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Low Wind Speed Turbine Developments in Convoloid Gearing

Final Technical Report
June 2005 – October 2008

Genesis Partners, LP
Montgomeryville, Pennsylvania

Subcontract Report
NREL/SR-500-45103
August 2010

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NREL Technical Monitor: Scott Schreck
Prepared under Subcontract No. ZAM-3-33200-12

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List of Acronyms

AGMA	American Gear Manufacturers Association
B&S	Brown & Sharpe
CMM	coordinate measuring machine
COE	cost of energy
CV	constant velocity
FCR	fixed charge rate
FEA	finite-element analysis
GB	gearbox
HS	high speed
IP	intellectual portfolio
LS	low speed
LVR	load distribution analysis
LWST	low wind speed turbine
O&M	operations and maintenance
PMM	precision measuring machine
VFD	variable frequency drive
WT	wind turbine
WTG	wind turbine generator

Executive Summary

Introduction

This report presents the results of a study conducted by Genesis Partners LP as part of the United States Department of Energy Wind Energy Research Program to develop wind technology that will enable wind systems to compete in regions having low wind speeds. The purpose of the program is to reduce the cost of electricity from large wind systems in areas having Class 4 winds to 3 cents per kWh for onshore systems or 5 cents per kWh for offshore systems. This work builds upon previous activities under the WindPACT project, the Next Generation Turbine project, and Phase I of the Low Wind Speed Turbine (LWST) project. This project is concerned with the development of more cost-effective gearing for speed increasers for wind turbines.

Approach

The project statement of work required the refinement in the proposed technical approach to achieving the LWST cost of energy (COE) objective, focusing on the development the Convoloid gear shape—a novel new design approach for gear teeth. (See Appendix V) This new design approach is expected to provide gears that will have increased load-carrying capacity within the same gear envelope of the existing gear design.

Results

The project statement of work was to test Convoloid gearing in a 4-square test rig based on 108-kW wind turbine gearboxes. For the "slave gearbox," optimized involute gearing was manufactured and assembled into the 108-kW housing. Convoloid gearing of the same materials, ratio, bearings, and face widths as the involute design (only change was in the tooth form) were assembled into the "test gearbox."

After 942 hours of testing, the Convoloid gearbox suffered a major bearing failure (spherical roller bearings) on the intermediate shaft. Comprehensive analysis showed that reconditioning both boxes would be time consuming and expensive when considered in the context of the contract constraints. After consulting with NREL personnel, the decision was made to scale down the size and increase the cost efficiency of testing. The "3-Hole" Test project then was initiated on May 18, 2006. It included the objectives of lower required test cycles, potential "24/7" (24 hours per day, 7 days per week) test-rig operation, increased numbers of gear-pair specimens (to enhance the statistical surety of the results), greatly increased efficiencies in gearbox change-outs, and multiplicity of test results. Test results for the NEG108 Micon testing are found in Section 4 and those of the 3-hole testing in Section 7.

Key Findings

1. A proven method of adapting the non-involute Convoloid tooth form coordinate system to the industry's CNC gear tooth grinders was found using the established machine software oriented only to involute forms. This advancement makes the economical production of Convoloid forms with the world gear industry's present capital asset infrastructure possible and does it with ease.

2. The tooth tip relief protocols for Convoloid were developed before the tests and applied to the Convoloid gears. The low speed, high torque gears were perfectly relieved but the high speed lower torque gears had too much relief due to the high tooth stiffness of the Convoloid design. Tip relief protocols were altered accordingly to match the test results and have been successfully used since.

3. The Convoloid forms developed micropitting at the entry and exit to the transition zone. New protocols have been developed to preclude this occurrence in future designs.

4. At the conclusion of the tests at 965 hours of operation with substantial segments of the tests run at over 4 times rated loads, the Convoloid micropitting had steadied out whereas the involute gears showed a steady migration of the micropitting phenomenon. This fact proved to the test observers that Convoloid gearing will be far superior to involute gearing of the same size, material, face width, and center distance.

5. Using only NREL COE protocols it was determined that (paragraph 9.5 of the Final Report)--"The energy capture of the cost neutral Convoloid gear box/blade diameter increase typified by alternative #3 using Class 3 winds (5.539 m/s) equals the energy capture of the classical involute Baseline design using Class 4 winds (5.8 m/s). This result exactly matches original NREL LWST Program Objectives as directed: to improve energy capture of the wind resources found in the Midwestern United States."

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1. Introduction

1.1. Background

1.1.1. Low Wind Speed Technology

The U.S. Department of Energy Wind Energy Research Program has begun to develop wind technology that will enable wind systems to compete in regions that experience low wind speeds. The sites targeted by this effort have annual average wind speeds of 5.8 m/s, measured at 10-m height. Such sites are abundant in the United States, and would increase by twenty-fold the available land area which can be economically developed. The stated program goal is to reduce the cost of electricity from large wind systems in Class 4 winds to 3 cents per kWh for onshore systems or 5 cents per kWh for offshore systems, by the year 2012. A three-element approach has been initiated and consists of (1) concept design, (2) component development, and (3) system development. This work builds upon previous activities under the WindPACT project, the Next Generation Turbine project, and Phase I of the Low Wind Speed Turbine (LWST) project. If successful, DOE estimates that this new technology could result in 35 GW to 45 GW of additional wind capacity being installed by 2020.

1.1.1.1 Concept Design

Convoloid gear shapes are a novel new design approach that are expected to provide gears that have increased load-carrying capacity but which fit within the same gear envelope of an existing gear. (See Appendix V.)This final report includes results from the 100-kW scale testing and comparisons to predicted results for a 100-kW design. It describes the design studies conducted, including evaluation of the baseline 100-kW gear design loads, predicted lifetime, and other appropriate operating parameters. It includes a description of the new gear 100-kW design and predicted performance. The report also describes the design studies conducted to scale the 100-kW subscale testing and analysis to a MW-class gearbox design, including evaluation of baseline MW-class gear-design loads, predicted lifetime, and other appropriate operating parameters, and a comparison to the new full-scale Convoloid design. It explains how the proposed full-scale concept is expected to achieve the COE objective for a MW-class system and describes additional phases necessary to bring the proposed technology to market. The subcontractor will conduct a final review meeting at which the contents of the final report are presented to NREL reviewers.

