SERI/TP-257-3984 • UC Category: 261 • DE91002161

Variations in Gear Fatigue Life for Different Wind Turbine Braking Strategies

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Prepared for AWEA Wind Power '90
Washington, D. C.
24-28 September 1990

Solar Energy Research Institute
1617 Cole Boulevard
Golden, Colorado 80401-3393
A Division of Midwest Research Institute
Operated for the U.S. Department of Energy
under Contract No. DE-AC02-83CH10093

Prepared under Task No. WE011203

June 1991

Printed in the United States of America
Available from:
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

Price: Microfiche A01
Printed Copy A02

Codes are used for pricing all publications. The code is determined by the number of pages in the publication. Information pertaining to the pricing codes can be found in the current issue of the following publications which are generally available in most libraries: *Energy Research Abstracts (ERA)*; *Government Reports Announcements* and Index (*GRA* and I); *Scientific and Technical Abstract Reports (STAR)*; and publication NTIS-PR-360 available from NTIS at the above address.

VARIATIONS IN GEAR FATIGUE LIFE FOR DIFFERENT WIND TURBINE BRAKING STRATEGIES

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ABSTRACT

A large number of gearbox failures have occurred in the wind industry in a relatively short period, many because service loads were underestimated. High-torque transients that occur during starting and stopping are difficult to predict and may be overlooked in specifying gearbox design. Although these events comprise a small portion of total load cycles, they can be the most damaging. The severity of these loads varies dramatically with the specific configuration of the wind turbine.

The large number of failures in Danish-designed Micon 65 wind turbines prompted this investigation. The high-speed and low-speed shaft torques were measured on a two-stage helical gearbox of a single Micon 65 turbine. Transient events and normal running loads were combined statistically to obtain a typical annual load spectrum. The pitting and bending fatigue lives of the gear teeth were calculated by using Miner's rule for four different high-speed shaft brake configurations. Each braking scenario was run for both a high- and a low-turbulence normal operating load spectrum. The analysis showed increases in gear life by up to a factor of 25 when the standard high-speed shaft brake is replaced with a dynamic brake or modified with a damper.

INTRODUCTION

Historically, gearbox problems have plagued the wind power industry. Virtually all modern wind turbine designs have experienced some type of generic gearbox problem. Manufacturers of both wind turbines and gearboxes have, through hard-earned experience, developed an appreciation that a wind turbine's environment can be more severe than most other gearbox applications.

Many factors should be considered in selecting a gearbox for either a new wind turbine design or in replacing parts in an existing gearbox. Among those that should be given primary consideration are gear design, manufactured gear quality, lubrication, maintenance (or the lack of it), and load rating [1]. Each of the above factors can greatly influence the life expectancy of the gearbox. Unfortunately, a detailed discussion of each of these factors is beyond the scope of this paper. The interested reader should see reference 1 for further details. This paper examines the torque loads on a single 65-kW wind turbine gearbox. It gives a qualitative study of how gear life is affected by modifications made to the turbine's high-speed shaft mechanical braking system. Although exact numbers are given for gear life under the different scenarios presented, the reader is cautioned against using these figures for anything but a comparative analysis.

Torque loads on a wind turbine gearbox are caused by the combined action of the wind acting on the rotor, the electrical generator, and the turbine's braking system. The magnitude and coupling of these loads are quite unpredictable. The problem is more difficult because the gearbox is usually placed atop a swaying tower in a high vibrational environment, with relatively little scheduled maintenance. As a result, gearbox load requirements are difficult to specify and many gearboxes were selected without a full understanding of the true load requirements. In many cases, the peak transient loads caused by starting, stopping, or generator speed changes can be greater than the peak loads experienced during normal operation. Although these transients may comprise only a small fraction of the total operating time, they can significantly affect the lifetime of the gearbox and other components on the wind turbine. Moreover, understanding and controlling these transient loads is an an effective way to extend the life of a wind turbine without negatively affecting its productivity.

In a wind farm with more than 200 Micon 65 wind turbines, more than 20% experienced complete gearbox failures in an average of 10,000 hours of operation. Of the same group, an additional 40% were in severely degraded, but operable, condition with such failures as chipped teeth and significant axial shaft play. In another field of 150, 15% failed and 25% were damaged but operable after 2.5 years of operation [2]. Wind farm operators have been seeking solutions to these problems for years.

The primary objective of this analysis was to determine how the transient torque loads resulting from high-speed shaft brake applications affect the life expectancy of a 65-kW wind turbine gearbox. Four different braking strategies were measured and compared in this analysis under two different wind turbulence conditions. A single test turbine was used for measuring transient events and normal operating loads.

