Final Technical Report

Supporting Wind Turbine Research and Testing - Gearbox Durability Study

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1 EXECUTIVE SUMMARY

The combination of premature failure of wind turbine gearboxes and the downtime caused by those failures leads to an increase in the cost of electricity produced by the wind. There is a need for guidance to asset managers regarding how to maximize the longevity of their gearboxes in order to help keep the cost of wind energy as low as possible. A low cost of energy supports the US Department of Energy's goal of achieving 20% of the electricity in the United States produced by wind by the year 2030. DNV KEMA has leveraged our unique position in the industry as an independent third party engineering organization to study the problem of gearbox health management and develop guidance to project operators. This report describes the study.

The study was conducted in four tasks. In Task 1, data that may be related to gearbox health and are normally available to wind project operators were collected for analysis. Task 2 took a more in-depth look at a small number of gearboxes to gain insight in to relevant failure modes. Task 3 brought together the previous tasks by evaluating the available data in an effort to identify data that could provide early indications of impending gearbox failure. Last, the observations from the work were collected to develop recommendations regarding gearbox health management.

Key findings from the research include: the ability to see long term changes in turbine performance with careful examination of SCADA data, the effects of misalignment on high speed bearings, and the usefulness of individual data streams available to project managers as indicators of gearbox health. We conclude that no single source of data can be used to accurately predict failure. This leads us to conclude that an integrated approach to gearbox health management should be established, where multiple data sources are assembled to assess gearbox health.

Through the process of investigating the data associated with gearbox failures, our industry interactions, and reviews of current standards and literature, DNV KEMA has collected information that supports development of recommendations regarding gearbox health management. These recommendations for wind turbine gearbox health management are included in Appendix A.

2 INTRODUCTION AND BACKGROUND

Of all wind turbine subassemblies, gearbox failures cause the most turbine down time (1). This down time, and the high costs of the labor, parts, and equipment needed to replace a gearbox, result in high costs for gearbox failures. This project sought to reduce the cost of wind-generated energy by researching the correlation between operational conditions and gearbox condition and the gathering and disseminating of operations practices that will maximize gearbox life.

The specific objectives of the project were:

1. To develop correlations between a wind turbine gearbox condition and the data generally available to wind project owners and operators (operational data).
2. To develop a framework for recommended practices for wind turbine gearbox health monitoring.
3. To support the National Renewable Energy Laboratory Gearbox Reliability Collaborative (NREL GRC) project.

DNV KEMA's approach to achieving the project objectives is outlined by task below.

Task 1: Supervisory control and data acquisition (SCADA) data collection and analysis

The goal of Task 1 was to collect operational data from operating turbines, including some turbines that experienced gearbox failure. These data were encoded and examined for broad patterns and correlations between parameters related to gearbox health and turbine operational parameters, turbine and gearbox types, and maintenance practices.

Task 2: Detailed gearbox health data collection

The goal of Task 2 was to study a smaller number of turbines in detail by instrumenting specific turbines to collect condition monitoring data from gearboxes. These detailed data allow closer examination of correlation between operations and incipient failures in the turbine gearboxes.

Task 3: Establish correlations between operational parameters and gearbox health

The detailed data collected in Task 2 were evaluated to identify correlations between incipient failure and standard SCADA channels.

Task 4: Develop a framework for recommended practices for gearbox health monitoring

In Task 4 the information learned in earlier tasks was consolidated with other industry knowledge to create recommendations for gearbox health management. The recommendations are intended to provide wind project asset managers actions that can be taken to maximize the life of gearboxes currently in operation.

Non-technical tasks such as project management and reporting were also a part of the work.

3 RESULTS AND DISCUSSION

The results of the technical work conducted during this study are presented and discussed here by task.

3.1 Task 1. SCADA Data Collection

3.1.1 Introduction

The objective of Task 1 was to collect operational data from operating turbines, including some turbines that experienced gearbox failure (as “failure” is defined in Appendix B). The approach to meeting this objective was to identify and select data partners, develop data requests, and send the data requests to selected data partners. DNV KEMA has existing business relationships with wind turbine owners and operators and maintenance providers; multiple organizations were contacted and three were selected to supply data for this study (“Data Partners”).

3.1.2 Data Request Development

In order to guide the creation of meaningful and easily-completed data requests, DNV KEMA identified, ranked, and organized parameters likely to enable detection of an influence on gearbox health. Table 1 lists these parameters.

Table 1. Data Parameters

(1 = high priority, 5 = low priority)

Data Parameter	Priority
Turbine manufacturer and type (categorized and encoded)	1
Gearbox manufacturer and type (categorized and encoded)	1
Gearbox configuration (0 = 3-point, 1 = double bearing, etc.)	1
Gearbox replacement periods by individual turbines having experienced failures (failure history)	1
Turbine locations (coordinates of turbines)	1
Date of oil changes	1
Oil type (categorized)	1
SCADA temperature records around gearbox	1
Gearbox failure mode	1
Water content in the oil (Karl Fischer method C)	1
Gearbox oil filtration type and size	1
Number of crow-bar ¹ events	1
Loading factors on gears and bearings	1
Temperatures	1
Peak torque	1
Load at yaw deviation	1
Oil cleanliness per ISO 4406	1

¹ Crow-bar events are activations of a circuit (the “crow bar”) in turbines with doubly-fed induction generators to protect the power electronics from high currents during low voltage grid faults. These events generate torque spikes in the drive train that cause high loads on gears and bearings.

Data Parameter	Priority
Gearbox conditions where estimated by inspection or detailed monitoring (ranked pristine = 0, 1, 2, awaiting replacement = 3, add 4 for each replacement in the history)	2
Energy produced per gearbox failure	2
Site mean shear	2
Site median maximum diurnal shear	2
Site median minimum diurnal shear	2
Site wind direction short term variation	2
Site wind direction long term variation	2
Site TI	2
Maximum gust	2
Maintenance records	2
Mean time off-yaw	2
Extreme direction change events	2
Alignment of generator and gearbox	2
Long-term mean capacity factor	3
Maximum monthly annual capacity factor	3
Number of emergency stops on gearbox.	3
Icing events	3
Mass imbalance	3
Design limit for off-yaw operation	3
Site atmospheric salinity	4
Number of normal starts and stops on gearbox	4
Aero imbalance	4
Vibration faults	4
Pitch faults	4
Type class of turbine (10 = IIA, 9 = IIB, etc.)	5
Class of site (10 = IIA, 9 = IIB, etc.)	5
Maintenance organization	5
Summary history of SCADA data collected during project history	5
Torque histogram	5
Design of support structures for gearbox and generator	5
Bedplate stiffness	5
Weight of generator	5
Weight of gearbox	5
Accuracy of tower top machining	5
Qualitative information on overall project health	5
Calibration data for nacelle and met tower anemometry	5

After creating the list shown in Table 1, discussions with wind project operators were held in order to refine which information would be requested. From the identification of parameters and prioritization and the discussions with Data Partners, DNV KEMA selected a subset of parameters that were practical to request from Data Partners, and requested the data.

3.1.3 Data Provided

SCADA data were made available to DNV KEMA from three Data Partners for the purpose of identifying trends which may indicate imminent or early stages of drive train failure. These data are representative of what would be currently available to wind project operators without more sophisticated vibration or other

condition monitoring tools. While the sensor types available varied across projects and turbine models investigated, the following channels were available in all cases:

- Gearbox oil temperature
- High speed bearing temperature
- Ambient temperature
- Active power produced
- Nacelle wind speed
- Generator and/or rotor rpm

Additional channels were available from some Data Partners and utilized when available. Data were available as 10-minute average values; and in some cases 10-minute minimums, maximums, and standard deviations were also provided.

The three Data Partners, called A, B, and C in this report, provided information to support the study. Turbine numbers in this report are preceded by the Data Partner number, for example, A-1 is the first turbine from Data Partner A. The contributions from these three Data Partners are discussed below.

3.1.3.1 Data Partner A

Data Partner A provided SCADA data for three adjacent turbines, described as turbines A-1, A-2, and A-3 in this report. The data are 10-minute records from January 6, 2007 through October 29, 2011 (4.6 years). The following measurements were provided:

- Wind speed (average, maximum, minimum, and standard deviation for each 10-minute period)
- Generator RPM (average)
- Power (average, maximum, minimum, and standard deviation)
- Ambient temperature (average)
- Gear bearing temperature (average)
- Gear oil temperature (average)
- Active alarm
- First alarm

Data Partner A provided a data set containing total fault counts in roughly 190 fault categories for all turbines at one wind power project. These data were combined with gearbox condition data, described later, for a fault analysis in Task 3.

3.1.3.2 Data Partner B

Data Partner B provided SCADA data from two sets of three adjacent turbines (six turbines in total). Turbines are identified as turbines B-1 through B-6. The data from turbines B-1, B-2, and B-3 are 10-minute records from January 1, 2006 – January 1, 2010 (4 years of data). The data from turbines B-4, B-5, and B-6 were from January 1, 2009 – July 1, 2011 (2.5 years of data). The following measurements were provided:

- Wind speed (average, maximum, minimum, and standard deviation for each 10-minute period)
- Wind direction (average)
- Nacelle direction (average)
- Nacelle position/wind direction difference (average)
- Rotor RPM (average)
- Power (average, maximum, minimum, and standard deviation)
- Ambient temperature (average)

- Nacelle temperature (average)
- Gear bearing temperature (average)
- Gear oil temperature (average)

3.1.3.3 Data Partner C

Data Partner C provided SCADA data from three adjacent turbines, designated as turbines C-1, C-2, and C-3. Data Partner C provided 10-minute records that span the time from January 1, 2006 – September 26, 2011 (5.7 years of data). The following measurements were provided:

- Wind speed (average)
- Wind direction (average)
- Nacelle direction (average)
- Generator RPM (average)
- Power (average)
- Ambient temperature (average)
- Gear bearing temperature (average)
- Gear oil temperature (average)
- Seconds turbine “ok” in 10-minute record (count)

3.1.4 Summary of Task 1

Summarizing Task 1, DNV KEMA collected SCADA data from three Data Partners. These data included operational parameters that DNV KEMA had prioritized based on likelihood that the parameters would correlate to gearbox health. Gearbox failure data were also provided.

One of the challenges encountered during the execution of the project was collection of data from a broad range of Data Partners. Far less data than originally anticipated were collected. There is a general perception within the U.S. wind industry that operational data need to be protected as intellectual property. In DNV KEMA’s opinion, the industry can advance more rapidly if greater transparency with operating data is achieved. DNV KEMA was able to use the information collected to reach the conclusions discussed in Task 3.

3.2 Task 2: Detailed Gearbox Health Data Collection

3.2.1 Introduction

DNV KEMA collected detailed gearbox health information through borescope (visual) inspection of multiple gearboxes, vibration measurements, and oil analysis results. This detailed data collection also enabled collection of failure data in addition to the failure data collected as a part of Task 1. A subset of the information collected in Task 2 helped to establish the actual condition of the gearboxes so that the health of the gearboxes could be correlated to Task 1 SCADA data during the analysis performed in Task 3. In addition, the multiple types of detailed data collected enable a comparison of the basic methods of gearbox health assessment. This section of the report describes the detailed data collected.

The borescope data and vibration data were collected by DNV KEMA engineers at Data Partner wind projects. DNV KEMA inspected seven gearboxes at a wind project operated by Data Partner A and one gearbox at a wind project operated by Data Partner B, and none of the gearboxes from Data Partner C. These inspections are discussed below.

3.2.2 Data Partner A

3.2.2.1 Data Partner A - Gearbox Failure Data

Data Partner A provided gearbox failure information for all turbines at one Data Partner A site. This included details on type and location of failures in the gearbox. These turbines are not numbered in this report, but we discuss some of the characteristics of this failure data set in Task 3 below and used these data for a fault analysis.

The gearbox in turbine A-2 failed in September 2009 with damage to the high-speed pinion. Data Partner A provided a vibration report from the turbine manufacturer which had been used to identify the failure in turbine A-2; this is discussed further in section 3.2.2.3 below.