1.1.1.2 Component Development

The component development element required the subcontractor to examine and refine the proposed technical approach to achieving the LWST cost of energy (COE) objective. The focus of this effort is the development of more cost-effective gearing for speed increasers in wind turbines. The principal subcontract deliverable is this final report describing the testing and scale-up engineering of a Convoloid gear set for a sub-megawatt-scale gearbox for an existing wind turbine. The work consists of detailed shop and field testing of a standard 100-kW scale gearbox using involute gears and a modified 100-kW scale gearbox using a new Convoloid gear set to accurately determine and compare the tooth stresses and operating conditions. Based on the test results, the next step is to design a replacement gear set for a larger wind turbine gearbox design. The larger gearbox will be 500-kW or greater and be representative of designs used in modern megawatt-class machines.

1.1.1.3 System Development

In June 2005, Genesis Partners LP was awarded contract number ZAM-5-33200-12 under the U.S. Department of Energy Wind Energy Research Program, to examine and develop more cost-effective Convoloid gearing and then relate these savings to LWST COE.

1.1.2. Organization

The organization of the project and its personnel is shown in Figure 1.1.

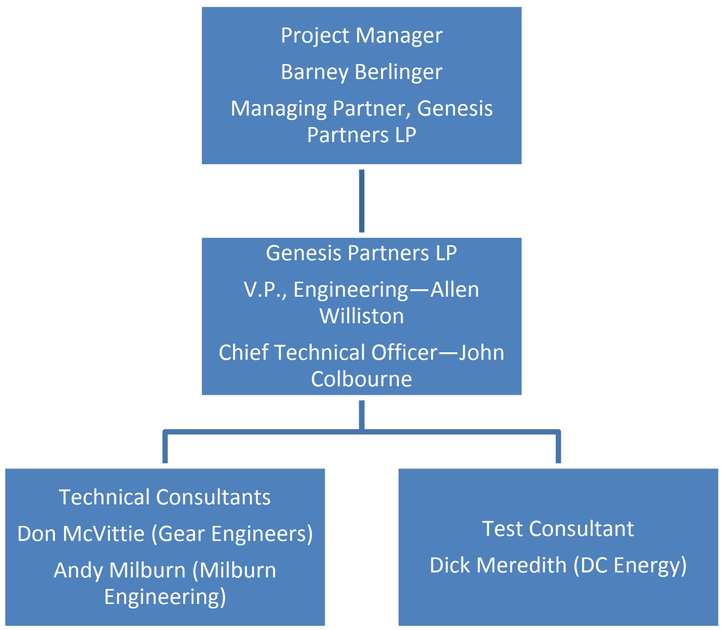

Figure 1.1. Organization of project personnel

1.2. Objectives

The objective of this study is to develop a Convoloid gear design and to demonstrate its potential to reduce the size and weight of gearboxes compared to a baseline involute gear design. This improved gear design is quantified in terms of a decrease in capital cost, reduction in gearbox size, and potential increase in performance or efficiency.

1.3. Scope

This report describes the approach and rationale used to reach the objectives of the study and the organization of the work between Genesis and its subcontractors. It presents the work plan that was followed and describes the results of each task. Detailed results of all aspects of the calculations of the cost of energy are presented, together with explanations of the appraisal system used to select the most promising configurations. A set of optimized configurations is identified and compared with the initial baseline design.

2. Methodology—Common to Both Micon 108 and 3-Hole Tests

2.1. Work Plan

The project was divided into six tasks that reflect the objectives and organization.

- Task 1 involved preparation of comparisons of an enhanced Convoloid gearbox with the existing involute-based gearbox for a 108-kW wind turbine. These comparisons addressed stresses and operation and maintenance (O&M) costs as well as COE reductions between the two. The task led to a kick-off meeting at which Genesis presented its intended approach to NREL.
- Task 2 focused on specific aspects of the design process leading to testing. Investigations into manufacturing methods and sensitivities, gearing inspection, definition of gear failure, and component analysis lead directly to the test load values as defined in the test protocol.
- Task 3 involved executing the test protocol for involute and Convoloid gear sets in two different test stands.
- Task 4 performed a detailed analysis.
- Task 5 delineated field testing; the enhancement of the testing in Task 3 precluded field testing for this phase of the project.
- Task 6 created a detailed design of scaled-up gearbox using Convoloid technology and COE savings estimates.

2.1.1. Procedure

The project goal was to develop a Convoloid gear design and to demonstrate its potential to reduce the size and weight of gearboxes compared to a baseline involute gear design. This improved gear design resulted in decreased capital costs through a reduction in size while potentially increasing performance and gearbox life. Development of an appropriate gear design encompassed consideration of traditional gear design and rating factors as well as manufacturing methods. Further, a method for checking the gear teeth of a manufactured part was needed to insure that the tested geometry was true to theory. Before testing began, test criteria for load and failure definitions was established and an analysis of all gearbox components was performed to assure that no unexpected failures occurred.

Initial testing involved using the existing configuration for two Micon 108 gearboxes and replacing the gearing with optimized involute and Convoloid gearing of the same ratio. Subsequent testing followed with a “3-Hole Test” that was not dependent upon the involute and Convoloid gearing having the same ratio. An optimized design of the involute gearing was prepared by Don McVittie (Gear Engineers). The Convoloid design was developed through an algorithm developed by John Colbourne (Genesis Partners). Analysis of all test components at the anticipated test loads was necessary to assure survival during the test life span. A summary of Mr. McVittie’s recommendations is included as Appendix Q.

The gearboxes were mounted to a back-to-back test stand (Figure 2.1.) that allowed a great amount of torque to be circulated through the gearboxes yet minimized the true power usage to only the amount dissipated by system inefficiencies. This method of torque application ensures that both gearboxes are subject to the same load.