TEST TURBINE DESCRIPTION

Turbine Description The test unit was a Micon 65-kW three-bladed, Danish-designed wind turbine. The 7.5-meter blades were made from fiber-reinforced polyester resins and equipped with pitching tips for centrifugal overspeed control. The blades connect to a ridged, fixed pitch hub, which rotates at a synchronous speed of 48 rpm. Peak power from the rotor is regulated by aerodynamic stall, which is designed to flatten the power curve near 65 kW in high winds. With clean blades, the actual power is regulated closer to 80 kW as shown in Figure 1.

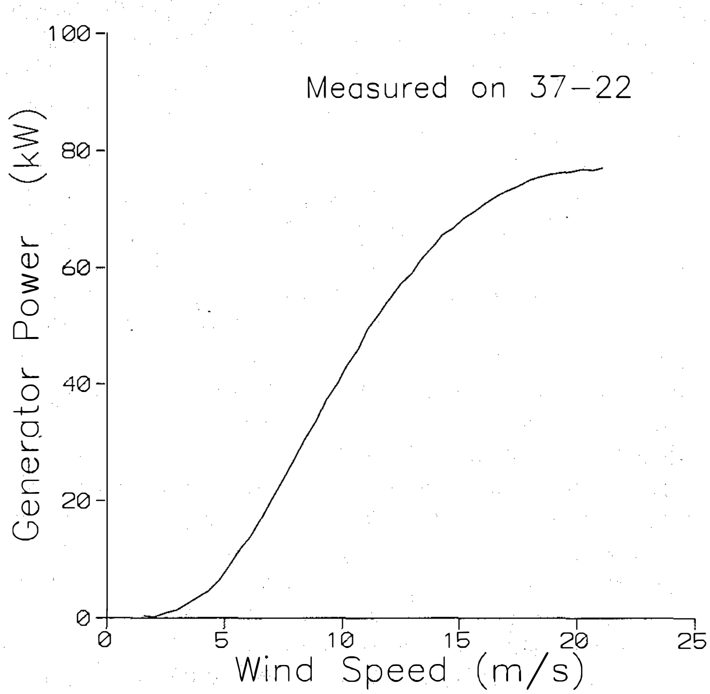

Figure 1. Micon 65 Power Curve

The low-speed shaft (LSS) enters a two-stage, parallel shaft, helical gearbox with an output speed of about 1200 rpm. The nacelle sits on a freestanding steel tube tower and is held upwind by a yaw drive. Electric power is generated by one of two 480 VAC induction generators in the nacelle. The small generator has a 13-kW capacity and operates until the winds exceed about 9 m/s. Above this level the turbine switches to the large generator rated at 65 kW.

Brake Description The factory-provided brake (undamped mechanical brake) on the test turbine is a spring applied, mechanical caliper electrically held off from a 590-mm-diameter disc that is mounted to the gearbox high-speed shaft (HSS). The brake is applied when the controller senses a fault, when the machine is manually shut down, or during a loss of line condition. The mean mechanical torque applied at the HSS axis was measured to be approximately 1288 Nm (950 ft-lb). Typical stopping time for this brake is 1.3 seconds or less than two rotor revolutions. Wind turbine operators believed that these high torque levels imparted to the drive train may be damaging the machine.

To reduce the peak braking loads, a damper was added as a retrofit (damped mechanical brake) to delay the full application of the brake. Since the typical stopping time for the damped brake is 4.5 seconds, the pads absorb more energy during the braking process, which could diminish the life of the brake.

The Dynamic Brake, developed by Second Wind Inc., uses a transfer switch to disconnect the generator from the utility and apply a resistor–capacitor network to the still-excited windings. A resonant circuit is created and decayed through the resistors. This action delivers a variable 575 or 970 Nm (424 or 715 ft-lb) brake at the generator shaft. Braking torque decreases with speed until a constant speed or hover mode is attained at approximately 9 rpm at the rotor. The hover mode can be maintained indefinitely. In these tests, the mechanical brake was applied after the Dynamic Brake attained the hover mode.

TEST DESCRIPTION

Test Turbine The test turbine was a fully instrumented test bed located in the San Gorgonio Pass near Palm Springs, California. Personnel at SeaWest and the Solar Energy Research Institute (SERI) installed the instrumentation. These two groups have been working together to test a new blade design developed by SERI and Phoenix Industries. A side-by-side comparison of the new and old blade designs was conducted using these turbines. For exact details about this test and the data acquisition system used, the reader is referred to reports by J. Tangler [3,4]. The gearbox testing for this project was carried out on the standard configuration or "benchmark" turbine.

Normal Operating Tests Testing began during the 1989 wind season. The HSS from the gearbox was removed and instrumented to allow strain gages to be placed on the shaft and on the teeth of the high-speed pinion. Unfortunately, none of the gear teeth gages survived the initial spin up. The HSS gages were not working while most of the long-term operating data were being collected, but they were repaired when the transient load measurements were made.