3.2.2.2 Data Partner A - Visual Inspections via Borescope

In early August 2010, DNV KEMA traveled to a site operated by Data Partner A to collect detailed gearbox health data. Data Partner A had previously determined that some of the gearboxes at their site contained incipient failure modes. The gearboxes selected for inspection were selected from the gearboxes with known incipient failure modes. DNV KEMA inspected the gearboxes with a borescope, and found components with varying levels of failure. The severity of each failure was categorized as follows:

1. Normal wear for age.
2. Unusual wear observed; indeterminate failure state.
3. Abnormal condition for age.
4. Critical condition. Continued operation in current state may result in catastrophic failure of gearbox or significant collateral damage.

The failure modes, failure location within the gearbox, and failure severity, are summarized in Table 2 below.

Table 2. Data from Borescope Inspections of Data Partner A Gearboxes

Turbine	Failure Mode Description and Location	Category
A-10	Ring gear Hertzian fatigue, tooth root (dedendum) micropitting and tooth flank (addendum) macropitting	3
A-11	Ring gear Hertzian fatigue, tooth root micropitting and tooth flank macropitting	2
A-12	Ring gear Hertzian fatigue, tooth root micropitting and tooth flank macropitting	2
A-13	None	1
A-14	Bearing high cycle fatigue, high-speed bearing inner race cracks	3
A-15	Bearing Hertzian fatigue, planet bearing roller end wear	2
A-16	Ring and Sun gear Hertzian fatigue, micro and macropitting; high-speed bearing roller Hertzian fatigue, macropitting	3

Examples of each failure severity category are shown in Figure 1 and Figure 2 below.

Figure 1. Gearbox Failure Indications (from left to right: turbine A-10, A-11, and A-12)

Figure 2. Gearbox Failure Indications (from left to right: turbine A-15, A-16, and A-16)

According to Data Partner A, the most frequent gearbox failure mode experienced on their site is cracking of the inner race of high-speed bearings. This failure mode is shown in Figure 3, in an image taken from a Data Partner A gearbox. This failure mode is widespread in the industry, and has received considerable attention in recent research efforts (2). Because of the prevalence of this failure mode, we focus on this failure mode in our misalignment study in Task 3.

Figure 3. High-Speed Bearing Inner Race Crack

In addition to the borescope inspections performed by DNV KEMA, Data Partner A provided borescope inspection data for turbines in their fleet corresponding to the oil sample analysis reports discussed below.

3.2.2.3 Data Partner A - Vibration Measurements

Measurements of the vibrations of gearboxes can provide additional detailed information about the health of the gears and bearings. Turbines A-13 and A-14 were selected for vibration data collection. Turbine A-14 was selected because of the severity of the failure found on turbine A-14 during borescope inspections and the frequency at which the Data Partner reported that this particular failure mode is occurring on the site. Turbine A-13 was selected because it was not found to be damaged in the borescope inspection, and therefore is a control (undamaged) case.

During the August 2010 visit to the site operated by Data Partner A, two vibration data collection systems were employed for monitoring the two selected turbines. Accelerometers were placed over the upwind and downwind high-speed shaft bearings, and data were sampled at 1-minute intervals for 24 hours. The analysis of these data is discussed in Task 3.

In addition to the DNV KEMA-collected data, Data Partner A provided a vibration report for turbine A-2 from the turbine manufacturer.

Concurrent with the vibration measurements, 10-second SCADA data were collected for turbines A-13 and A-14; those data are discussed further in the analysis provided in Task 3.

3.2.2.4 Data Partner A – Oil Sampling and Analysis

Analysis of oil samples provides information about the status of the machine lubricant, which may be correlated with machine condition. Oil analysis information was provided by Data Partner A for 199 oil samples from gearboxes throughout their fleet of turbines. These data span the time period from February 3, 2010 through March 25, 2011. Analysis of the oil sample information is provided in Task 3.

3.2.3 Data Partner B

3.2.3.1 Data Partner B – Gearbox Failure Data

The gearboxes in Data Partner B turbines experienced failure on the dates listed in Table 3.

Table 3. Gearbox Failure Dates and Failure Modes from Data Partner B

Turbine	Date of Failure	Failure Mode
B-1	February 2, 2009	Splash ring contact with gears
B-2	September 15, 2009	High speed shaft (HSS) bearing, rotor side
B-3	September 14, 2010	HSS bearing, rotor side
B-4	January 8, 2011	HSS bearing, rotor side
B-5	August 20, 2010	HSS bearing, rotor side
B-6	September 8, 2010	HSS bearing, rotor side and intermediate shaft bearing, rotor side

3.2.3.2 Data Partner B – Visual Inspections via Borescope

On May 12, 2011, DNV KEMA inspected the gearbox in one turbine (identified in this report as turbine B-7) on a Data Partner B wind project with a borescope (Figure 4). The inspection discovered extensive damage to the inner race of the rotor-side high speed shaft bearing (Figure 5).

Figure 4. DNV KEMA Borescope Inspecting Gearbox in Turbine B-7

Figure 5. Borescope Image of Cracked Bearing Race in Turbine B-7

The failure mode is a fatigue fracture of the inner race, at multiple locations on the inner race. There is also cracking of the race surface, with spalling at multiple locations. Corrosion spotting was detected on the rollers, and there was some roller end scoring. The bearing was in an advanced stage of failure.

3.2.3.3 Data Partner B – Vibration Measurements

At the same time that the borescope inspection was performed, DNV KEMA installed vibration measurement equipment turbine B-7. Eight vibration sensors were placed on the gearbox, main bearing, and generator bearings. Data were collected for approximately four days before the equipment was removed. As with the Data Partner A vibration information, these data were of interest because they were collected from a turbine with a known failure mode and a known severity level. Analysis of these data is discussed in Task 3 below.

3.2.3.4 Data Partner B - Oil Sampling and Analysis

The gearbox inspected by DNV KEMA at the Data Partner B site has both online and offline filtration systems; the filters were not examined as a part of the DNV KEMA site visit. An oil sample was collected from the gearbox in the turbine (Figure 6). The oil sample was collected at the oil distribution manifold, and is a pre-filter sample.

Figure 6. Collecting an Oil Sample

Discussion of the oil sample analysis from Data Partner B is presented in Task 3.

Data Partner B also provided a spreadsheet containing oil sample data from 2006 to 2010 in 6-month intervals for turbines on another wind project.

3.2.3.5 Data Partner C – Gearbox Failure Data

Data Partner C provided gearbox failure data, as shown in Table 4 below.

Table 4. Gearbox Failure Dates and Failure Modes from Data Partner C

Turbine	Date of Failure	Failure Mode
C-1	February 2, 2009	Case fracture
C-2	September 14, 2009	High-speed stage
C-3	September 14, 2010	Intermediate bearing, rotor side

3.2.4 Summary of Task 2

In Task 2, detailed data collection allowed exploration of the basic methods of assessment of gearboxes (visual inspection, vibration analysis, and oil sampling). The data collected provided reference information for the analyses conducted in Task 3.

3.3 Task 3: Establish Correlations between Operational Parameters and Gearbox Health

In Task 1, a broad set of data were collected from Data Partners, and in Task 2, detailed gearbox health information was collected. For Task 3, we sought to establish correlations between operational parameters

and gearbox health. To establish the correlations, we examined time series plots and cumulative trends, analyzed turbine faults, and compared the data against known failures, as described by project operators. These analyses are discussed in detail below.

3.3.1 Analysis of SCADA Data

3.3.1.1 Approach

DNV KEMA compared multiple years of data (collected in Task 1) from each Data Partner (A, B, and C) from turbines with known drive train failures with two adjacent turbines which did not experience failures, or experienced failures at later times. Comparing adjacent turbines allowed for an assumption of similar inflow conditions. In these comparisons, turbines with identical drive train models were also selected. This approach allowed for a more thorough investigation of a few turbines over a long period of time, typically two to six years. To the extent possible, the same evaluations were made on each of the data sets made available by each of the Data Partners. This section provides highlights from the data analysis.

In addition to investigating the channels directly available from the SCADA system, DNV KEMA also produced calculated channels to determine if combinations of channels might illustrate conditions which precede failures. Examples of calculated channels include:

- Rise in bearing temperature over gear oil temperature (both the ratio of, and difference between, these two temperatures were investigated)
- Rise in bearing and oil temperature over ambient temperature (again, both ratio and difference were investigated)
- Coefficient of Performance (C_p), as defined in IEC 61400-12
- Difference between wind direction and nacelle position (yaw error)
- Daily and monthly averaging of channels

The comparison was performed by conducting a series of tests with the data and analyzing the results. The tests and the results are described in detail below.

3.3.2 Tests Conducted

3.3.2.1 Time Series Plots

Time series plots of similar channels were plotted to determine if any clear trends or abnormal values were present prior to failures. For example, bearing temperatures of three adjacent turbines were plotted to see if higher temperatures were experienced in the turbine where the failure occurred. In this section, the time of failure is defined as the time when the gearbox goes offline to service the failure.

In one case the difference between the bearing temperature and oil temperature was observed over time and the bearing was consistently 5 to 15 °C hotter than the oil temperature, as a daily average. The signature of this temperature relationship changed noticeably after a case fracture occurred in the original gearbox and it was replaced. Figure 7 shows the difference between bearing and oil temperature for a single turbine over time and the change that occurred when the gearbox was changed. The new temperature signature also varied significantly from that seen in neighboring turbines, which had a bearing minus oil temperature operational window of 5 to 15 °C, similar to the range observed in the original gearbox. Figure 8 compares the bearing temperature verses power of the failed turbine (center) to two adjacent turbines which did not experience failures and further illustrates the change in nominal operational conditions.

Figure 7. Example of Temperature Signature Change in Time Series Data (5.7-year period in total) for Turbine B-2. Offline Date and Gearbox Change Indicated by Black Line.

Figure 8. Example of Temperature Signature Change

2-year period shown; data in center plot from Turbine B-2 with a failed gearbox; data in green are after the gearbox was replaced

A different make of gearbox was used when the gearbox was replaced after the failure, and the different make of gearbox has a cylindrical roller bearing under the temperature sensor instead of a tapered roller bearing. This is likely to be the cause of the change in nominal operating conditions. While in this case the sudden change in temperature does not provide an early indication of the gearbox failure, it does illustrate the variation in data that occur when a major component is changed. If this signature were being monitored for changes, it would be necessary to establish new acceptable thresholds for the period after the gearbox failure and replacement.

In another situation where the high speed bearing failed, DNV KEMA investigated the temperature measured at the high speed bearing and that of two adjacent turbines which did not experience bearing failures. Surprisingly, no significant variation in bearing temperature is seen between the turbine with the failed bearing and neighboring turbines. Oil temperature was also pursued as a possible indicator of failure, but generally correlates with bearing temperature and exhibited even less variation. Figure 9 shows the time series data for bearing temperature in the 26 days preceding the bearing failure (which

occurred in Turbine B-2). All bearing temperatures track closely and none exceed the expected temperature range during this period. This indicates that monitoring bearing temperature is not always an effective means for identifying failed bearings.

Figure 9. 10-Minute Average Bearing Temperature Preceding Turbine Offline Date (indicated by black line on 10/11)

Approximately six months before the turbine went offline to service the failed gearbox, turbine B-2 did exhibit a 1-month period where its bearing temperature was periodically 5-10 °C greater than neighboring turbines (Figure 10). The reason for this is not clear, and other channels indicate the turbine was functioning normally. Before and after this period the bearing temperature did not deviate significantly from neighboring turbines. Additionally, turbine B-1, which did not suffer a bearing failure, experienced similar periodic bearing temperature deviations. Therefore it is not possible to confirm that the bearings on turbine B-2 were damaged during these high temperature periods.

Figure 10. 10-Minute Average Bearing Temperature with Abnormal Peak Temperatures

3.3.2.2 Cumulative Trends

Histograms were generated to investigate if certain cumulative events experienced by turbines were more frequent or occurred in more extreme conditions in turbines with drive train failures than in turbines without. Specific questions addressed include:

- Do the number of startups a turbine experiences lead to greater failures and/or do turbines that experience failures have a history of more frequent startups?
- Do sudden ramp-ups in power have a similar effect?
- Does the amount of time stopped have a similar effect?
- Does ambient temperature have an added effect for any of these cycling events?
- Does the wind speed the events occurred at have an effect?
- What oil temperature trends occur prior to failures?