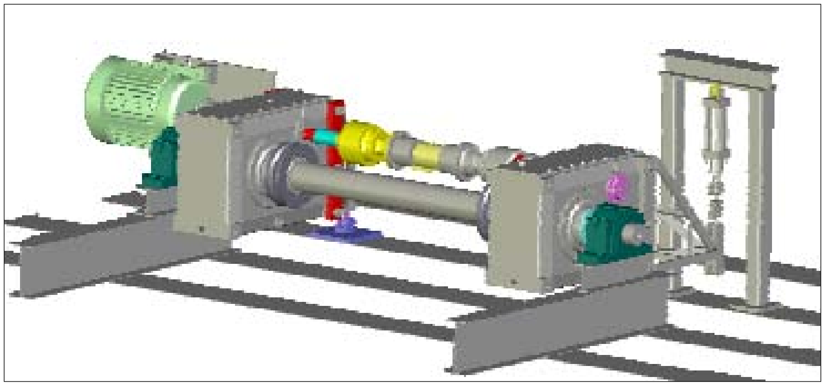

Figure 2.1. Micon 108 back-to-back test stand arrangement

The realm of operation for wind turbine gearboxes involves extended operation, therefore, a fatigue failure mode was desired during testing. Proper selection of the applied torque was necessary to assure a contact fatigue failure on the gearing. If the applied load is too great, then tooth breakage could result—potentially damaging the tested component as well as the test equipment. If the load is too small, then the time-to-failure of the test is prohibitive. It was necessary to find a balance between these extremes.

It also was important to constrain as many variables not related to the gearing geometry as possible. These constraints included, in part: gearing material and heat treatment, loads, and oil. The lubrication is of key importance. A high test load dictated that an assured supply of cool oil was available for the bearings and gears. Cross-contamination also must be prevented. Therefore, each gearbox was supported by its own oil reservoir, pump, filter, and heat exchanger. Instrumentation for both gearboxes was similar and included bearing and lubricant temperatures at various points, lubricant cleanliness, and vibrations in three axes. Additional signals tracked circulated torque, motor amperage, and speed.

Modifications to the Convoloid tooth profile were expected to account for the realities of operating under high contact stresses. Although theory might be able to estimate an ideal fatigue life, real-world considerations of lubricant film thickness, deflections under load, and manufacturing tolerances were expected to require adjustments to the tooth profile. Upon confirmation of these adjustments, full testing was possible. Data representing actual surface fatigue failure test results must be compared to theoretical estimates to establish a confidence level for the technology.

A determination of the material cost savings, was conducted on a 750-kW wind turbine gearbox. Using conservative estimates, these savings were extrapolated to a 1.5-MW size turbine. In addition to cost savings, the extended life of the gearing components was determined through reduced stresses in the Convoloid gear teeth. The COE analyses of gearboxes using Convoloid technology were developed to emphasize the potential impact on the industry.

2.1.2. Gear Design Methodology

The following rules were observed in the gear-design process.

- Tooth size and geometry was optimized for the intended test loads.
- Ratings were based on current American Gear Manufacturers Association (AGMA) standards.

- Gearing material was identical for both involute and Convoloid gearing.
- Tooth modifications (crown and profile) were made to compensate for deflections of gear teeth, as well as of all components, creating a smooth mesh under load. The modifications also were designed to address lubrication film thickness needs.

2.1.3. Test Fixture Design Methodology

To test the gearing and avoid failures in the test equipment, the following rules were observed.

- Design (or, in the instance of the Micon 108, modify) gearbox housings for maximum stiffness and bore alignment
- Include additional safety factors in the design of connecting couplings and instrumentation components between gearbox shafts.
- Over-design non-gear components (e.g., bearings, shaft strength, keys) when possible.
- Provide a convenient means for inspecting gear teeth.
- Verify measurement of test loads through multiple means.
- Assure lubrication and include automated warning and/or test shutdown system.
- Make visual verification of instrumentation available throughout the test.

Figure 2.2. Micon 108 test fixture, involute gearbox on the left and Convoloid gearbox on the right

Figure 2.3. Micon 108 test fixture, Convoloid gearbox with torque application hydraulic cylinder

2.2. Testing

2.2.1. Description

The study included two tests. Each test consisted of two gearboxes mounted in a back-to-back configuration. Figure 2.4. shows the 3-Hole Test configuration as an example of this method of testing.

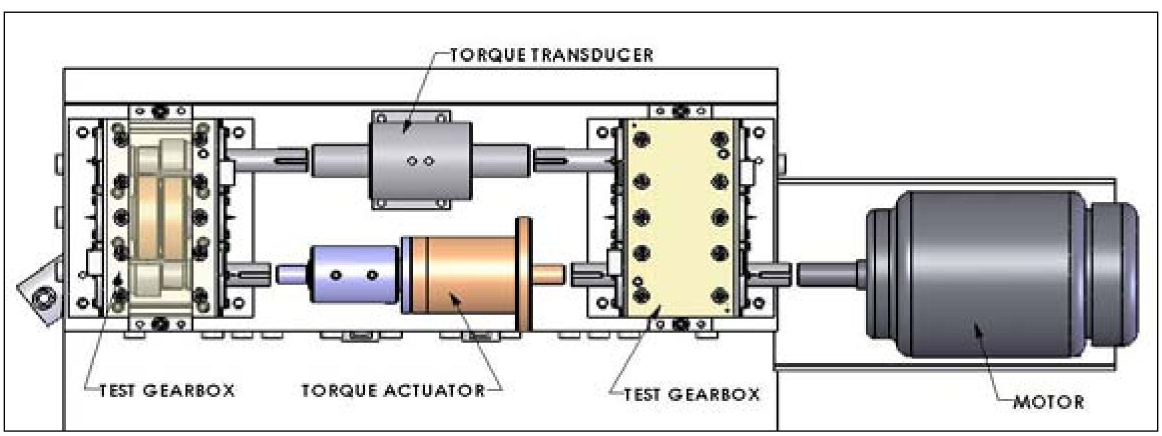

Figure 2.4. The 3-Hole Test configuration

A key advantage of this type of test is the ability to circulate a tremendous amount of torque and power through the gearboxes using only a small percentage of motor power.. Torque is introduced by a torque actuator located between coupled shafts (as shown above) or by rotating one of the gearboxes with respect to the other, the method used for the Micon 108 test. Torque control feedback is gathered through a torque transducer that is connected to the remaining driveshaft between the test gearboxes.

To prevent cross-contamination of the lubricant, both test configurations were designed with separate oil systems that consisted of separate reservoirs, supply pumps, cooling radiators, and filters. These were instrumented for torque, temperature, oil cleanliness, oil flow, and vibration.