All of the normal operating data was taken from 14 hours of LSS torque data collected during the blade testing project. The analog signals were multiplexed and recorded onto three analog tapes using a 14-channel Honeywell 101 tape recorder. The data used was collected during three days of continuous operation.

Brake Event Tests A separate, isolated data acquisition system was used to perform the brake torque measurements. The main objective of the brake configuration testing was for research and development of the Dynamic Brake by Second Wind.

Stepped down voltages and shunted currents were collected along with HSS torque, wind speed, rotor speed, and turbine status. Data were acquired with Labtech Notebook on a Toshiba T3200 portable 286-IBM-compatible PC with a Metrabyte Dash 20 analog/digital conversion card. The signals were sampled at 1000 Hz for all braking events.

The standard damped mechanical brake was installed on the machine. The damper was physically disconnected to allow for testing of the undamped mechanical brake configuration. For its tests, the controller was modified to allow the Dynamic Brake to apply while the normal brake was electrically held off. The small generator was also disconnected to eliminate the possibility of a "downshift" during the braking process.

The brakes were manually applied and data were collected during the process. The data sets for these events lasted between 10 and 20 seconds. A data set was defined as the period from initiation of the braking event to full stop of the rotor. The data sets chosen for the analysis were brake applications when the turbine was operating at power levels ranging between 30 and 38 kW. A typical time series for each brake configuration is shown in Figure 2.

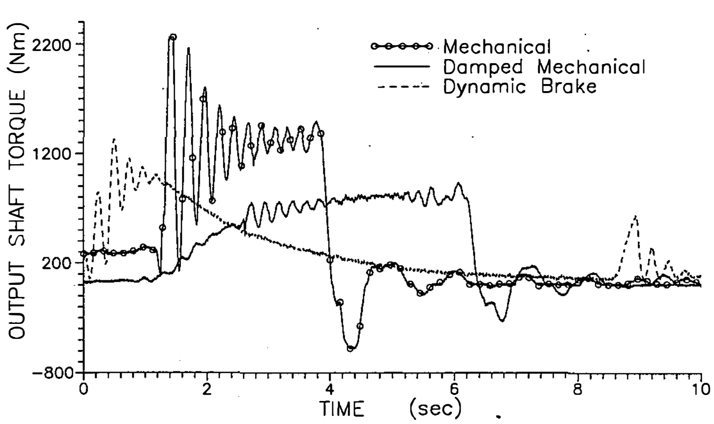

Figure 2. Braking Torque Time Series

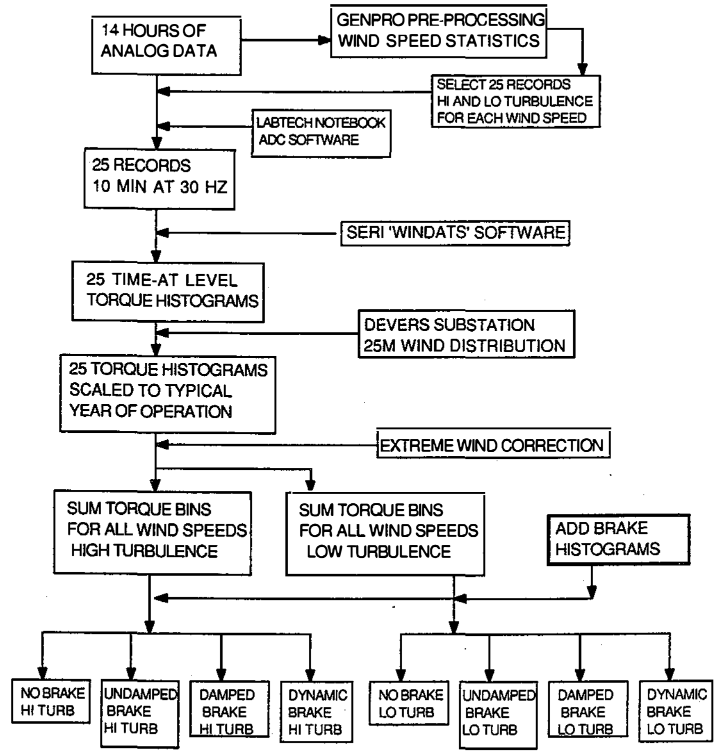

Figure 3. Data Reduction Path

GEARBOX TORQUE SPECTRUM ANALYSIS

A two-part approach was taken for the data reduction and analysis. First, the raw data from the normal operating loads and the transient loads were processed to construct eight different gearbox torque spectra. These torque spectra were then used as inputs, along with the specifics of the gearbox geometry, to a custom gear analysis code, AGMA218, developed by GEARTECH. This section describes the data reduction process for the torque spectra. The fatigue analysis is described in the following section.