To answer these questions, DNV KEMA counted the number of startups, ramp-ups, and stopped time events. Cumulative counts of these events were compared against the temperature they occurred at and, separately, the wind speed they occurred at. A “startup” was defined as any event where the generator transitioned from less than nominal operating rpm (e.g., 1800 rpm) to over nominal rpm between 10-minute periods. A “ramp-up” was defined as any event where the average power increased by 25% of the rated power or more between 10-minute periods. A “stopped” event was defined as any 10-minute record where the generator or rotor rpm was equal to 0 (i.e., not pin-wheeling).

The plots below present the cumulative number of startup, ramp-up, and stopped events from the beginning of the data set until the time of gearbox failure for one site, a period of 3.5 years. In this example turbine B-2 experienced a high-speed bearing failure. Each plot contains a box with the total number of counts of each event type for each turbine, in addition to the plot which indicate the temperature or wind speed the events occurred at. If any of these event types were significant contributors to the failure, we would expect to see a higher count of events and/or some deviation in the wind or temperature conditions during events for turbine B-2.

Figure 11. Cumulative Startup Events and Ambient Temperature, C°

Figure 12. Cumulative Startup Events and Nacelle Wind Speed, m/s

Figure 13. Cumulative Ramp-up Events and Ambient Temperature, C°

Figure 14. Cumulative Ramp-up Events and Nacelle Wind Speed, m/s

Figure 15. Cumulative Stopped Events and Ambient Temperature, C°

Figure 16. Cumulative Stopped Events and Nacelle Wind Speed, m/s

The data presented in Figure 11 through Figure 16 indicate that turbine B-2 did not experience any unusual number of startups, ramp-ups, or stopped periods, or an unusual number of these events at extreme temperatures or wind speeds. In all cases the total number of events and conditions under which the events took place were not significantly different for the turbine which experienced the failure and those without failures. Similar observations were made in the SCADA data of other projects. It is therefore considered unlikely that tracking these events would be useful in predicting gearbox failures.

Total operating hours and cumulative energy production were also investigated for correlations with gearbox failures. No correlation with either energy production or operating hours was seen at any of the sites. Figure 17 shows an example of energy production plotted over time from Data Partner A with two failures noted. While the rate of production is slightly higher for turbine A-2, the difference with the neighboring turbines is not significant and failures did not occur in neighboring turbines once they reached or exceeded the production where turbine A-2 failed. Turbine A-1 experienced some type of failure in 2009, but it was not confirmed this was related to the drive train.

Figure 17. Cumulative Energy Production for Three Turbines (failures noted by arrows)

Some operators discussed concern that rolling elements within bearings could be periodically sliding over the races, potentially initiating damage that may eventually propagate to failure. If sliding were to cause the temperature of the bearing to rise significantly and suddenly, there may be a reflection of this in the SCADA data. To investigate this possibility, the difference in average bearing temperature between 10-minute records was calculated for three turbines from Data Partner B (B-1, B-2, and B-3) where Turbine B-2 experienced a bearing failure. Histograms were generated to compare the count of occurrences in each 1°C bin. Figure 18 shows a comparison of these histograms.

Figure 18. Cumulative Count of Bearing Temperature Difference, Site B

The histograms are heavily weighted around a temperature difference of 0°C, which is expected if the turbine is in steady operation where the temperatures would fluctuate slowly between 10-minute periods. There are few occurrences where the temperature change exceeds more than 15°C and the turbine with the failed bearing showed no significant deviation in the number of occurrences of high temperature difference events. In this case, observing the variation in bearing temperature between 10-minute records did not predict the bearing failure.

The 10-minute timescale of SCADA data may not be fine enough resolution to capture temperature rise due to bearing sliding. Statistics on the 10-minute data, however, may be used to detect this temperature rise. Standard deviation, minimum, and maximum of generator speed provide a measure of the acceleration and deceleration of the drive train. This acceleration and deceleration may lead to sliding contact (rather than rolling contact) in large bearings such as those in wind turbine gearboxes, which can be damaging to bearing races. Standard deviation, minimum, or maximum data for drive train speed were not available in the data sets collected; however, we recommend analyses of these statistics as further research work.

Previous work, such as (3), has shown observable trends in gearbox oil temperature rise in the time period before a planetary gear failure. DNV KEMA completed a similar analysis of bearing temperature rise (bearing temperature minus ambient temperature) for turbine B-2 which experienced a bearing failure. Three periods of data, each three months in length, were examined to see if the bearing temperature rise showed any appreciable change in the period three months before the failure, when compared to the periods six and nine months before the failure. Figure 19 shows histograms of the bearing temperature rise for the three periods.

Figure 19. Histograms of Gearbox Bearing Temperature Rise for Three Periods, Turbine B-2

This analysis shows a counter intuitive result, with the temperature rise actually greater in the period nine months before the failure than in the period three months before the failure. Analysis of the time series data from the same turbine showed a period of abnormally high bearing temperatures a little more than six months before the failure, as described earlier; this is captured in the significantly greater number of occurrences of temperature rise greater than 60°C in the nine month pre-failure period. The analysis of the histograms also demonstrates that the method described in (3) was not sensitive enough to capture a bearing failure; the failure mode in that paper was described as a “planetary gear failure.”

3.3.2.3 Variation in Operational Patterns

DNV KEMA generated scatter plots comparing multiple channels, looking for deviations between turbines and variation over time which might correlate with gearbox failures. Scatter plots generated included:

- Power versus wind speed (power curve)
- Power versus generator speed
- Temperature (oil and bearing) versus power
- Bearing temperature versus oil temperature

Comparing plots of oil temperatures versus power revealed increasing operating temperatures over time in some turbines. Figure 20 shows an example of oil temperature versus power from three years for a single turbine. In this plot, the maximum oil temperatures observed at or near rated power have increased by more than 10°C between 2007 and 2011. Additionally, the oil temperature at lower power levels has increased with time.

Figure 20. Oil Temperature versus Power Produced from 2007 to 2011

The reason for the increasing temperature is not known, but could be caused by several factors or combination of factors (e.g., a fouled heat exchanger, oil depletion, loss of oil, increased friction in drive train, etc.). In any case, the increasing oil temperatures do not seem to correlate with drive train failures, as all turbines investigated from this particular project show a similar trend but not all turbines experienced failures.

Examination of power curves did not show any correlation between gearbox failures and changes in the power curve; however, DNV KEMA did observe degradation in several cases which may be of interest to operators. The power curve scatter plots were filtered and binned to produce average annual curves and were compared from year to year. In the most extreme case, one turbine showed a 9% decrease in annual energy production (AEP) from the most productive year (2007) to the most recent year (2011). The degradation of the power curve was most pronounced between 2009 and 2010, as shown in Figure 21.

Figure 21. Power Curve Degradation between 2006 and 2011 for a Sample Turbine

The reason for the apparent decay in the power curve is not clear. Data were filtered to remove the majority of outliers. There was a HSS bearing replacement in 2010, and the turbine availability was generally down for that year. 2011 only includes data through September 2011, which could make this year appear slightly worse than years with complete years of data; however, a similar decay in the power curve can be observed when comparing power curves from a single month (e.g., June) across multiple years. From discussions with the Data Partner, it may also be possible that the anemometer signal has been corrupted and generated erroneous data. Neighboring turbines show some decay in their power curves, but not to the extent seen in Figure 21. In any case, it is unlikely the decay in the power curve is related to the drive train, but this analysis discovered a problem which can be identified by the SCADA data and addressed by the operator.

The relationship between power produced and generator speed was examined for correlations with drive train failures. In general no correlations could be made, but in one case a noticeable shift in the operational window, or “signature,” of generator rpm versus power could be seen in a turbine that experienced a gearbox failure. Figure 22 shows power plotted against generator rpm for two turbines, A-2 and A-3, for the years of 2008 and 2010. Turbine A-2 experienced a gearbox failure in early 2011, whereas turbine A-3 has not experienced a failure. In 2008 and all previous data, the two turbines had nearly identical signatures. In 2009 the signature of turbine A-2 shifted, with the turbine running at higher rpm to produce the same amount of power when compared to turbine A-3. Data from a third turbine were available to confirm that it was only turbine A-2 which experienced this change.

Figure 22. Power versus Generator rpm Signature Change between 2008 and 2010

Change is indicated by green arrows.

The change cannot be directly linked to the gearbox failure in turbine A-2, but it is an unusual trend that preceded the failure in 2011. Insufficient data were available to determine if the trend continued after the gearbox was replaced.

3.3.2.4 Fault Analysis

Wind turbines contain multiple sensors that monitor the status of turbine systems. When these sensors detect a parameter that is outside of normal conditions, they will trigger a “fault,” which may cause a variety of actions, including shutting down the turbine. The number of turbine faults in a given period of time may have a relationship with gearbox condition. Data Partner A provided DNV KEMA with a dataset containing gearbox condition (1 = new, 2 = wear appropriate for age, 3 = abnormal, 4 = critical with gearbox or bearing failure) and total fault counts in roughly 190 fault categories for all turbines at one wind power project. The date range for these fault counts is unknown, as is whether they run from any previous repairs or replacements of gearboxes. DNV KEMA conducted an analysis of these data to examine if a relationship exists between gearbox condition and turbine fault count.

A first pass through the data created a regression model for each fault category, taking condition as the response/dependent variable and the fault counts as the predictor/independent variable. The intent here was to examine the data to detect any relationships that might not be expected with a more targeted pass, and to identify faults that should be examined at other projects. This analysis alone is exploratory, and is not a basis for drawing conclusions about gearbox condition. Relatively few significant relationships were found, and all of them were either 1) for faults that had no plausible relationship to gearbox condition or 2) were driven by one or two outlier data points.

In addition to the simple regression data mining, plots were prepared to allow visual inspection of the results. For each scatter plot, y-axis shows gearbox condition and the x-axis shows fault count. Lines in the scatter plot are for a regression model that assumes condition as dependent variable and fault count as independent variable. This regression model is slightly flawed because gearbox condition is ordinal, so $2 < 4$ implies 2 is better condition than 4, but the interval between 2 and 3 is not necessarily the same as the interval from 3 to 4.

Box plots show condition on the x-axis and fault count on the y-axis. The boxes show the median as a thick line, the interquartile range (25% to 75% percentiles) as a box, and lines out to the 5% and 95% (or thereabouts) extent. Box plots provide a good visual indicator of the significance of data such as the fault count data. For our data, the box plots indicate a significant result when:

- 1) the median lines of the boxes are at significantly different locations, and
- 2) the boxes generally overlap vertically.

Plots were prepared for four conditions:

- 1) all faults using all conditions (2, 2.5, 3, 3.5, and 4)
- 2) all faults using binary conditions (2 and 2.5 become 2, while 3, 3.5, and 4 all become 4)
- 3) faults of interest for gearboxes using all conditions, and
- 4) faults of interest for gearboxes using binary conditions.

Figure 23 shows an example of a box plots for the “gearbox oil temperature too low” fault.

Figure 23. Box Plots Allow Visualization of the Data; this Plot Shows the Gearbox Oil Temperature Too Low Fault for Various Gearbox Conditions

Figure 23 shows that for the gearbox oil temperature too low fault, there is no correlation between an increasing fault count and a worsening gearbox condition. This result is typical of other faults investigated.

No evidence was found of a significant relationship between the measured fault counts and gearbox condition. This conclusion does not preclude the existence of such a relationship in a different project or a different turbine or SCADA combination. This conclusion is also based on an assumption that no repairs or replacements led to turbine condition “resetting” to 2 when it had earlier been a 3 or 4 during the course of the fault count collection.

3.3.2.5 Conclusions from Analyses of SCADA Data

DNV KEMA did not observe any trends in the SCADA data available for this project which clearly identifies early stages of drive train failure. This is not surprising, as the SCADA systems were not intended to be used for this purpose and in many cases failures would need to be measured indirectly by

the SCADA system. For example, a failure on a high speed pinion does not seem to measurably affect the temperature of the gearbox oil or high speed bearing, where temperature sensors exist.