2.2.2. Test Gearbox Component Analysis

As with the gearing, all other components were analyzed in an attempt to prevent any non-gearing failures within the equipment. Most components experience far more cycles than do the test gears, therefore, additional safety factors were included where possible.

2.2.3. Testing Methods

Testing of the Convoloid gearing for this project was divided into two areas, Convoloid gear tooth profile development, and a comparison between Convoloid and involute gearing under accelerated loading conditions. From the outset of the project, an understanding that the mathematical theory for the Convoloid tooth form yields tremendous advantages, production and operation of that profile under real-world conditions would require minute modifications.

A key interest during these tests was the development of an oil-film thickness sufficient to prevent metal-to-metal contact of the gear teeth. During light operation, oil coating the gear teeth completely separates the two metal surfaces. As loads increase, surface pressures increase and cause the oil film to be reduced. Although the film thickness in the middle of a gear tooth could be very good, a condition exists at the edges where oil is “squeezed” out,allowing a metal-to-metal condition due to the reduced film thickness. At this interface, premature wear occurs often resulting in a matte-gray surface known as “micropitting.”

Although this failure mode does not greatly alter the tooth shape, microcracks on the tooth surface could precipitate early macropits that shorten the life of a gear tooth. Proper modifications at the edges of contact have been shown to minimize or completely alleviate this wear. Determining these modifications is the purpose for the initial test protocol.

If the gearing can operate without premature wear, the full fatigue testing can commence. Test protocols were developed to ensure that clean, temperature-controlled oil is applied to the bearings and gears. Application of loads must be gradual to prevent scuffing between gear teeth. This wear mode is characterized by localized welding and then tearing of material at the microscopic level. Degradation of the tooth profile occurs rapidly under this failure mode. Prevention is accomplished by assuring proper lubricant supply along with the measured increase in tooth loads.

2.2.4. Test Protocol

Development of test protocols involved preparation of the test fixture and test gearing, preliminary testing of the fixture, assurance of oil cleanliness, load application, inspection criteria, determining conditions for halting the test, and data collection and storage. Initial operations include marking gear teeth for inspection and covering loaded gear flanks with a thin dye for confirmation of tooth contact under load. In instances where progressive wear is expected, repeated inspection of the teeth is necessary. Dyeing the tooth flanks allows inspection of the tooth contact under varying loads (*see* Figure 2.5). Proper contact at test load is necessary to prevent introduction of excessive stresses in the gear tooth.

Figure 2.5. Example of contact pattern for lightly loaded involute gearing in the 3-Hole Test

Prior to beginning a test, all aspects of the test fixture must be verified. In addition to calibration of sensors and thermocouples, torque application methods must be confirmed and lubricant flows verified. Alarm points within the control equipment also must be confirmed. The final step before applying power to the main drive motor is to ensure that contaminants are eliminated from the oil system. After oil reaches an acceptable level, gradual application of speed and torque allow any trapped particles to be removed. Once the test fixture is ready for full-load testing, a gradual method of increasing the torque must be followed to prevent premature lubrication failure.

2.2.5. Definition of Failure

Prior to beginning testing, a specific definition of failure is necessary. The intent of the testing programs is to develop a fatigue related failure comparison, preferably of a surface contact type. Investigation into various failure modes and definitions reveals a widespread range of acceptable (or not acceptable) wear on gear teeth. The discussion in Appendix D gives a taste for the types of gear failures and the ambiguity surrounding when a gear has "failed."

AGMA 925-A03: Effect on Lubrication on Gear Surface Distress states:

> Laboratory testing commonly uses a 1% limit on tooth surface area damage as a criterion to stop a test. However, for field service applications one should always abide by the equipment manufacturer's recommendations or guidelines for acceptable limits of damage to any gear or supporting component.

The purpose of the test was to replicate, as much as possible, a real-world application and to establish an emphatic point of gear failure. Therefore, a value of 1.5% of the active tooth face was chosen. Once macropitting reaches this point, the loaded flank is assumed to have "failed."

2.3. Cost of Energy Analysis

The basis of COE used for this study is the 2002 LWST Baseline COE incorporating a 1.5-MW turbine in a Class 4 wind environment. Benefits to the COE are calculated by using Convoloid gearing and the resultant reduced material usage. Additional COE advantages due to increased service life and the effect on repairs during the service life have not been included. See Appendix A and Section 9 for a more complete discussion.

2.4. Convoloid Gear Inspection Methods

Due to the unique nature of Convoloid gearing, no methods existed for verifying gear tooth profiles, nor were the cutting tools used to manufacture the gearing available. During this project, some methods were developed and are discussed here. As more is understood and equipment software is updated, further refinement will be necessary.

2.4.1. Hob Inspection

All Convoloid profiles are specified with x and y coordinates, usually in the transverse plane. In addition, the pressure angle is supplied for a given point. From this information, the profile of a basic cutter and, ultimately, the true hob form can be created.

After the hob profile is determined, information can be sent to a coordinate measuring machine (CMM) for inspection. A text file containing CMM moves and inspection coordinates is created from the hob profile. Deviations from the ideal location are returned by the CMM and can be plotted. Although the inspection method would be enhanced with a precision measuring machine (PMM), the data gathered is sufficiently accurate to determine whether the physical specimen (hob) is close to the desired profile.

2.4.2. Gear Inspection

For Convoloid gearing, two of the three basic inspections for gearing (index or spacing and lead) can be accomplished with existing gear-inspection equipment. The third measurement—profile—only can be performed using machines having the latest software. Even with the software, a secondary routine currently is necessary to create the deviation charts familiar to those in the gear industry.

Preparation for inspection for Convoloid gears is similar to that for the hobs. However, data supplied is in a simple tabular form. After a gear is inspected, the resulting deviations can be plotted. The amount of deviation then can be compared to industry standard values for gear accuracy.

3. Project Initiation

3.1. Initial COE Projection

During the project kick-off meeting held August 5, 2005, a projection of the cost of energy for the Micon 108 with enhanced Convoloid gearing was presented. The projection was based upon wind-load data representative of the full-size test location (near Palm Springs, CA). Harry Halloran (Energy Unlimited) supplied a power curve based on existing turbines at this location. The emphasis of the study was to reveal the change in COE when using the new Convoloid gear geometry versus the industry standard involute technology.