Normal Operating Data The data reduction sequence for normal operating loads is schematically represented in Figure 3. The first step of data reduction involved pre-processing the 14 hours of normal running data in bulk. Using the GENPRO processor at SERI, each tape was divided into contiguous 10-minute records and only the wind speed statistics were computed. Wind speed means were binned at 1-m/s intervals, and the bin center wind speed was plotted against its corresponding turbulence intensity (standard deviation divided by wind speed mean). This plot is shown in Figure 4. For each wind speed bin, the 10-minute records with the highest and the lowest turbulence intensity were selected to be digitized. Twenty-five 10-minute records were selected. The torque data were digitized from analog tape at 30 Hz along with generator power to help validate the torque data. For a range of bin center wind speeds from 4.5 m/s to 15.5 m/s, a high- and a low-turbulence data set were created.

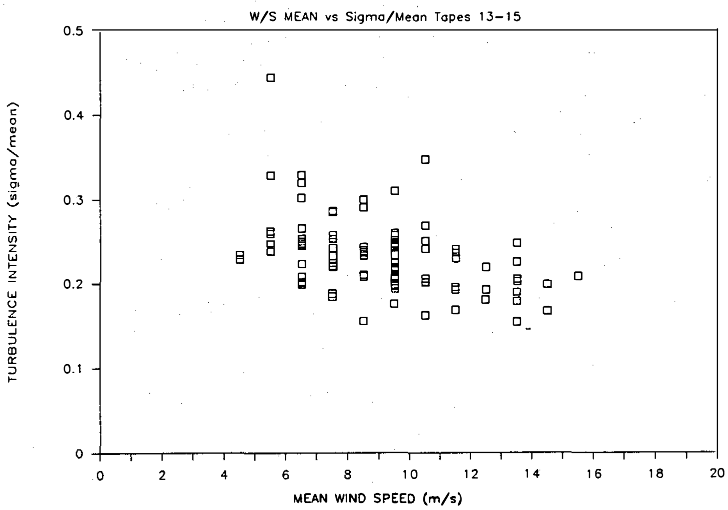

Figure 4. Wind Speed versus Turbulence Intensity

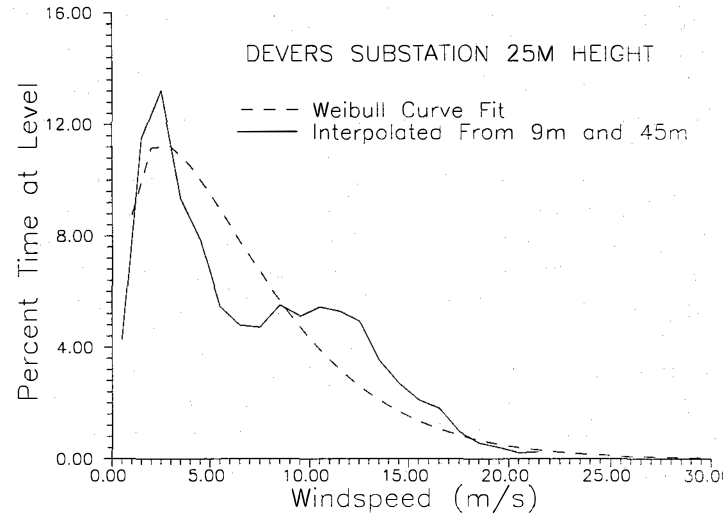

Figure 5. Devers Wind Distribution

For each digitized record, a time-at-level histogram of low-speed shaft torque was generated using another SERI data reduction program, WINDATS. This created 25 histograms, each corresponding to a particular wind speed average.

One of the major assumptions made was that the shape of each of these histograms was representative of the torque distribution seen by these gearboxes for any time period having the same mean wind speed and turbulence intensity. In other words, it was assumed that the shape of the distributions would not change when the total time was increased to 1 hour. Given this, the time at each torque level within the torque histograms was scaled by the number of hours per year in the corresponding wind speed bin of a 25-m wind distribution at Southern California Edison's Devers Substation in San Gorgonio Pass.

The Devers Substation wind distribution was chosen because it was the most extensive data set available. The wind distribution was collected by Battelle Pacific Northwest Laboratories [5] over a period of five years, and was representative of a good high wind site near the test turbine. Data collected at 9.1 m and 45.7 m were linearly interpolated to create the 25-m (hub height) wind distribution. This distribution is shown in Figure 5. Its bimodal shape is probably due to seasonal characteristics of the wind at that site. A Weibull curve fit of the 25-m data was included as a comparison to a true Weibull shape. The analysis was done for both the actual 25-m data and the Weibull fit to this data. However, only the results using the 25-m data are presented in this report, because the spectra developed using the Weibull curve fit data were less conservative.