DNV KEMA did not identify any cumulative events measured in the SCADA system that can predict drive train failures. Turbines which experienced failures did not show significant differences in the number of stops or starts, periods without movement, occurrences of high or low temperatures, etc. than turbines that did not experience failures. DNV KEMA was able to identify periods of abnormal operation in some turbines which experienced failures, but these events cannot directly be linked to the failures.

Previous work (3) has shown observable trends in gearbox oil temperature rise in the time period before gear failure; however, DNV KEMA did not observe similar trends in any of the failure cases when looking at either oil or bearing temperatures. This may be attributable to the fact that the failure types in the data available to DNV KEMA were less severe than in the previous work, and thus temperature changes of oil and bearing, a key indicator of failure, were less or negligibly variable in the DNV KEMA data.

Through the investigation of SCADA data, DNV KEMA did observe a number of interesting events and trends, including power curve shifts and gradual oil temperature increases in gearboxes over the course of several years that are indicative of abnormal operation. While the cause of these events has not been determined and is outside the scope of this work, identification of such trends is useful to project operators and may allow for detection and early remediation of issues. None of these findings provide specific information regarding the condition of the gearbox. The key outcome from this portion of the study is that there is no clear evidence that SCADA data contain detailed gearbox health information.

DNV KEMA understands from project operators that automated systems can be helpful when examining SCADA and other operational data. These systems are useful for detecting problems such as failed yaw motors and clogged heat exchangers (4); however, operators should not rely on automated systems to successfully detect incipient gearbox problems.

3.3.3 Analysis of Vibration Data

DNV KEMA analyzed the vibration data collected from turbines A-13 and A-14. No bearing or gear failure modes were present within the gearbox on turbine A-13, and the vibration data from this turbine formed the baseline for comparison with turbine A-14. Data from turbine A-14, however, showed that alarm limits as defined in VDI3834 were being exceeded, and the vibration data suggested that a rotor side high speed shaft bearing inner race fault was present. Borescope inspection confirmed the presence of a partial crack in the inner race of the rotor side high speed shaft bearing. This is an example of vibration analysis detecting a basic failure mode of high speed bearings.

In addition to the DNV KEMA vibration data from turbines A-13 and A-14, Data Partner A provided a summary vibration report from the turbine manufacturer for turbine A-2. This report identified a tooth fracture on the high speed pinion. The vibration system had been recently installed as a permanent installation, intended for long-term vibration monitoring. The location of the failure was identified by the vibration analysis with increased vibration levels around the high speed mesh frequency. Acceleration spikes were present once per revolution of the high speed shaft, which is indicative of a defect in a pinion tooth. A subsequent borescope inspection confirmed significant damage to a tooth on the high speed pinion. In this case, it was demonstrated that vibration analysis can accurately detect a significant gear failure in the high speed section.

DNV KEMA analyzed vibration data collected from one Data Partner B turbine (turbine B-2) with a failed bearing. The analysis indicated that alarm levels specified in VDI3834 were being exceeded, which would be expected for the severe level of bearing damage observed with the borescope (see Figure 5).

In addition to DNV KEMA analysis of the data, the data collected from turbine B-2 were sent to the supplier of the vibration instrumentation and vibration data analysis software. After analysis, the supplier indicated that there was some evidence in the data collected that there was a problem in the high speed section of the gearbox; however, the supplier was not able to locate the damaged bearing or describe the nature of the failure.

No vibration data were available from Data Partner C.

Our analysis indicates that vibration-based condition monitoring systems is useful for detecting high speed section bearing damage and significant damage in gear meshes. Discussion with vendors of vibration equipment and experience gained by NREL's Gearbox Reliability Collaborative indicate that detection of incipient damage and damage in planetary stages of gearboxes is more challenging for vibration analysis (5). This indicates that vibration analysis can be suitable for detecting some failures, but vibration analysis alone is not always a reliable source of bearing failure information. Vibration analysis that indicates possible damage should be followed by borescope inspection to accurately characterize the damage and determine the appropriate action.

3.3.4 Analysis of Oil Sample Data

Data Partner A provided oil sample reports and borescope inspection reports for turbines in their fleet. From the borescope inspection data, 25 of the gearboxes had failed, due either to a bearing or a gear failure. We investigated the oil analysis reports for the failed gearboxes in order to determine if a correlation exists between observed failures and oil analysis results. Of the 25 failed gearboxes, 11 had bearing failures and 14 had gear failures. From the laboratory analysis, 28% (seven of twenty-five) of gearbox failures at the site resulted in the oil analysis reporting an "Abnormal Machine" condition, 9% of oil samples on machines with bearing failures (one of eleven) resulted in "Abnormal Machine" condition, and 43% of oil analysis on machines with gear failures (six of fourteen) resulted in an "Abnormal Machine" condition. Other observations from borescope and oil analyses data provided by Data Partner A are:

- Iron content on turbines with ring gear failures in progress is slightly elevated over the turbines without ring gear wear.
- Elevated water content in the oil was found on several turbines. This is cause for concern because excessive water content in oil has multiple deleterious effects (6).

Similar results were found when analyzing data from Data Partner B.

Our examination of the oil sample analyses data from Data Partner A leads to two conclusions. First, laboratory oil analysis of machine condition does not detect bearing failures with the same reliability as gear failures. Second, the laboratory analysis of oil samples from turbines with known bearing problems was not effective at finding the types of bearing problems experienced on the turbines.

The oil sample taken by DNV KEMA from the Data Partner B turbine was sent to an accredited laboratory for analysis. The analysis report indicated that the "machine condition" was normal, and the "lube condition" was normal. This oil sample had been taken from a turbine with a severely failed bearing, yet the analysis report shows a normal machine condition. This information from the Data Partner B site is consistent with the conclusion based on information collected at the Data Partner A site:

gearbox condition assessment through laboratory oil analysis alone cannot be relied on as an accurate source of failure information.

The information from the Data Partner B turbine is a single point in time, and literature and discussions with wind project operators indicate that trending oil analysis data over time is useful for tracking oil condition. From (7), trending oil analysis data has some limitations but when performed properly can recognize bad samples, oil filter bypass operation, additive depletion, abnormal wear, and a change in oil type. In addition, a broad study of wind turbine gearbox oil over time (8) indicates that sufficient information is available when data are trended to determine end of oil life and schedule oil changes.

3.3.5 Misalignment Study

Operators are experiencing multiple failure modes in gearboxes, including high speed bearing failures. High speed bearing failures are occurring in machines built by many turbine manufacturers, gearbox manufacturers, and bearing manufacturers, in turbines with a variety of lubricant properties in a range of wind regimes. This widespread problem has significant ramifications for cost of operations and maintenance of wind projects, resulting in increased cost of energy.

Misalignment is one cause of bearing overloading, leading to premature bearing failure. A subtask for Task 3 involved an investigation of the effect of misalignment between the gearbox high speed output 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). Figure 24 shows a typical wind turbine high speed coupling. The laminate on either end of the coupling flanges allow for angular flexure; the fiberglass center spacer provides electrical isolation; a torque limiting feature is often included. Some turbine manufacturers design and manufacture their own couplings, others partner with coupling manufacturers such as KTR, Zero-Max, and Centa.

The impact of misalignment between the output shaft of the gearbox and the shaft of a generator was analyzed and discussed in (9). This paper showed some reduction in fatigue damage to gearbox high speed shaft downwind bearings with increased negative misalignment and increased coupling stiffness; however it showed an increase in fatigue damage with increased positive misalignment and increased coupling stiffness. The paper also indicated even at low coupling tilt stiffness and no misalignment, that fatigue damage exceeds premature failure thresholds. Positive misalignment is defined as shown in Figure 24.

DNV KEMA was able to further the work done by Whittle et. al. by examining measured gearbox movement data and then relating the results to realistic couplings stiffness values provided by coupling manufacturers. These data were used to calculate dynamic misalignment over a 2-month period of record. This calculated dynamic misalignment was then combined with assumed static misalignment and used to determine the force on the high speed shaft, which was translated to a bearing force, which was used to calculate L10 bearing life in accordance with (10). Three stiffness values and two levels of misalignment were considered. The details and results of this analysis follow.

Figure 24. Typical Wind Turbine High-Speed Coupling

The gearbox is on the right, and the generator is on the left.

In this analysis, three levels of misalignment were considered: (1) no misalignment, (2) a nominal level of static and dynamic misalignment, and (3) an extreme level of static and dynamic misalignment. Static misalignment is the misalignment between the generator and gearbox high speed shafts that exists when the turbine is not rotating. Static misalignment can be as-installed or due to deformation or movement of the bedplate, or feet of the gearbox or generator. For example, if the tower top is not perfectly level, the bedplate will shift and deform as yaw position changes.

From our experience inspecting operating turbines, 5 mm or greater static misalignment is not uncommon, particularly in aging machines. Depending on the direction of the static misalignment, it may be mitigated or exacerbated by thermal growth during operation (thermal growth driven misalignment was not specifically considered for this study). Static misalignment is typically measured on a cold machine. Some turbine manufacturers account for thermally-driven misalignment, as reflected by their commissioning instructions to align gearboxes and generators nominally off-zero. Adjusting for differences in alignment from off-line to running conditions, also called OL2R conditions, is common practice in coupled rotating machines.

Gearboxes have been observed to shift upwards of 15-20 mm under light wind loading. Insufficient evidence exists to know if this is typical or unusual; further investigation is warranted.

Torque acting on the gearbox is one factor driving dynamic misalignment. As torque is transmitted from the rotor through the gearbox, the gearbox torque arms experience reaction forces. Torque arms are typically secured to the bedplate via bushings, commonly made of elastomer. These bushings allow the gearbox structure to shift or translate in operation. Further, since the high speed shaft is not aligned on the same axis as the main shaft, the bearings supporting this shaft react some portion of the torque-driven forces as the gearbox tilts and rotates relative to the generator shaft. Figure 25 illustrates the forces that must be reacted by the gearbox and generator bearings.

Figure 25. Schematic of Drive Train Segment Indicating Forces on Bearings

A turbulent, variable wind drives the main shaft torque, so the torque too is variable. The variable load cycles are transmitted through the gearbox to the high speed shaft. If the coupling on the high speed shaft is very flexible, then the loading will lead to misalignment and generate heat in the flexing laminae of the coupling. Or, if the coupling is rigid, the loading must be reacted in the gearbox and generator bearings. We refer to this as torque-driven dynamic misalignment. Typically couplings are moderately flexible and some torque-driven dynamic misalignment is absorbed by the coupling and some is reacted in the bearings. This analysis aims to define this relationship between coupling stiffness and bearing loading. The loading is highly cyclical but also contains some excursions during high torque events, so both fatigue and extreme loading was considered. The topic is given minimal attention in standards. From (11), “special alignment considerations must be taken in to account for equipment that is started and stopped frequently 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.

Considering this point of view, we examined data measured on a turbine over a period of about 2 months. The turbine studied is rated in the 1-2 MW range, with a standard drive train configuration including elastomeric torque arm bushings and a flexible coupling. Measurements were taken of the torque arm displacements in three directions as well as the vertical displacement of the downwind (back) side of the gearbox. Other parameters such as torque, power, wind speed, and yaw position were also measured.

Gearbox displacement data analysis yielded the following conclusions. These conclusions apply to one specific test turbine and may not be applicable across turbine types:

- The maximum magnitude of torque arm displacements observed were on the order of 4 mm.
- Vertical torque arm displacement magnitude generally dominated over horizontal displacement magnitude.
- Vertical torque arm displacements generally followed torque trending, indicating torque is a significant driver for vertical gearbox movement.
- Horizontal torque arm displacements are less dependent on torque, indicating torque is not a significant driver for horizontal gearbox movement.
- The back of the gearbox deflects up to 3.5 mm, and is also somewhat driven by torque.

Figure 26 shows the relationship between torque and vertical displacement, calculated as the vertical displacement of one side of the gearbox minus the vertical displacement on the other side. Each point represents data from one 0.05-second time period.