Costs were based upon a production quantity of 250 pieces. The specific area of cost reduction is due to a 20% reduction in volume and weight of the gearbox, yielding a savings in material costs. This COE projection is not based upon an increase in power capacity, although that later could prove valid.

The following aspects of COE for the new gear configurations are evident.

- Existing Applications—An existing drive could be rebuilt, maintaining the same interface but having a higher capacity and/or life. Older units might benefit greatly from the technology due to the savings in maintenance and repair costs. Observations from operators in the industry reflect that the cost of repair is a large part of the COE.
- New Installation—Gear drives could be a smaller size but have similar gear-stress levels, saving weight on the platform. A balance between reduced size and lower gear stresses (yielding longer life) could affect new systems dramatically. For large turbines, a reduction in size and weight of the gearbox directly affects other systems (e.g., tower, nacelle).

Cost of energy advantages are directly related to the cost for O&M and repairs. Operations and maintenance costs include scheduled maintenance; repair costs generally are unscheduled maintenance, typically costing between $7,500 to $10,000 per MW per year. Cost of energy trade-offs are addressed in detail in Section 9. A summary sheet of the COE is shown in Figure 3.1.

COE PROJECTION SHEET

Baseline Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - 70 Meter Roto
Improved Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - Advanced Power Converte

Rating (kWs)	1500	1500		
Component	**Baseline Component Costs $1000**	**Projected Component Costs $1000**	**Component Percent Improvement**	**Major Cost Element % Improvement**
Rotor	**248**	**248**	0.0%	
Blades	148	148	0.0%	
Hub	64	64	0.0%	
Pitch mchnsm & bearings	36	36	0.0%	
Drive train,nacelle	**563**	**533**	-5.4%	
Low speed shaft	20	20	0.0%	
Bearings	12	12	0.0%	
Gearbox	151	121	-20.0%	
Mech brake, HS cpling etc	3	3	0.0%	
Generator	98	98	0.0%	
Variable spd electronics	101	101	0.0%	
Yaw drive & bearing	12	12	0.0%	
Main frame	64	64	0.0%	
Electrical connections	60	60	0.0%	
Hydraulic system	7	7	0.0%	
Nacelle cover	36	36	0.0%	
Control, safety system	**10**	**10**	0.0%	
Tower	**101**	**101**	0.0%	
TURBINE CAPITAL COST (TCC)	**921**	**891**	-3.3%	-3.3%
Foundations	49	49	0.0%	
Transportation	51	51	0.0%	
Roads, civil works	79	79	0.0%	
Assembly & installation	51	51	0.0%	
Elect interfc/connect	127	127	0.0%	
Permits, engineering	33	33	0.0%	
BALANCE OF STATION COST (BOS)	**388**	**388**	0.0%	0.0%
Project Uncertainty	**162**	**162**	0.0%	
Initial capital cost (ICC)	**1,472**	**1,442**	-2.1%	
Installed Cost per kW for 1.5 MW turbine (cost in $)	**981**	**961**	-2.1%	
Turbine Capital per kW sans BOS (cost in $)	**690**	**669**	-3.0%	
LEVELIZED REPLACEMENT COSTS (LRC) ($10.7 kW)	16	16	0.0%	0.0%
O&M $20/kW/Yr (O&M)	30	30	0.0%	0.0%
Land ($/year/turbine)	5	5	0.0%	
NET 5.8 m/s ANNUAL ENERGY PRODUCTION MWh (AEP)	4439	4439	0.0%	0.0%
Net 6.7 m/s ANNUAL ENERGY PRODUCTION Energy MWh (AEP)	5519	5519	0.0%	0.0%
Fixed Charge Rate	11.85%			
COE at 5.8 m/s $/kWh	**0.0480**	**0.0472**	-1.7%	
COE at 6.7 m/s $/kWh	**0.0386**	**0.0380**	-1.7%	

Figure 3.1. Initial COE comparison with Convoloid technology gearbox

3.2. Project Task Outline

The following is a basic summary of the project task outline. The detailed task outline document is included in Appendix C. The original outline was modified in May 2006 to shift testing from the Micon 108 test to the new test fixture referred to as the 3-Hole Test. These items are included under Task 4.

- Task 1—Initial Cost of Energy Projection
- Task 2—Kick-Off Meeting, Conducted 7-27-05

- Task 3—Gearbox Comparison Test and Protocols
    - 3.1. Refine manufacturing processes
    - 3.2. Compare stresses at 200% rated load to ensure that bending strength life far surpasses that for surface durability
    - 3.3. Calculate the life of both boxes under their respective loads in a surface failure mode—define “failure”
    - 3.4—Analyze manufacturing sensitivities of Convoloid gearing and their effects on the rating factors that should be used
    - 3.5—Relate testing protocol to expected loads for a Class-4 wind site
    - 3.6—Perform full analysis of housings, bearings, shafts, keys, and other components to determine safety factors of these components under protocol loads and speeds (to preclude premature failure)
    - 3.7—Describe planned test protocol
- Task 4—Testing of Involute and Convoloid Gear Sets
- Task 5—Detailed Analysis of Test Results and Predictions
- Task 6—Field Testing
- Task 7—Detailed Design of Scaled-Up Gearbox Using Convoloid Gear Sets

3.3. Stress Comparisons

3.3.1. Rating Factors

Methods for rating gearing are based upon determining the maximum stresses of the gear tooth (both contact and bending) and relating those stresses to the material allowables. Various factors are used to modify the applied or the material allowable stresses to compensate for variations in the tooth geometry and function. Two important factors for gear rating are gear speed (velocity factor) and tooth mesh alignment (load distribution factor). The determination of equations for calculating these factors is reviewed continually within the gear industry, as the physics is very complex. Application of these factors to the Convoloid tooth form follows the same principles and also requires continual development as more is learned.

3.3.2. Projection of Rating Factors and Stress Levels

Table 3.1 and Table 3.2 show the comparison data for the low-speed and high-speed passes of gearing for the Micon 108. The general rules governing this comparison were identical to those for the low-speed pass, including the determinations of “limiting input torque,” material, heat treatment, and manufacturing sequence. In summary, the rules are as follows.