Scaling these distributions creates larger histograms for each wind speed bin with the same shape as the original distribution. Next, for all wind speeds, the number of hours in each torque bin were summed for each torque load time-at-level histogram to develop the final distribution of normal operating loads. This was done for both the high- and low-turbulence cases.

The effect of extreme wind speeds was accounted for by assuming that torque levels due to winds in excess of the highest data set (15.5 m/s) would be sustained at the 15.5-m/s level. Therefore, the highest wind speed torque histogram was used for all wind speeds above 15.5 m/s, which increased the time-at-level contribution of the highest data set by a factor of approximately 3. This approach gives a less conservative load spectrum than the actual load spectrum, but is probably a good model, since on this wind turbine torque is self-regulated at high wind speeds by aerodynamic stall. In reality, the average power output of the turbine does exceed the 65-kW rating by 10% to 20% when the blades are clean. A better analysis would model the high wind torque levels as a function of the actual power generated. This model will be run in a future study.

Figures 6 and 7 show the final torque load distributions for high- and low-turbulence operating loads. For comparison, the corrected and uncorrected data are plotted with the data derived from a Weibull curve fit of the actual wind distribution. It can be seen from this data that the bimodal shape of these distributions is only partially explained by the bimodal wind distribution shape, since even the Weibull-derived data are bimodal. Apparently, the effect of stall regulation at high wind speeds is the dominant reason for the double hump of the normal operating torque distribution.

Transient Event Data A description of the event testing and data processing follows. Included is a description of the process used to count brake events and extrapolate that to a typical year.

The WINDATS software was used to apply the method-of-bins to generate a HSS torque histogram for each brake configuration. The torque was transformed to the LSS and the time was normalized to hours, assuming the torque sample remained constant during the sample period. For an average yearly torque histogram, the time in each bin was multiplied by the number of events that would occur in a typical

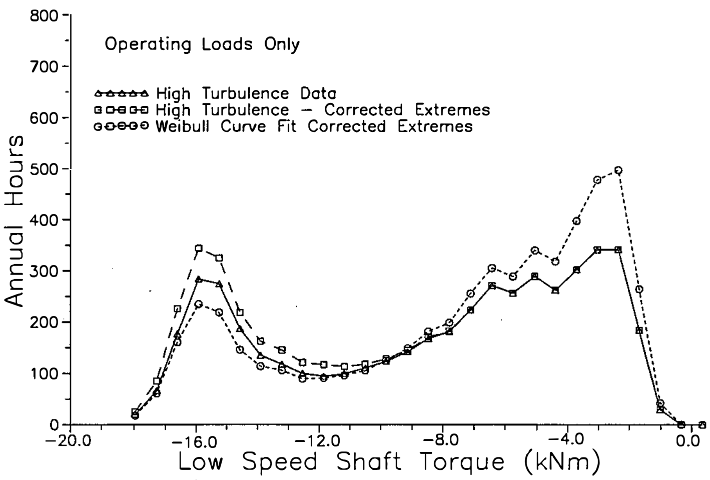

Figure 6. High-Turbulence Torque Loads

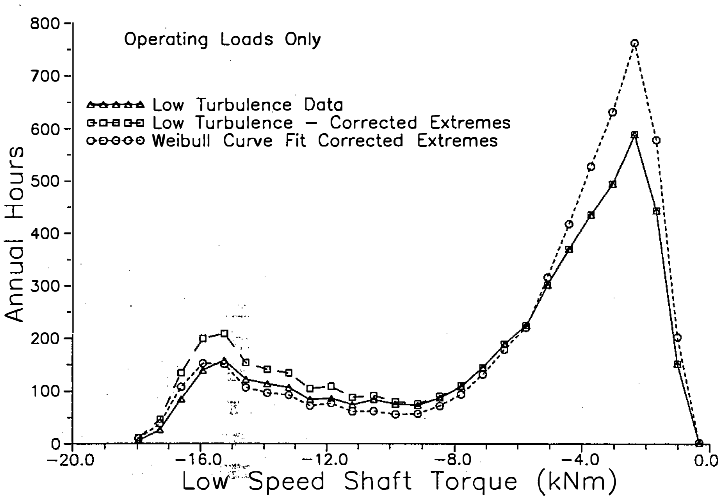

Figure 7. Low-Turbulence Torque Distribution

year. The process for determining this number is described below.