Figure 26. Torque Arm Displacement versus Torque Normalized to Maximum Torque

Using the generator displacement data, and gearbox geometry, we calculated movement of the high speed shaft. The assumptions behind this calculation include the following:

- The gearbox is a rigid body
- The main shaft does not translate
- The torque arm displacements reflect rotation of the gearbox around the main shaft
- The high speed shaft movement is the cumulative movement of the gearbox rotation (from torque arm displacements) and tilting of the gearbox (measured by the displacement at the back of the gearbox)
- All parts exhibit uniform properties that do not change with load
- This is a static analysis in that transient events were not assessed

The high speed shaft movement was calculated for each record, representing dynamic misalignment. Dynamic misalignment was combined with an assumed static misalignment and the force on the high speed bearing was calculated using three separate coupling stiffness values. The assumptions behind these calculations include:

- The coupling stiffness is constant through any load or misalignment
- Generator side of the coupling is rigid; meaning that the generator is fixed and does not move

Extreme loading was determined to be benign compared to the typical design basis; however, fatigue loading was significant in some cases. Bearing L10 life was calculated using the method defined in IEC 61400-4. Results from the calculations are shown in Table 5.

Table 5. Reduction in L10 Bearing Life for Three Coupling Stiffnesses and Two Misalignment Cases

Coupling Stiffness (Nm/rad)	2400	15000	400000
No misalignment (baseline)	0%	0%	0%
Dynamic misalignment plus 5 mm static misalignment	0.2%	1.4%	32.7%
10x dynamic misalignment plus 10 mm static misalignment	0.7%	5.0%	54.8%

A coupling stiffness of 2400 Nm/rad is at the lower end of the range of stiffnesses provided to us from conversations with wind turbine coupling manufacturers. 15000 Nm/rad is a coupling stiffness at the upper end of the range of values we obtained through discussions with coupling manufacturers. The extreme stiffness value of 400000 Nm/rad was used as an upper bound on stiffness values, following Whittle et. al. Stiffness is dependent on temperature, load, and misalignment, so using one stiffness value across a large range of misalignments may lead to a bias in the results. We do not have detailed information on how the stiffness changes with misalignment, but recommend the industry better understand this property, as our results show that it strongly influences bearing life.

We also confirmed the results of our measurement-based analysis by comparing them to the results of a model-based analysis. Using multi-body drive train analysis software developed by DNV, we modeled the drive train elements, loading, and estimated fatigue life using the same L10 approach. The results of the two methods were generally in agreement.

Additionally, a calculation of static force was performed and a sensitivity analysis was run. The sensitivity analysis showed that the gearbox geometry, specifically the distance between main shaft and high speed shaft and the distance between the main shaft and the torque arms, was the most significant factor in static force. The angular stiffness of the coupling was the second most influential factor.

This study shows that static forces generated by misalignment in normal operational range of couplings are small compared to loads due to torque that high speed bearings are designed for. Highly flexible couplings operating in their nominal range effectively mitigate bearing forces due to parallel misalignment, and bearing life is not significantly reduced by misalignment. However, if misalignment exceeds the operating range of the high speed coupling, then large forces will be generated and bearing life will decrease appreciably.

The study points to ongoing research needs. A number of the assumptions made in this study should be systematically removed in order to obtain results that better represent actual turbine behavior. First, dynamic misalignment measurements from an operating turbine will allow characterization of actual misalignment magnitudes. Coupling temperature is one indicator of misalignment, and could be used as a surrogate for actual dynamic misalignment measurements if calibrated. Static misalignment should be better understood across turbine age and type. Second, full characterization of coupling stiffness as a function of parallel misalignment is needed. This includes understanding the stiffness of the coupling when it is operating outside of its nominal parallel offset range. Additionally, the study was performed on

data measured from one turbine. Different drive train designs will impact how much the gearbox moves during operation, and therefore, will impact the dynamic misalignment expected.

The study also raises questions that should be kept in mind when pursuing further investigations. First, how do dynamic and static misalignments impact the cost-benefit analysis of utilizing couplings with a broader range of allowable misalignments? Second, should wind turbine loads simulation models include gearbox motion and the effect on misalignment? Finally, should IEC type testing of wind turbines include dynamic alignment measurement? Each of these topics should be examined further if future studies show fatigue loads associated with misalignment to be a problem.

This study has launched a new avenue of investigation into the root cause of gearbox high speed bearing failure. This information is also applicable to the study of the cause of generator bearing failure, and of failure of high speed couplings themselves.

This topic was presented at the AWEA O&M conference in San Diego in January 2012, and was selected for a podium presentation at AWEA Windpower in Atlanta in 2012. Further, a draft of a journal article for *Wind Energy* (ISSN 1095-4244, published bimonthly by John Wiley & Sons, Ltd.) is under development.

3.3.6 Summary of Task 3

Key findings from Task 3 are:

- There is no clear evidence that SCADA data contain detailed gearbox health information. Once a gearbox has reached an advanced stage of failure, SCADA data are likely to indicate changes that can be correlated to the failure. Our study found, however, that insufficient information is present in typical 10-minute SCADA data to provide early indication of gearbox failure, or even to identify existing damage. SCADA data interpretation is useful for detection of problems with some turbine systems, but not necessarily for problems with the gearbox. Automated systems may improve this in the future, but industry experience suggests that these systems are currently limited in their ability to provide early indication of gearbox failure.
- Visual inspection by borescope provides a direct indication of the health of the gearbox internal components. There may be access limitation, so all internal components cannot be observed, and the process is invasive and may be time consuming, but it provides the most direct indication of the gearbox condition.
- Gearbox condition assessment through laboratory oil analysis alone cannot be relied on as an accurate source of gearbox failure information.
- Vibration analysis can provide indication of bearing and gear failure; however, detection of bearing failures in planetary stages can be challenging for vibration analysis.
- No evidence was found of a significant relationship between measured turbine fault counts and gearbox condition.
- Highly flexible couplings operating in their nominal range effectively mitigate bearing forces due to parallel misalignment, and bearing life is not significantly reduced by misalignment. However, if misalignment exceeds the operating range of the high speed coupling, then large forces will be generated and bearing life will decrease appreciably.
- In order to accurately monitor and manage gearbox health, multiple sources of data must be collected and assessed.

3.4 Task 4: Develop a Framework for Recommended Practices for Gearbox Health Management

The framework for a Recommended Practice for Wind Turbine Gearbox Health Management is a set of practices for wind project asset managers who want to minimize the risk of premature wind turbine gearbox failure. The intent of the recommendations is to enable asset managers to minimize the risk of premature failure of gearboxes that they operate. Unlike gearbox design standards, these recommendations focus on operational monitoring of gearbox health, and are intended to treat topics where wind project operators or asset managers have control over actions that can be taken.

The recommendations are broadly divided into three subject areas that span the field of gearbox health management. These subjects: maintenance practices, condition monitoring, and turbine performance monitoring, are further subdivided into specific areas where our research shows that guidance is warranted. Each of these subject areas is treated in detail in the Recommendations Appendix (Appendix A).

Easily accessible data on the factors that maximize gearbox life are not readily available in industry. Multiple database efforts are currently underway, including the NREL Gearbox Reliability Collaborative, Sandia's CREW database, and efforts in Europe. The recommendations incorporate best practices that are based on data primarily from a small selection of operating projects in North America, but also incorporate knowledge and data from all of these efforts where necessary and possible.

The recommendations build on the substantial body of work related to best practices for wind turbine gearbox durability ((10) and (12)). As update cycles on standards such as the 81400-4 standard are lengthy, current research and experience are not reflected in the most up to date edition of the standard. The recommendations incorporate current information.

The American Wind Energy Association (AWEA) has convened operations and maintenance (O&M) working groups to produce recommended practices around a variety of topics, including gearboxes. The recommendations from this study are intended to support the development of recommended practices for gearbox O&M.

Specific activities from this program that supported the development of the recommendations involved the data investigation and analyses described in the previous sections, plus multiple industry interactions over the course of this project. Industry interactions included discussions with project operators on gearbox operational practices, attendance at the NREL GRC meetings, and observation of gearbox teardown inspection after failure.

From discussions with project operators, we found a theme that each gearbox is a different machine. One industry expert indicated that each gearbox has a "unique personality," and an asset manager simply said that some gearboxes are looser than others. We worked to accommodate the range of gearbox characteristics when formulating general recommendations for gearbox health management.

As a part of our development of the recommendations for gearbox health management, we examined current standards covering wind turbine gearbox design. We examined the DEA (Danish) gearbox guideline and ISO 81400-4. The objective of the assessment and comparison of the standards was to determine if any areas of current standards are not accounting for actual experiences of wind project asset managers. Generally, these standards are used by turbine and gearbox manufacturers to generate design specifications, and contain more information regarding design and load criteria and less information on

the service phase of a gearbox. There is some information about lubricant properties and requirements that is useful for the service phase of a gearbox, and Appendix F of the ISO standard contains some guidance for the service phase of a gearbox. The key finding from the assessment is that the standards lack specific requirements for condition monitoring, predictive health assessment, and preventive maintenance actions. This is further indication that there is a need for industry-wide recommendations on gearbox health management for wind project managers.

4 ACCOMPLISHMENTS

Over the course of the project, there have been multiple significant accomplishments. The accomplishments can be grouped into three categories: *competence development*, *research progress*, and *industry collaboration*. Accomplishments in each of these categories are described below.

4.1 Competence Development

Members of the project team have all gained knowledge related to wind turbine gearbox durability by participating in the program. Our team has evolved since project inception, and consists of a full range of experienced engineering personnel from within and outside of DNV KEMA. The project has provided participants with a broader knowledge of gearbox failure modes, data analysis methods, and techniques that can be used to detect impending gearbox failures.

As a part of the program, DNV KEMA held a bearing failure analysis workshop, exposing personnel within DNV KEMA to the basics of wind turbine gear and bearing failure. Furthermore, experience and background knowledge that DNV KEMA employees and others have gained and shared through this DOE-funded effort has benefited other projects within DNV KEMA and for DNV KEMA clients.

The industry has also benefited through progress in research and collaboration, discussed below.

4.2 Research Progress

Two specific avenues of research within this project have advanced the knowledge base of the wind industry.

First, our examination of SCADA data and actual gearbox failure information has enhanced the body of knowledge of the use of SCADA data for determination of gearbox health.

Second, our in-depth look at the effect of misalignment on high speed bearing loads has broken new ground in the field, including putting data behind a cause of bearing loads that has only recently been studied in open literature applied to wind turbines. The work was presented at the AWEA Wind Project Operations, Maintenance, and Reliability Seminar, and was accepted for a podium presentation at the 2012 AWEA Windpower conference. A submission to the peer-reviewed journal *Wind Energy* on the topic is being prepared.

The study of the effect of misalignment on bearing life reduction has the potential to lead to new practices and improve wind turbine operational life. The study points to the need for measuring dynamic alignment in turbines. If measured misalignments are resulting in high speed couplings operating outside of their nominal stiffnesses, then wind project operators should take action to limit dynamic misalignment or install couplings with a greater low-angular-stiffness range. A change like this may reduce bearing failures, which will reduce maintenance costs. This has the potential to lower operating costs for project operators, ultimately leading to a lower cost of energy for wind-generated electricity.

Finally, development of recommendations for wind turbine gearbox health management provides guidance to wind project asset managers.

While wind turbine gearboxes are likely to remain problematic components, this research has moved the industry a step closer to reducing the cost of energy by improving gearbox durability.

4.3 Industry Collaboration

This DOE project has created multiple avenues of collaboration within the wind turbine gearbox community.

As a part of our activity in the project, we have met and talked with multiple wind project operators regarding the state of their gearboxes, conditions that lead to gearbox failure, and data that support conclusions that can be made regarding gearbox health management. Our interaction has included climbing turbines at Data Partner project sites to collect condition monitoring data, inspecting Data Partner gearboxes, and in one case, discovering a previously-unknown high speed bearing failure in a Data Partner gearbox with our borescope.