- Ratio: Very nearly the same for both cases; slight ratio changes were made at final manufacture to achieve exact ratio equality so that the 4-square rig operated properly
- Center Distances: Same
- Face Widths: Same
- Helix Angle: Slightly larger for Convoloid

- Module: Heavier (larger size gear teeth) for the Convoloid
- Specific Sliding Values: Convoloid is well within tolerance of the values of the involute pair

Table 3.1. Wind Turbine Gear Drive System (Low-Speed Pass)

Low-Speed Pass	Optimized Involute	Convoloid
N1	16	13
N2	80	65
Center Distance	342.9 mm	342.9 mm
Face Width	127 mm	127 mm
Helix Angle	9.0°	12.3894°
Module	7.0	8.6
Gear Ratio	5.0	5.0
Limiting Input Torque	7,681 Nm	10,221 Nm
Ratio of Convoloid limiting torque to involute limiting torque 1.33		
Specific Sliding	-1.21 to + 0.826	-0.617 to + 0.738

Table 3.2. Wind Turbine Gear Drive System (High-Speed Pass)

Low-Speed Pass	Optimized Involute	Convoloid
N1	15	13
N2	94	81
Center Distance	254.0 mm	254.0 mm
Face Width	79.37 mm	79.37 mm
Helix Angle	11.0°	12.19°
Module	4.57	5.28
Gear Ratio	6.267	6.231
Limiting Input Torque	1,712 Nm	2,595 Nm
Ratio of Convoloid limiting torque to involute limiting torque 1.46		
Specific Sliding	-1.216 to + 0.815	-0.575 to + 0.698

3.3.3. Project Operation and Maintenance Costs as a Function of Stress

When determining the operating and maintenance costs for a wind turbine system, a significant variation can occur due to replacement of the main gearbox. This is especially troubling when repairs are unexpected. There is a history of premature gear failures involving involute technology, and most O&M calculations include planning for gearbox replacement.

The advantage of Convoloid gearing is that, for a given size, gear stresses are much less than with same-sized involute gearing. Lower stresses mean increased gearing life. Replacing involute gearing with Convoloid technology—maintaining the existing gearbox and gearing size—and operating under the rated loads yields tremendously longer life for the gearbox. Therefore, O&M costs are reduced due to less frequent repair and replacement of the gearbox.

3.4. NEG Micon 108 Gearbox Housing Rebuild

Preparation of the housings for testing required careful machining to keep bearing bores aligned and maintain proper bearing setting.

3.4.1. Housing Rigidity

Initial testing revealed that the single-piece housing lacked sufficient rigidity to properly set the tapered roller bearings on the high speed shaft. To prevent the housing faces from spreading apart, four tapered pins were installed through the cover plate. This greatly increased the housing stiffness. Figure 3.2 shows the involute test gearbox without the cover plate. Additionally, mounting bolts repeatedly pulled out of tapped holes in the housing.

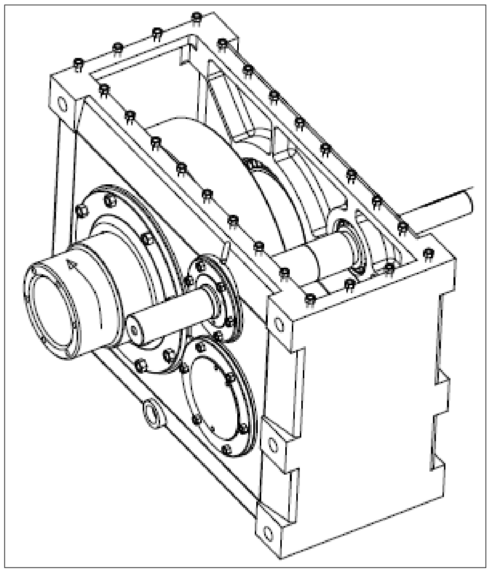

Figure 3.2. Micon 108 involute test gearbox configuration

3.4.2. Bearing Bores

All bearing bores were bored oversized and sleeved to provide dimensional integrity for the bearings. The housings had marginal stock around the bores therefore the machining required great care and precision, and only very minimal stock could be removed.

3.5. Involute Gearbox Design

3.5.1. Involute Gearbox Component Manufacture

Manufacture of the involute components follows standard industry practice. The original low speed shaft was used for the test as it only required minor clean-up machining. Bearings caps were modified for lubrication piping and bearing temperature probes. The high speed shaft was designed with two extensions to facilitate coupling to the test drive motor. Steel used for the gearing (identical for both the involute and Convoloid) was selected for optimum chemistry and hardenability. Gear blanking, rough hobbing, heat treatment, and finish grinding were completed in accordance with standard operating procedures.

Modifications to the keyed connection in the intermediate shaft were necessary because of increased stresses that were to be imposed due to the higher test torques. To prevent a buildup of stress concentrations at the gear shoulder (point where the gear slides on the shaft and butts up against the pinion shoulder), the keyway was extended slightly into the shoulder and the key was lengthened. The shoulder radius was also polished.

3.5.2. Involute Gearbox Inspection Procedures

Inspection followed industry practice. Gears were inspected using specialized gear-inspection equipment and conformed to AGMA quality 12. Further, no-load contact between the pinions and gears was used to confirm proper starting contact of mating gearing. The bearing bores diameter and location in the housing were verified using a coordinate measuring machine. Upon assembly, all gearing was checked to confirm that all rotated smoothly.

3.6. Optimized Convoloid Design Gearbox

3.6.1. Convoloid Gearbox Component Manufacture

3.6.1.1. General

As discussed, the design and manufacturing of the Convoloid gearbox, accessories, etc. essentially was the same as that used for the involute gearbox.

3.6.1.2. Blanking

The blanking operations for all Convoloid gearing essentially were the same as for the involute gears.

3.6.1.3. Rough Hobbing

The rough hobbing operation essentially was the same as that used for the involute gearing, except that small changes in feeds and speeds of the gear hobbing machine could have been advisable to take into account the heavier module of the Convoloid designs. Convoloid hobs are made in the same manner as highly altered involute hobs, using computer programs to describe the required Convoloid hob rack forms. Hobbing Convoloids takes place in a manner exactly as with involutes.

3.6.1.4. Heat Treating

The Convoloid gears were carburized, quenched, and tempered using exactly the same process as the involutes. The only specified difference was the required effective case depth, which allowed for the slightly heavier effective module of the Convoloid pairs. The normal precautions regarding case depth versus tip width and subsurface shear considerations are consistent with those considerations for involute designs.