A Second Wind central turbine monitoring system was used to count events, such as yawing, braking, start-ups, and generator shifts, and average other inputs, such as power, wind speed, wind direction, and on-line time. More than 50 Micon 65 turbines were monitored in the San Gorgonio Pass of California between October 1989 and May 1990. Custom data base management software was used to process the data. The information was first filtered for bad or incomplete data. Next, 10-minute averages of wind speed were binned with the corresponding number of brake events counted in that 10-minute period. This was done for all data collected, giving a six-month relationship between wind speed and number of braking events for all of the turbines. All units with over 90% data recovery were averaged, and using the Devers wind distribution, the average unit was extrapolated to a full year's operation. For the wind plant monitored, 112 brake cycles per year were found to be representative. It should be noted that the units monitored were well-maintained turbines at a high wind location.

The array of wind speed versus braking events was scaled to a full year using the Devers site wind distribution described in the previous section. Each wind speed bin's brake cycle count was multiplied by the ratio of that bin's hours at Devers to the corresponding actually monitored hours of the turbines.

Combined Torque Spectra Each braking scenario was created by combining the transient brake loads with a normal operating load spectrum. The low-turbulence and high-turbulence spectrum were added to the torque spectrum for each brake configuration to give eight composite torque spectra, each describing a different operating strategy. High- and low-turbulence cases are included to illustrate the sensitivity of fatigue life to an operational parameter other than the transient events described above. Each spectrum is described in Table 1.

Table 1. Wind Load Spectra

No.	Brake Description
1.	High-turbulence undamped mechanical
2.	Low-turbulence undamped mechanical
3.	High-turbulence damped mechanical
4.	Low-turbulence damped mechanical
5.	High-turbulence dynamic brake
6.	Low-turbulence dynamic brake
7.	High-turbulence without HSS brake
8.	Low-turbulence without HSS brake

GEAR FATIGUE LIFE ANALYSIS

The gear life analysis is a qualitative comparison of the effects of different brake systems and the influence of wind turbulence on gear fatigue life. The conclusions regarding gear life are generally applicable to any wind turbine gearbox. However, the suitability or performance of any specific gear set for a particular application should not be judged solely on this analysis.

Gearbox Description The gearbox used for the gear life calculations is a typical wind turbine speed increaser, with parallel shafts and two stages of single-helical gears. The gear teeth are carburized, hardened, and ground. The low-speed gear set increases the rotor speed from 48 rpm to an intermediate-speed shaft of 216 rpm. The high-speed gear set increases that speed to the generator speed of 1238 rpm.

Computer Program The AGMA218 computer program [6] is based on the American Gear Manufacturers Association (AGMA) standard for gearbox rating [7]. It performs either life ratings or power ratings of spur or helical gears. Given the transmitted power and speed, the tooth pitting and bending fatigue lives can be calculated for a single load and speed or for a spectrum of loads, with the resultant life determined from Miner's rule [8].

The program does not consider the effect of torque loading on other gearbox components such as shafts and bearings. Analysis of these components cannot be done with a simple time-at-level torque distribution. A more sophisticated cycle-counting algorithm would be needed in order to properly predict the fatigue life of these components.

Analysis Assumptions The calculated gear lives are contingent on the following constraints and limitations of the fatigue analysis:

- Loads do not exceed those shown in the wind load spectra.
- Metallurgical quality of the gears conforms to at least AGMA grade 2 requirements.
- Gear accuracy conforms to at least AGMA quality no. 11.
- Gear mesh alignment under load gives full-face contact consistent with a maximum load distribution factor, Cm, of 1.2.
- Gear tooth stresses remain within the elastic limit. Stresses due to gear tooth impact are not considered.

Wind Load Spectra Some of the wind load spectra were reduced by averaging or condensing data points. The times corresponding to the eliminated loads were added to higher loads, resulting in a slightly more conservative load spectra.

Figure 8 compares the brakeless high- and low-turbulence spectra. The high-turbulence spectrum is more damaging to gear life because it has a greater number of loads in the range of -5.15 to -19 kNm (-3.8 to -14 ft-kps) than the low-turbulence spectrum. The low-turbulence spectrum has a greater number of loads below -5.15 kNm (-3.8 ft-kps), but they can be ignored due to their relatively insignificant contribution to fatigue damage.

DISCUSSION OF RESULTS

The fatigue life and stress level of each gear interface were predicted for each of the eight torque load spectra in Table 1. The Hertzian contact stresses, which relate to pitting wear, and the tooth bending stresses were calculated for each case. Results show that the low-speed gear set has less load capacity than the high-speed gear set, and the low-speed pinion has the shortest life [9]. Due to the volume of data, only the results for the low-speed pinion gear are presented in this report.

The pitting and bending fatigue lives for each brake type and turbulence level are presented in Figures 9 and 10, respectively. Figure 9 shows the reduction in the pitting fatigue life for the low-speed pinion when the different braking configurations are substituted. Similarly, Figure 10 shows the reduction in the gear tooth bending fatigue life.