In addition to collaboration with wind project operators, we worked with U.S. Government laboratories involved in wind turbine research related to gearbox durability. Both NREL and Sandia are working to improve gearbox durability. Support of NREL GRC is an explicit task of this project, and we participated in GRC meetings throughout the project duration.

Our presence at NREL's Condition Monitoring workshop and the NREL Advanced Drive Train Workshop, and NREL GRC meetings allowed interaction with key research and industry community members.

The Sandia reliability workshop is a venue for the industry and research community to share approaches to improving wind turbine reliability. Through this DOE project, our presence at the workshop enabled learning and collaboration regarding data capture.

Engaging with external consultants has expanded our reach. The industry benefits most when collaboration and cooperation, rather than competition, drive progress. We brought in expertise from GL Garrad Hassan, normally a DNV KEMA competitor, for a review of this report. Further, we brought in Andy Milburn, one of the industry's most experienced gearbox experts, for a brief review of some elements of this report.

All of these collaborative experiences helped to develop the primary deliverable from the project, which are recommendations for gearbox health management. This pioneering document brings together practical experience from project operators and knowledge from current research to provide guidance to wind project asset managers on maximizing the longevity of the gearboxes in their turbines.

5 CONCLUSIONS

We draw the following conclusions from our study:

- *The common gearbox health management tools available today are useful, but their limitations must be understood.* Visual inspection, oil analysis, and vibration analysis are typical tools used to assess gearbox health. Each has strengths, but the limitations of each must be understood and multiple information streams must be brought together to gain a complete picture of gearbox health.
- Visual inspection of gearboxes with a borescope provides a direct indication of gearbox health, but many areas of the gearbox cannot be inspected due to access limitations. Borescope inspections are also costly and time consuming, so these inspections may not be practical as a tool for monitoring degradation over time, as most operators do not inspect gearboxes on even an annual basis.
- Oil analysis characterizes gearbox oil condition, which is useful, but the analysis cannot reliably predict specific failures of the gearbox itself. In many cases where gearboxes failed, oil test results indicated no abnormal conditions.
- Vibration analysis has potential as a method of detecting anomalies, but results can vary widely based on the data quality and the skills of the analyst examining the data. Long-term vibration measurements may provide some predictive capability and detect early stages of failure, but this was not proven in this study or demonstrated in any vibration measurements from Data Partners.
- *There is no clear evidence that SCADA data contain detailed gearbox health information.* Once a gearbox has reached an advanced stage of failure, SCADA data are likely to indicate changes that can be correlated to the failure. Our study found, however, that insufficient information is present in typical 10-minute SCADA data to provide early indication of gearbox failure, or even to identify existing damage. SCADA data interpretation is useful for detection of problems with some turbine systems, but not necessarily for problems with the gearbox. Automated systems may improve this in the future, but industry experience suggests that these systems are currently limited in their ability to provide early indication of gearbox failure.
- *If gearbox-generator misalignment exceeds the operating range of the high speed coupling, then large forces will be generated and bearing life decreases appreciably.* Overload has been postulated as one possible source of bearing failure, and we have identified a potential source of overload. Investigation of misalignment, the forces it generates, and the impact on bearing life requires further study.
- *Openness is required in data collaboration.* One of the problems encountered during the execution of the project was collection of detailed data from a broad range of Data Partners. Far less data than originally anticipated was collected. There is a general perception within the U.S. wind industry that operational data need to be protected as intellectual property. We believe the industry can advance more rapidly if greater transparency with operating data is achieved.
- *A source of guidance to the industry for gearbox health management is needed.* Maximizing gearbox life increases uptime and decreases maintenance cost, resulting in a reduction in the cost of wind energy. Good gearbox health management practices can help maximize gearbox life. A substantial amount of guidance is available to operators, and we have developed recommendations for gearbox health management that bring together these current industry perspectives on methods for maximizing gearbox life.

The accomplishments made in this study can be grouped in to three categories: competence development, research progress, and industry collaboration. We developed internal competence regarding SCADA data analysis and gearbox inspection. We advanced the state of the art in understanding sources of overload on

bearings through our study of misalignment. We contributed to interactions within the industry, including working with wind project operators as Data Partners and supporting the ongoing Gearbox Reliability Collaborative hosted by the National Renewable Energy Laboratory. The NREL GRC is providing useful research into the effectiveness of design changes to gearboxes, assessment of condition monitoring systems, and improvements in gearbox modeling technology.

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Appendix A. Recommendations for Wind Turbine Gearbox Health Management

A.1 Introduction

A.1.1 Background

DNV KEMA prepared these recommendations for wind project asset managers who want to minimize the risk of premature wind turbine gearbox failure. The recommendations are intended to enable asset managers to minimize the risk of premature failure of gearboxes that they operate. Unlike gearbox design standards, these recommendations focus on operational monitoring of gearbox health.

A.1.2 Objectives

The objective of the recommendations is to provide specific actions that wind turbine operators can take to maximize the useful life of wind turbine gearboxes and understand the condition of gearboxes. The actions are intended to guide asset managers with regard to practical operational aspects of wind turbine gearboxes.

A.1.3 Scope

The recommendations cover health management of wind turbine gearboxes and apply to gearboxes that are installed and in operation, rather than new gearbox designs.

Gearbox configurations in wind turbines vary. As a result, not all recommendations are applicable to all gearboxes. The general principles in this document, however, are intended to apply industry-wide.

The recommendations do not aim to comprehensively cover all maintenance actions and condition monitoring techniques that should be applied to a wind turbine gearbox. Rather, we seek to provide guidance to asset managers for making decisions regarding their gearboxes. Even with proper maintenance and careful, attentive condition monitoring, gearboxes may fail prematurely, and industry experience indicates that even well-cared-for gearboxes fail prematurely. The recommendations are intended to support decision making that will maximize the life of wind turbine gearboxes.

Safety should be any machine operator's first priority. Appropriate safety measures must be applied when working with wind turbine gearboxes.

A.1.4 Overview of the Recommendations

The content of this document is based primarily on data and research conducted by DNV KEMA under contract to the U.S. Department of Energy. The recommendations are based in part on data collected from operating projects in North America, as well as the sources cited in the body of the report, and DNV KEMA's professional experience. Wind project operating data are often tightly held by wind plant operators, gearbox manufacturers, and turbine manufacturers. The recommendations herein simultaneously respect the intellectual property rights of operators by keeping confidential data undisclosed, while reflecting broad industry experience through our interaction with multiple wind project operators.

Easily accessible data on the factors that maximize gearbox life are not readily available in industry. Multiple database efforts are currently underway, including the NREL Gearbox Reliability Collaborative,

Sandia's CREW database, and efforts in Europe. The recommendations herein incorporate best practices that are based on data primarily from a small selection of operating projects in North America, but incorporate knowledge and data from all of these efforts where necessary and possible.

The recommendations apply to topics where wind project operators or asset managers have control over actions that can be taken. In this document, we collect the best practices and experience of turbine operators and present guidance relevant to wind project asset managers to help maximize gearbox life.

The recommendations are broadly divided into three subject areas that span the field of gearbox health management: maintenance practices, condition monitoring, and turbine performance monitoring. These subjects are further subdivided into specific areas where our research shows that guidance is warranted. Each of these subject areas is treated in detail below.

A.1.5 Information Sources for Project Operators

There are multiple sources of information for gearbox maintenance practices, including the turbine manufacturer, the gearbox manufacturer, industry standards, and institutional knowledge. Annex D of ISO 81400-4:2005 describes operation and maintenance (O&M) procedures for wind turbine gearboxes. There is some overlap between the content of these recommendations and Annex D; however, these recommendations expand on Annex D by highlighting industry best practices.

The American Wind Energy Association (AWEA) is currently in the process of assembling O&M recommended practices, which will include guidance on gearbox O&M. These important recommendations are technician-level, whereas the content of this document is focused on guidance for asset managers who are directing the work of technicians. Again, some overlap may exist, and these recommendations are intended to complement the AWEA work.

A.2. Maintenance Practices that Maximize Gearbox Life

A.2.1 Lubrication

Good lubrication is essential for maximum gearbox life. Providing good lubrication means keeping the gear meshes and bearings properly supplied with the right type of cool, clean, and dry oil. Many decisions must be made to achieve proper lubrication. The subsections below are intended to aid in making the right decisions regarding lubrication.

Some elements of lubrication are designed into the gearbox, and the operator has little influence on these parameters. For example, the plumbing and oil distribution system within a gearbox may be difficult or impossible to successfully modify without great expense and careful experimentation. The metallurgy of gears and bearings are predefined, and the bearing type selection has been made. Generally preloads and clearances are set, except for some uptower repairs. This section covers elements of lubrication over which wind plant operators and asset managers can exercise control.

A.2.1.1 Selection of Oil Type

Gear oils commonly used in wind turbines can be divided into the structure shown in Figure A-1 below (following (10) and (13)).

Figure A-1. Wind Turbine Gear Oil Structure

Figure A-1 shows the structure for typical oils in wind turbines; other base stocks (silicone, diester, neopentyl polyesters, phosphate esters) oil are available (14). Thickeners are also available in addition to common additives.

There is no consistent industry best practice regarding which type of oil to use in a wind turbine. While sources such as (15) indicate that PAG oil will provide the best performance because the maximum elastohydrodynamic film thickness will be obtained, and increasing this film thickness increases gear and bearing life, there are multiple factors that must be considered when selecting the proper gear oil.

Proper oil type must be selected for each stage of gearbox operation, including run-in, flush, and normal operation. Lubricant selection is discussed in Annex F (section F.1.3) of ISO 81400-4. DNV KEMA recommends following the lubricant selection guidance in this Annex.

Adjustments to oil properties such as viscosity can have certain short-term benefits, such as decreasing wear rates in a specific section of a gearbox. The oil, however, circulates throughout the whole gearbox, and the effect of oil modifications will affect all components within the gearbox. If oil is reformulated to enhance the life of specific gearbox components, the effect of the reformulation on all gearbox components (including paints, seals, and filters) should be considered.

A.2.1.2 Oil Filling

When oil is added to a gearbox, the potential exists for 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 published a “Recommended wind turbine gearbox lube oil change procedure” (16). At a minimum, we recommend following this procedure for wind turbine oil changes, with the exception of the use of solvents discussed in section 3 of the procedure; we do not recommend use of solvents for flushing gearboxes. From this procedure, proper oil filling is achieved by adding “prefiltered oil from a filter cart connected to the gearbox with quick-connect couplings.” Some turbines are not equipped with quick-connect couplings; we recommend retrofitting oil circuits with quick-connect couplings if they are not supplied as standard equipment.

A.2.1.3 Oil Sampling

The process of oil sampling may introduce dirt, moisture, or other contamination in to the gearbox lubrication. We recommend following the sampling procedure described in ISO 81400-4 section F.5. We further recommend (following (17)) flushing 10 times the volume of dead space between the sample valve and the active system prior to sampling.

Filtration removes particles from oil, so the oil sample should be collected upstream of the filter. The sampling procedure described in ISO 81400-4 section F.5 includes this recommendation.

A.2.1.4 Oil Filtration

Clean oil is essential for gearbox longevity. Bearing life has been shown (18) to decrease as coarser filtration levels are used. Figure A-2 shows the relationship between filter rating and bearing life. Caution should be used when extrapolating these specific results from (18) to large bearings; however, the general relationship between filtration ratio and bearing life can be seen.

Figure A-2. Relationship between Filter Rating and Bearing Life

While Figure A-2 shows the general relationship between filter rating and bearing life, bearing life calculations include consideration of lubricant cleanliness, temperature, and other factors. A detailed discussion of the calculations of these bearing calculations is beyond the scope of these recommendations; however, other factors influencing bearing life are discussed within this document.

There is broad data indicating oil containing excessive contaminants can cause premature wear of bearings and gear meshes (19). In order to maintain adequate cleanliness and pump the required capacity

of oil, oil should be filtered through two circuits, an online and an offline circuit (20). Filtration levels specified in ISO 81400-4 Table 17 should be achieved.