3.6.1.5. Final Tooth Grinding

Most—if not all—gear grinding machine manufacturers utilize software protocols based on basic involute parameters with additional subroutines to add specified tip relief, twist correction, crown, and other anomalies. To date, it has been necessary to use the existing software to grind Convoloid forms. Certain procedures have been developed to overcome some of the apparent differences in grinding totally non-involute forms using involute-based routines. Wheel dressing techniques, inclusion of tip relief, and practices to minimize twist have been developed by Genesis Partners LP.

3.6.1.6. AGMA/ISO Gear Accuracy Standards

The accuracies of lead, profile, spacing, and other important quality standards for Convoloid gears mirror that of the involute world. Classification of Convoloid gears therefore takes into account the errors found in these parameters and then relate those errors to like errors in involute gears.

3.6.2. Convoloid Inspection Methods

Due to the unique nature of Convoloid gearing, no methods existed for verifying gear tooth profiles, nor did the cutting tools used to manufacture the gearing. During this project some methods were developed and are discussed herein.

3.6.2.1. Gear Inspection

For Convoloid gearing, two of the three basic inspections for gearing (index or spacing and lead) can be accomplished with existing gear-checking equipment. The third measurement, profile, only can be performed using machines with the latest software. Even with the software, a secondary routine currently is necessary to achieve the deviation charts familiar to those in the gear industry. Figure 3.3 shows an early profile chart of the 13-tooth Convoloid pinion from the Micon 108 gearbox. The amount of deviation can be compared to industry standard values for gear accuracy.

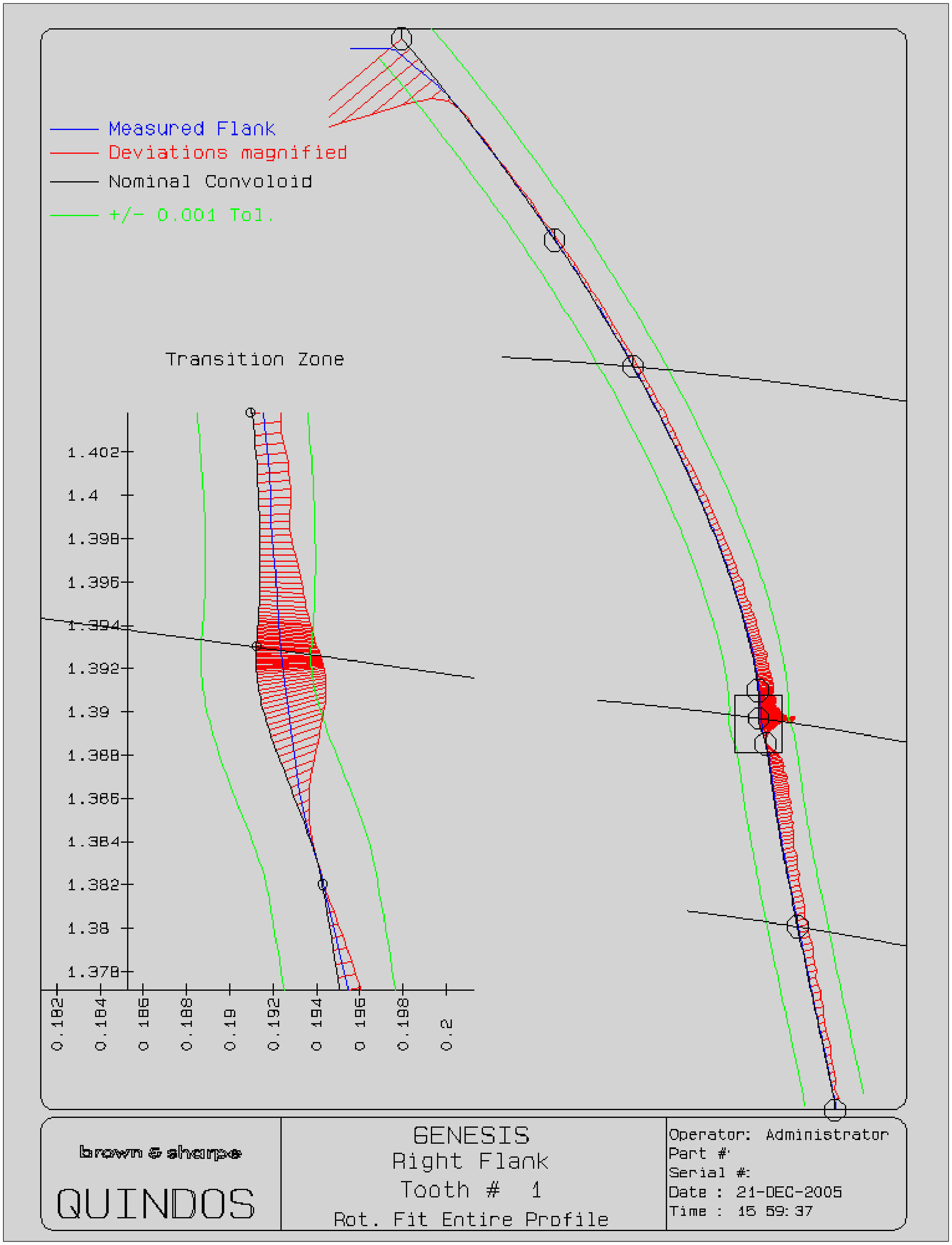

Figure 3.3. Early profile inspection trace of Convoloid 13-tooth pinion

3.6.2.2. Hob Inspection

Traditional hob inspection techniques often employ a visual comparator with scaled representations of the intended tooth form. Although this method can be very accurate, no quantitative data can be easily gathered, making comparisons with AGMA and ISO accuracy standards difficult. Genesis Partners developed an alternative method that has proven accurate and reliable. As with gear profile accuracies, Convoloid rack forms can be quantified and matched to AGMA/ISO standards. Although the inspection method would be enhanced with a precision measuring machine, the data gathered is sufficiently accurate to determine whether the physical specimen (hob) is close to the desired profile.

