of WTG gearbox failure analysis still remains as an unexplored area. The usage of wind turbine is rapidly increasing all over the world. Therefore, the demand for the failure analysis in wind industry is both critical and ever increasing. The published articles by Beghini et al (2004), Spitas et al (2005), Florin Tutulan et al (2004), Robb (2005), Smith (2007) have tempted the scholar to do research in WTG for preventing frequent breakdown in wind turbines.