The influence of wind turbulence is best seen by comparing the gear lives shown in Figures 9 and 10, which give the results for high- and low-turbulence spectra without brake loads (spectra 7

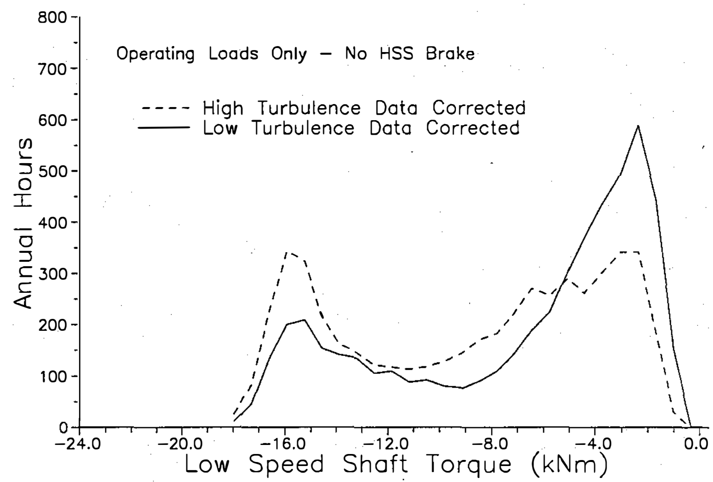

Figure 8. High- and Low-Turbulence Spectra Comparison

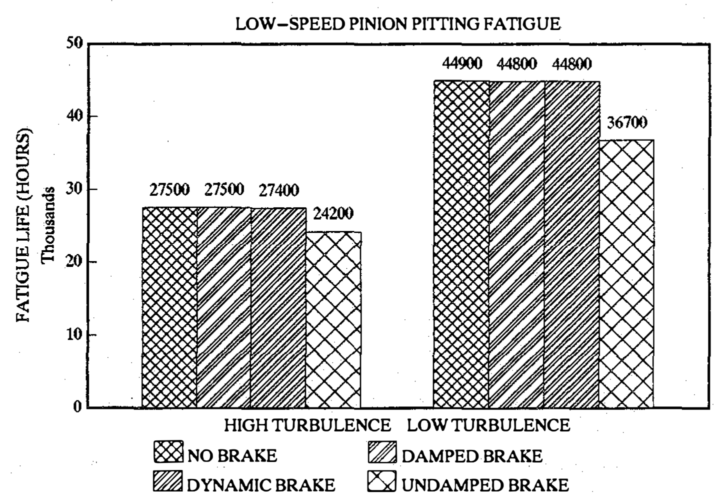

Figure 9. Gear Tooth Pitting Fatigue Life

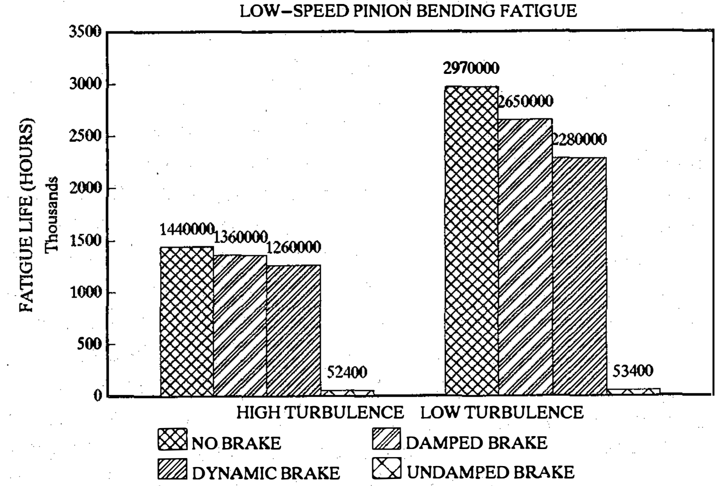

Figure 10. Tooth Bending Fatigue Life

and 8). At low turbulence, the low-speed pinion pitting life increases by a factor of 1.63 over the high-turbulence case. The low-turbulence bending fatigue life for the same gear increases by a factor of 2.06 over the high-turbulence spectrum.

These figures also show that the undamped mechanical brake creates very high stresses and results in the shortest gear lives. The Hertzian stress on the low-speed gear set is high enough to cause permanent plastic deformation in the subsurface layers of the pinion and gear teeth. Plastic deformation is detrimental to carburized gear teeth because it reduces beneficial, compressive residual stresses. The actual gear lives are likely to be less than those shown because the AGMA218 program does not account for the effects of plastic deformation.