We recommend that filters are checked and cleaned or changed no less frequently than every 6 months (the minimum recommended in ISO 81400-4 section F.4). While some gearbox manufacturers are recommending longer intervals for filter changes, dirty oil in offline filtration systems may fill filters more frequently than the 6-month filter inspection interval, which may cause an in-line system to operate the bypass valve, where the benefit of fine (3 micron) filtration is lost. Close monitoring of signals from differential pressure sensor across the filter should be performed, including trending to allow prediction of filter cleaning or replacement requirements.

There is speculation that too much filtration removes oil additives. Very fine filtration removes particles that are at a typical minimum dimension of 1 micrometer. Additives in oil are typically long-chain molecules with lengths on the order of 10 to 100 Angstroms (0.001 to 0.01 micrometers). The filter will not block passage of these molecules. Filters with adsorbents that are chemically active may remove some oil additives through chemical reactions, but depth-type absorbent filters will not remove additives in solution.

Fine filtration ratios may cause electrostatic discharge, which may lead to lubricant decomposition (2). Bypass filtration systems flow the oil at a lower rate, reducing the possibility of electrostatic discharge.

A.2.1.5 Oil Circulation

Even when the turbine is idling, gears and bearings need lubricant under sufficient pressure, and the oil distribution system should provide the appropriate quantity and pressure of lubricant to the bearings and gear meshes. Oil distribution system components are typically fixed by design and are difficult for an operator to modify. There are, however, several components in the oil circuit that the operator should maintain in order to keep oil circulating properly.

Oil pumps and hoses should not be overlooked as maintenance items. A complete oil pump or hose failure is likely to cause a turbine fault and shutdown. In order to avoid even brief periods of operation with insufficient oil pressure or insufficient flow rates, oil pump condition and function should be checked proactively during maintenance cycles. Any leakage around the oil pump, pump motor, or hoses should be corrected, and electrical connections to the pump motor should be checked during visual inspection of the gearbox.

A.2.1.6 Oil Temperature

Operation at elevated temperatures reduces the service life of bearings for three reasons (18):

1. The bearing internal clearances are changed by thermal expansion, which affects load distribution within the bearing and individual roller contact loads.
2. The lubricant film thickness is changed (reduced with increasing temperature).
3. The hardness of the bearing steel decreases. This effect is generally not a concern with typical wind turbine gear oil operating temperatures.

Furthermore, higher oil temperatures lead to reduced oil life, as the rate of oxidation of the constituents of the oil increases with increasing temperature.

Proper and regular maintenance of the gear oil cooling system will help to ensure a consistent supply of cool oil to gear meshes and bearings, and may reduce turbine faults due to high temperatures. Both the internal and external surfaces of heat exchangers can foul (Figure A-3), reducing the efficiency of the heat

exchanger. Heat exchanger sizing includes a factor for fouling; however, operator data have shown that oil temperatures can creep up over time due to accumulation of airborne debris in heat exchangers exposed to the air.

Figure A-3: Heat Exchanger in Oil Cooling System with Fouled Fins

Image provided courtesy of Whitewater Hill Wind Partners, LLC

Check the cleanliness of heat exchangers in the oil cooler circuit during any routine visit to the turbine. The external surfaces of air/glycol or air/oil heat exchangers can be checked visually. A regular maintenance schedule for heat exchangers should be implemented and followed, particularly at sites where dust or organic debris may accumulate on heat exchanger elements or air filters exposed to the free stream of air. Cleaning the surface of the heat exchanger is not sufficient; the air passageways in the heat exchanger must be kept clear.

Heat exchanger cleaning intervals or air filter replacement intervals should be set based on visual inspection and oil temperature data. Monitoring airflow through forced-air heat exchangers (Figure A-4) can also be a useful measure of the amount of fouling and the effect of cleaning.

Figure A-4. Monitoring Air Flow through Forced-Air Heat Exchanger

Image provided courtesy of Whitewater Hill Wind Partners, LLC

A.2.1.7 Moisture Content of Oil

From (10) and (21), moisture in oil may contribute to the following problems:

1. *Corrosion.* Water gives acids increased potential for corrosion.
2. *Additive depletion.* May deplete antioxidants in the oil, and diminish the performance of other additives in the oil.
3. *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.
4. *Aeration and Foam.* Water lowers the interfacial tension of oil, increasing the risk of foaming.
5. *Impaired film strength.* Water's limited film strength limits its capability to bear loads at high pressure.
6. *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.
7. *Hydrogen-induced fracture.* There is a current hypothesis that risk of hydrogen embrittlement is posed by both soluble and free water. Sulfur from additives, mineral oils, and environmental hydrogen sulfide may accelerate the progress of the fracture.
8. *Micropitting.* The mechanism of water promoting micropitting is unproven (2), 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 earlier in the life of the gearbox that moisture is detected in oil and removed, the better. In our experience, 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 (22) developed a relationship between moisture content and bearing life (Figure A-5), and Day and Vesala (23) adapted Cantley's data to provide guidance for selecting minimum appropriate levels of free water in oil. Barnes (24) presents similar results showing the degradation of bearing life with increasing water content in oil. Based on results from these studies, rather than choosing the ISO 81400-4 upper limit of 500 ppm, we recommend that operators set a lower water content limit for the turbines at their site in order to maximize bearing life.

Figure A-5. From Cantley (22), where bearing life relative to 100 ppm = $(100/(\text{water concentration in ppm}))^{0.6}$, test parameters are SAE 20 mineral oil, 65.5 °C oil temperature, 2.03 GPa stress level, 2700 RPM Speed

From Figure A-5, reducing water content from 500 ppm to 200 ppm will increase bearing life by 27% for the specific parameters studied by Cantley. While these parameters are not directly representative of typical wind turbine conditions, Figure A-5 provides an indication of the effect of water removal on bearing life. The protection level and life extension level approaches discussed by Day and Vesala provide specific guidance on setting limiting values based on oil formulation and minimum expected temperatures. We recommend setting water content limits lower than 500 ppm based on this guidance, and controlling moisture to remain as low as practically achievable.

Vacuum dehydrators or other separators are available to dry oil (6). Membrane dehydrators (25) are an emerging technology that shows promise for online moisture removal.

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; we concur with this recommendation.

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. To reduce airborne moisture content operate gearboxes with filtered and regenerable desiccant breathers, or a sealed gearbox (26), to avoid ingestion of airborne particles into the gearbox. Breather clogging has been a possible cause of seal failure and oil leakage. Desiccant filters can fill up quickly with water, and regular maintenance is required. Ensure that breathers are properly maintained; inspect breathers regularly and replace any clogged breathers.

A.2.1.8 Oil Changes

Oil can degrade through multiple processes, including oxidation, thermal breakdown, pressure-induced thermal breakdown, additive depletion, electrostatic spark discharge, or contamination. As oil degrades, the water content may increase, cleanliness decreases, and viscosity changes. Degraded oil may require removal and replacement with new oil.

Oil manufacturers, turbine manufacturers, and gearbox manufacturers may provide conflicting information regarding oil change frequencies. Most operators base decisions on when to change oil on the oil condition by conducting oil sample analysis. Turbine manufacturers provide decision trees indicating if and when oil needs to be changed, based on oil sample analysis results. Furthermore, limits are provided in Table F.4 of ISO/ISO 81400-4, and the standard indicates that corrective action should be taken if the limits are exceeded. The cost of oil changes is not explicitly considered in these guidance documents.

Oil maintenance should be conducted in a manner that cost-effectively maximizes gearbox life. An analysis is required that minimizes total cost, given the cost of oil changes, the probability of failure due to oil degradation, and the cost of failure due to oil degradation. As a basic outline to the steps of this analysis, we recommend:

1. Track fleet history and oil analysis data to determine parameters defining oil condition degradation and probability of failure due to oil degradation.
2. Determine cost of downtime and maintenance if a failure due to oil degradation occurs.
3. Base oil change decisions on a comparison between oil change costs and costs of downtime and maintenance determined above.

A.2.1.9 Oil Leaks

Leaks indicate that there is an uncontrolled passageway between the inside of the gear case and the external environment. Leaks cause the following problems:

1. Ingress of moisture or other contaminants is possible.
2. Oil leaks may lead to safety hazards such as slipping. Oil leaks make other turbine maintenance more hazardous.
3. Oil leaks reduce the quantity of oil in the gearbox.

Any oil leaks should be corrected at the earliest possible opportunity.

A.2.2 Gearbox-Generator Alignment

Research indicates that gearbox-generator misalignment may be a contributing factor to gearbox high speed bearing failures and generator bearing failures. If excessive misalignment exists between the gearbox and generator shafts, the high speed coupling may be operated outside of its normal range of operation, and unanticipated high loads can be imparted to the bearings of the gearbox high speed shaft and generator.

Some turbine manufacturers incorporate an initial non-parallel alignment setting in an attempt to account for deflection of the drive train under load and thermal expansion of the gearbox and generator during operation. As temperatures change in the gearbox and generator due to power production or ambient temperature fluctuations, the geometry of the gearbox and generator will change due to thermal expansion and contraction. Initial alignment should be set to the turbine manufacturer's specification. Alignment

should be checked at regular intervals, and realignment should be performed if the alignment is outside of specifications.

When realigning the generator and gearbox, if possible, measure alignment when the turbine is:

1. Cold oil temperature, turbine not running,
2. Hot oil temperature, turbine not running, and
3. Hot oil temperature and running at across the full power/speed range (dynamic misalignment measurement).

With these measurements, a specification for static alignment can be developed that minimizes misalignment during operation. We recommend aligning the gearbox and generator such that parallel and angular offsets are minimized when the turbine is at nominal operating temperature and running at rated power.

Alignment check intervals vary among turbine manufacturers; typically alignment is checked yearly and after replacement of a gearbox or generator. We recommend that when alignment checks are performed the data are tracked and trended. If alignment is trending away from nominal at an increasing rate, inspect for possible causes of degrading alignment, such as bushing wear or degradation, and realign the generator to the gearbox.

If high dynamic misalignments are suspected or measured, for turbines out of warranty, consider replacing high speed couplings with couplings with the least possible angular stiffness and the greatest range of parallel misalignment. Torsional stiffness of any retrofit coupling must be equal to the torsional stiffness of the original coupling in order to avoid introducing new drive train oscillations.

A.3 Condition Monitoring

Condition monitoring of a wind turbine gearbox is action taken to monitor the health of the gearbox. Condition monitoring can take many forms, from basic visual inspection to monitoring based on extensive instrumentation and data analysis. Below are our recommendations regarding condition monitoring.

A.3.1 Establishing Baseline Conditions

Effective condition monitoring of any kind requires comparison of the condition of the gearbox to some prior known condition. Many operators have benefited by establishing baselines for critical turbine parameters during commissioning, before production starts.

It is recommended that baselines be established for, at a minimum, the following parameters:

1. Gearbox visual condition, internal and external, including borescope inspection
2. Gearbox-generator static alignment
3. Oil condition
4. Torque arm bushing geometry and condition
5. Generator support bushing geometry and condition
6. Vibration signature
7. Bearing temperature rise over ambient
8. Oil temperature rise over ambient
9. Power curve

A baseline should be established immediately after installation of condition monitoring equipment. For operating projects that do not have a baseline previously established, or instances where responsibility for monitoring the health of gearboxes changes hands, a baseline should be established as soon as is practical.

If major components are replaced, a new baseline will need to be established.

See also (27) Section 8.10 for information on baseline data in condition monitoring systems.

A.3.2 Visual Inspection

Visual inspection is the most fundamental method of condition monitoring. Visual inspection should be conducted regularly by trained and competent personnel. All safety protocol should be followed during visual inspection.

Borescope inspection, also called videoscope or video probe inspection, enables direct visual assessment of the condition of gearboxes.

Some gear and bearing failures initiate below the surface of the component, and are therefore undetectable by borescope analysis. While vibration analysis may not be able to detect early, subsurface initiation of damage, we recommend that a combination of visual inspection by borescope and vibration analysis be performed in order to maximize the chance of catching progressing damage as early as possible.

Furthermore, not all bearings and gear teeth are accessible by borescope. The configuration of the gearbox (including oil level), size of the borescope instrument, and skill of the operator all influence the inspection coverage possible. For example, bearing cages may limit access to inspection of the bearing races. It is important to maximize the inspection coverage by employing competent inspectors with proper equipment and turning the drivetrain to maximize the inspection.

The AWEA O&M Working Group is developing recommended practices for borescope inspection of wind turbine gearboxes. Once those recommendations are published, we will likely recommend following those guidelines as best practice.

During regular inspection intervals, visually examine the gearbox case for cracking or other damage to verify that the case is in good condition.

A.3.3 Oil Analysis

Oil analysis is used to determine the condition of gearbox lubricant. A three-tier approach has been suggested for oil analysis (28). This approach is as follows:

1. *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 (29). ISO 81400-4 section F.5.2 contains basic analysis procedures that can be conducted on site.
2. *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.
3. *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 a problem with the gearbox. Accounts can be set up with commercial oil analysis laboratories, and measurements taken by the laboratories can be tracked by the laboratory.

In order for oil analysis to be useful as a condition monitoring method for a gearbox, multiple samples from each gearbox must be taken and tracked over time. Even if laboratory oil analysis results are tracked, however, our work indicates that unless a severe gearbox problem exists, there may not be correlation between oil analysis reports and gearbox condition. There have been multiple instances where oil condition did not show signs of degradation prior to failure of a gearbox component. For some failure modes, oil sample intervals may not be sufficiently frequent to capture oil degradation in time to assess a failing gearbox. However, oil analysis and tracking analysis results should not be discounted because of these findings. We recommend oil analysis as a regular practice because of the need to trend oil condition to help inform decisions regarding oil changes, oil dehydration, or other oil maintenance activities.

Our findings indicate that oil analysis laboratory assessment of machine condition does not detect bearing failures with the same reliability as gear failures. Furthermore, the laboratory analysis of oil samples from turbines with known bearing problems was not effective at finding the types of bearing problems experienced on the turbines.

A.3.4 Oil Particle and Debris Monitoring

For determination of oil cleanliness in accordance with ISO 4406, two primary methods of laboratory oil particle counting are commonly used, optical methods and pore blockage methods. With increased water content, opacity of samples increases, resulting in optical methods giving an incorrectly high particle count (30). Pore blockage, however, requires more time and labor, 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.

Operators report mixed experiences with oil debris monitoring. Some failure modes, such as subsurface fatigue or gear cracking, may occur without generating debris. Debris monitoring, however, is one source of information that can provide an indication of gearbox health, and it can be installed on the gearbox and provide output that is integrated in to SCADA systems.

The location of an on-line particle counter (debris monitor) will affect the results. Best practice is to locate the debris monitor directly upstream of the online oil filter.

A.3.5 Oil Filter Inspection

Industry experience with examination of the contents of oil filters is varied. Some operators report high value in cutting and examining filters, and others report no value.

If the suction ports in the oil circulation system are not located where debris accumulates, then filter analysis will not provide a complete assessment of the debris in the oil.

We recommend visual examination of oil filters when filters are changed. If visual examination indicates debris in the filter, then analysis of the debris in the filter should be conducted. The analysis can provide useful information, particularly when oil sampling best practices have not been followed.

A.3.6 Differential Pressure Monitoring

In some gearbox lubrication systems, during cold starts, a combination of filter capacity and high gearbox oil viscosity may cause a pressure drop across the filtration system that activates the in-line filter bypass circuit. When this circuit is activated, unfiltered oil is fed to the gear meshes and bearings. Clogged filters,

cold starts, or a malfunction of this circuit could result in long-term bypassing of the filter system, allowing oil unfiltered by the in-line system to be supplied to the gearbox.

We recommend monitoring differential pressure across oil filters. The differential pressure signal should be included the SCADA data stream, and should be tracked and trended.

A.3.7 Vibration Monitoring

Our work and industry experience indicates that vibration monitoring of gearboxes enables early detection of bearing and gear problems. Operator experiences and the effectiveness of different types of systems vary, and technology development on types of systems and methods of data analysis is ongoing (31). Even though the technology is evolving, we recommend using vibration monitoring on all turbines, as operator experiences (32), (33), (34), (35) and some cost-benefit analyses (36), (37), (38) have shown that vibration monitoring can reduce inspection requirements and reduce overall cost of some failure modes.

Strengths of vibration monitoring include:

1. It is a non-intrusive, on-line monitoring technique. No downtime is required during system operation.
2. In addition to being useful for detection of flaws in gearboxes, vibration monitoring systems can detect problems with main bearings and generator bearings. For example, a generator that has had electrical discharge across the bearings (grounding through the bearing) will show a peak in vibration data at twice the electrical line frequency.

Weaknesses of vibration monitoring include:

1. Some failures are subtle; systems may be less effective in detecting faults located in planetary gears and planetary bearings than in parallel-shaft gear meshes and bearings.
2. Details of the failure mode are typically not identified.
3. Trained personnel may be required to interpret the data. New systems are under development that automate data interpretation and simplify data analysis.

Many new turbines include vibration monitoring systems as standard equipment; in many cases the terminology “condition monitoring system” is used to describe the vibration monitoring system. We recommend that operators ensure that access to data from vibration-based condition monitoring systems is available post-warranty, and that trained personnel are available to interpret the data.

A.4 Performance Monitoring

Performance monitoring is generally defined as examining the performance of turbines in order to assess the condition of the turbines. Data typically examined include SCADA data and operational data such as maintenance logs.

The use of SCADA data to detect gearbox failure is a topic of ongoing industry research. Automated systems may improve this in the future, but industry experience and our research suggest that these systems are currently limited in their ability to provide early indication of gearbox failure.

Once a gearbox has reached an advanced stage of failure, SCADA data are likely to indicate changes that can be correlated to the failure. Our study found, however, that insufficient information is present in typical 10-minute SCADA data to provide early indication of gearbox failure, or even to identify existing

damage. SCADA data interpretation is useful for detection of problems with some turbine systems (39), but not necessarily for specific and incipient problems with the gearbox.

DNV publicly available specification PAS55-1:2008 establishes good practice for management of physical assets throughout the life cycle of the asset. Performance monitoring is an integral part of the asset management. We recommend following PAS55 practices, including monitoring SCADA data for detection of problems with turbine systems. As research progresses, more information may be available in SCADA data regarding specific gearbox problems, however at this time there is no indication that SCADA data can reliably be used to detect and characterize incipient gearbox failure.

A.5 Turbine Operations

A.5.1 Receiving Inspection

For new or overhauled gearboxes arriving on site, a complete inspection should be performed. This includes all checks normally performed during commissioning. Particular attention should be paid to:

1. *Turning gear.* Verify that the turning gear operated during storage and shipping. If no turning gear was used or the gear did not operate, we recommend rotating the gearbox and inspecting contacting gear meshes for standstill marks or fretting corrosion.
2. *Environmental control.* Determine the environmental limits that were set for storage and shipping, and verify that the limits were maintained and monitored. If no environmental limits were set, we recommend visual external inspection of the gearbox and borescope inspection of the gearbox for indications of corrosion.

A.5.2 Run-in Procedures for New or Overhauled Gearboxes

ISO 81400-4 section D.1 contains a run-in procedure, however, this procedure lacks detail and current knowledge regarding:

1. Stepping up load to reduce the risk of scuffing gear tooth surfaces; number, speed, and load level need to be defined
2. Correlation between oil temperature and load level
3. Oil type and additives
4. Monitoring particle generation rates during run-in, and basing load levels on particle counts and types

Furthermore, the 81400-4 standard requires the same oil type during run-in and normal operation, which is not consistent with typical practice of running in with a lubricant without additives.

NREL has identified run-in procedures as an area requiring further research and testing (40). As run-in practices vary and research on best practices is ongoing. Following (41), in addition to the process in ISO 81400-4 section D.1, we recommend the following for run-in of a new or overhauled gearbox:

1. Use oil that does not have additives, as experience and modeling have shown that anti-wear additives promote micropitting.
2. Monitor oil debris count and cleanliness and use these factors to determine the length of each load stage (42).
3. Cool and fine filter the oil during run-in.
4. Drain oil, flush, and change the oil filter after run-in.

A.5.3 Tapered Roller Bearing Preload

Tapered roller bearings are often used to support high speed shafts in wind turbine gearboxes. One race of a tapered roller bearing can be moved axially relative to the other to obtain a bearing setting (preload, zero clearance, or end play). From (18) and (43), tapered roller bearings are more sensitive to excessive preload than to end play. The general effect of preload on fatigue life is illustrated in Figure A-7. From Figure A-7, bearing life decreases rapidly as preload is increased.

Figure A-7. General Relationship between Bearing Operating Setting and Fatigue Life

From (43). Gray areas and arrows depict the region and magnitude of the loaded zone of the bearing for the ranges of settings shown.

For tapered bearings sets that are intended to be preloaded during operation, extreme care is required when replacing the bearings in order to obtain the proper bearing setting. Furthermore, if a gearbox is replaced, then the bearing types in the new gearbox may not be the same as in the old. SCADA data such as high speed shaft bearing temperatures will be different before and after the change. Comparisons between data from turbines are valid only with the same type of bearing.

A.5.5 Locked rotor

There is a risk of generating standstill marks (false brinelling) or fretting corrosion during periods when the rotor is locked. We recommend minimizing time that the rotor is locked. Turning gears should be used for storage or transportation of turbines and gearboxes.

A.6 Communities of Practice

Participation in a community of practice raises the level of the community as a whole. Wind industry organizations such as the Utility Variable Integration Group (formerly the Utility Wind Integration Group) organize useful workshops where ideas are shared. Sandia Laboratories is assembling the CREW database, which will, in part, track component reliability. AWEA has convened working groups to assemble recommended practices. DNV KEMA recommends that operators join and participate in a community of practice, and contribute information to industry databases and benchmarking studies.

Appendix B. Gearbox Failure: Definition and Causes

B.1 Definition of Gearbox Failure

From (44), “‘Failure’ is defined as the inability of any asset to do what its users want it to do.” Gearbox failures may take many forms and be of varying levels of severity. The definition of failure of a wind turbine gearbox may play a role in commercial negotiations, and a clear and consistent definition of gearbox failure may help asset managers during these negotiations.

There may be differing opinions on the definition of failure. From (18), bearing endurance is influenced by fatigue, wear, contaminants, corrosion, fretting, or mishandling, and from (45), failure of gearbox components such as bearings can occur in a wide range of ways and severity levels. Failure can be precisely defined when formulated as “functional” failures. A functional failure is the “inability of any asset to fulfill a function to a standard of performance which is acceptable to the user.” (44)

DNV KEMA recommends that a gearbox is defined as failed if any of the following conditions exist:

1. A component such as a bearing, gear, shaft, structural component, or integral seal exhibits functional failure. This does not apply to ancillary components such as hoses, pumps, fans, sensors, wiring, support bushings, or heat exchangers.
2. Gearbox temperatures cannot be maintained within normal operating limits during normal operating conditions when the cooling system is fully operational.
3. The gearbox can no longer transmit torque.
4. The mechanical efficiency of the gearbox falls below 90% of its design efficiency.

Gearboxes that exhibit one or more of the conditions above have failed. A failed gearbox may be repairable while installed in the turbine, or may require removal from the turbine to be repaired. Once repaired and returned to service, the gearbox is no longer considered failed provided that the repair is successful and the repaired gearbox does not meet the above criteria.

B.2 Causes of Gearbox Failure

From (12), the causes of gearbox failure can be grouped in to six categories, shown in Table B-1. Subcategories indicate specific causes of failure.

Table B-1. Categories for Causes of Gearbox Failure, with Subcategories Describing Specific Elements that May Cause Gearbox Failure

Category	Subcategory
Incorrect Design Parameters	Load Spectrum Environment
Improper Design	Selection of geometry Selection of materials Material processing (heat treatment) Manufacturing methods Lubrication
Improper Manufacturing	Processing Assembly Testing
Improper Installation	Mounting Couplings Alignment
Harmful Environment	Physical Chemical Electrical
Improper Operation	Testing Start-up Operation Maintenance