As mentioned, the undamped mechanical brakes stop a wind turbine abruptly. In such severe stops, and even in the damped version, some of the kinetic energy is stored momentarily as elastic strain energy in the gears, shafts, and couplings. After the rotor has stopped, the strain energy is released when these drive train elements torsionally unwind. During this period, the gear teeth unload, travel through the backlash, and impact on their back sides. They then rebound, travel through the backlash, and impact on their front sides. These rapid torque reversals may be repeated several times while the transient vibration decays. Impact may cause very high stresses on the gear teeth. Again, gear life is probably overestimated for both mechanical brakes by the model due to the inability of the present analysis to include these torque reversals. The impact stresses could be determined in a future study with a nonlinear, dynamic analysis and/or actual measurement.

The results show that the undamped mechanical brake in high-turbulence spectrum 1 reduces the pitting life of the low-speed gear set to 88% of the no brake case (spectrum 7). With a low-turbulence spectrum, the undamped mechanical brake (spectrum 2) reduces the pitting life of the low-speed gear set to 82% of the no brake case (spectrum 8). The brake is more damaging with a low-turbulence wind spectrum because the brake loads are higher relative to the wind loads and therefore, the brake loads have greater influence on the gear lives. However, the damped mechanical and dynamic brakes do not decrease the pitting lives. Their effects on pitting lives are the same as those using no high-speed brake.

In contrast, the bending fatigue life is sensitive to peak loads, and all the brakes cause some decrease in the bending fatigue life. The results show that the undamped mechanical brake reduces the bending fatigue life of the gears by orders of magnitude. Compared with the standard undamped mechanical brake, the damped mechanical brake increases the bending fatigue life of the low-speed gear set by a factor of between 26 and 49.5, depending on the turbulence. In a similar comparison to the undamped brake, the dynamic brake increases the bending fatigue life of the low-speed gear set by a factor of between 24 and 42.7. Tables 2 and 3 summarize the effect in the low-speed pinion gear lives caused by each of the brakes.

Table 2. Summary of Low-Speed Pinion Lives for High-Turbulence Load Spectra

Spectrum No.	Hertz Stress Sc(psi)	Hertz Life Lc (hr)	Bending Stress St(psi)	Bending Life Lt(hr)
1	360795	24200	213224	52400
3	219629	27500	79012	1.36E6
5	258337	27400	109316	1.26E6
7	200765	27500	66022	1.44E6

Table 3. Summary of Low-Speed Pinion Lives for High-Turbulence Load Spectra

Spectrum No.	Hertz Stress Sc(psi)	Hertz Life Lc (hr)	Bending Stress St(psi)	Bending Life Lt(hr)
2	360795	36700	213224	53400
4	219629	44800	79012	2.65E6
6	258337	44800	109316	2.28E6
8	200765	44900	66022	2.97E6

CONCLUSIONS

The undamped mechanical brake creates very high stresses and the shortest predicted gear lives. It should not be used.

The damped mechanical and dynamic brakes are significantly more benign, and they increase gear tooth bending life by as much as a factor of 25. The effect of these brakes on gear tooth pitting is much better than the undamped brake and as good as using no high-speed shaft brake.

The model overpredicts gear life. Some of the reasons for this are as follows:

- Undamped and damped brakes experience torque reversals in the transient decay at the end of the brake cycle. The effect of these rapid strikes on gear life is suspected to be large.
- The undamped brake causes gear stresses outside the elastic range of the materials.
- It assumes near-perfect gearbox manufacture, assembly, and maintenance.

While the brake loads are the most significant contributor to gear damage, the influence of turbulence can reduce gear life by up to a factor of 2.

RECOMMENDATIONS

It is recommended that undamped mechanical brakes not be used because they cause high stresses on the gear teeth, plastic deformation of the subsurface layers of the gear teeth, and gear tooth impact. Gear life can be extended by replacing undamped mechanical brakes with either damped mechanical or dynamic brakes. Improved maintenance, a modified controller, or both are recommended to reduce actual brake applications.

It may also be possible to extend gear life by replacing a high-speed shaft brake with a low-speed shaft brake. However, during a stop using a low-speed shaft brake, the kinetic energy of the generator will cause loads on the gear teeth. Research is required to determine the gear tooth loads associated with low-speed shaft brakes.

Further study is needed to investigate the effect of gear tooth impact during a braking event by actual measurement, a nonlinear dynamic analysis, or both. It is also recommended that a complete cycle-counting fatigue analysis be carried out on a larger base of data that includes the loading of other gearbox components.

ACKNOWLEDGEMENTS

Many people have contributed either directly or indirectly to this report. The authors would like to thank SeaWest for their cooperation and support throughout the testing and analysis, specifically Gary Dodak and Mark Haller. In addition, the contributions made by Pat Quinlan, Brian Smith, Jane Muller, Sandy Butterfield, and Neils LaWhite were essential and are greatly appreciated. Finally, the United States Department of Energy is credited for its support of the wind turbine test site where this work was performed.

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