4. Testing—Micon 108

The focus of this test was to refine the Convoloid tooth form from theory to practical application. Once qualified, testing continued with the improved Convoloid gear profile in comparison with current involute geometry. Stress calculations, especially with respect to contact stresses, reveal dramatic reductions when the Convoloid tooth form is used. When designing a Convoloid gear set with contact stresses approaching those of involute gears, larger size (greater pitch) gear teeth could be chosen, increasing the bending strength of the gear teeth.

4.1. Description of Test

4.1.1. Gearbox and Test Fixture Configuration

The Micon 108 wind turbine gearbox was chosen for the initial comparison test, due to a readily available supply. This drive has seen widespread use in the industry, has an extensive history, and is of a convenient torque capacity and size for economic load testing (weighing approximately 1,000 kg). A detailed discussion of the test appears in Appendix E.

The two test gearboxes were rebuilt by The Gear Works (Seattle, WA) to like-new condition, which included pinning the housings to increase stiffness. The first drive was assembled with high-quality (AGMA Class 12) involute gearing. The second drive employed gearing that used the Convoloid technology. All material used for the gearing exceeded AGMA Class 2 material specifications, and was surface hardened prior to final tooth grind. The involute geometry could be verified by in-shop measuring devices, and the Convoloid geometry required outside confirmation via a precision coordinate-measuring machine, provided courtesy of Brown & Sharpe (B&S) (North Kingstown, RI). After the gearing was assembled with new bearings into the housings, no-load contact patches were obtained for all gearing. Approval of these contact patterns was required prior to execution of the test protocol.

The drives were instrumented to enable monitoring of conditions including (but not limited to) all bearing and sump temperatures, housing vibrations, and strain between the housing bores. Each drive was outfitted with separate oil delivery systems, including pump, primary filter, secondary filter, and heat exchanger. The oil was continuously monitored for particulates.

Load application to the drives was accomplished using the “four-square” method, incorporating a hydraulic cylinder to torque the common low speed shaft. A strain gauge at the cylinder and a separate torque gauge between the high speed pinions provided two methods for verifying the torque applied to the system.

4.1.2. Test Fixture

The two gearboxes were configured in a four-square, back-to-back test arrangement. Lubrication systems for each drive were isolated with separate filter and cooling systems. Extensive gearbox data was collected by a computerized monitoring system. The test fixture was located at The Gear Works (Seattle, WA) and was designed to provide a stable platform and withstand heavy loads.

4.1.2.1. Torque Application

The four-square test method is an efficient way to apply a substantial dynamic load without requiring extremely large electrical motors and generators. In this type of test, two drives with the same exact ratio (27:1, in this case) are mounted to a common low speed drive shaft. The high speed shafts are coupled with a torsionally stiff coupling that allows for a large angular misalignment. The advantage of this setup is that instead of requiring a motor that is rated for the full power of the tested gearboxes (456 hp), the motor only must supply the power lost due to inefficiencies in the test system (approximately 26 hp). One gearbox is held fixed and serves to support the motor. A torsional load is applied to the test gearbox, thereby twisting the common low speed shaft. The high-speed coupling must accommodate the resulting misalignment.

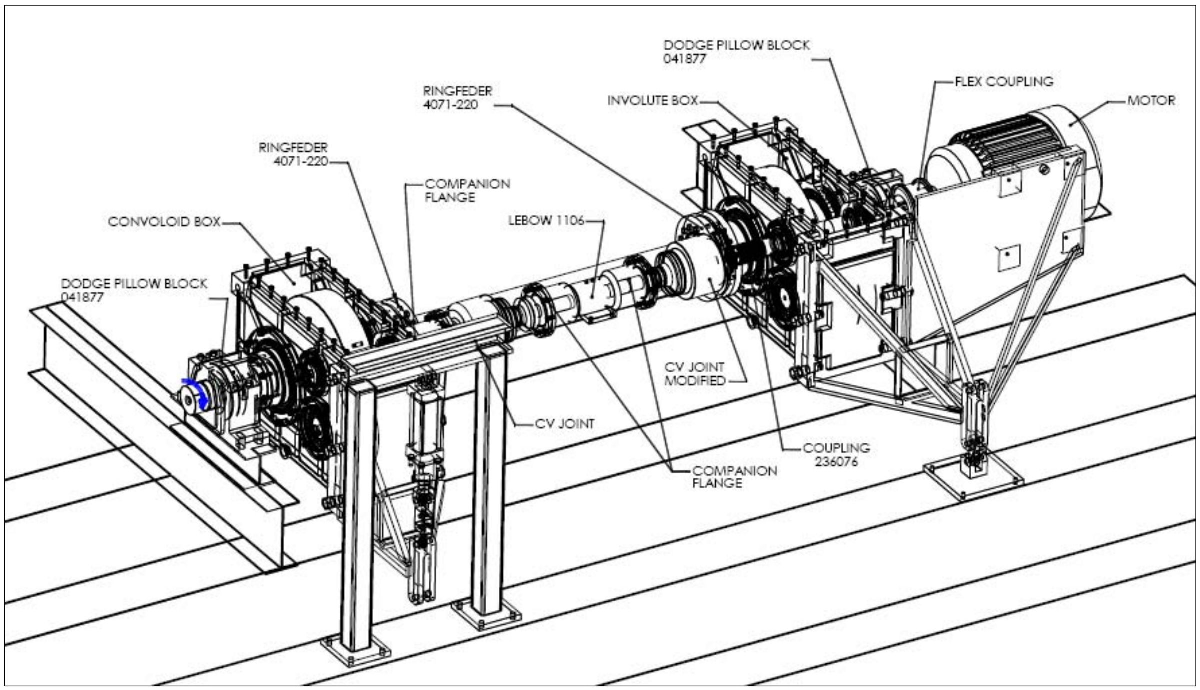

Figure 4.1. Conceptual layout of the Micon 108 four-square test

As shown in Figure 4.1, the involute gearbox (shown on the right side of Figure 4.1) is fixed via a pinned connection to the base structure, and the Convoloid gearbox (shown on the left side) is connected to a hydraulic cylinder. When pressure is applied to the cylinder, the Convoloid gearbox is rotated around the main shaft (supported by pillow block bearings) introducing a twist on that shaft. By tying the high speed shafts together and keeping the involute gearbox fixed, the twist induces a torque to the system.

The Convoloid gearbox rotates around the main drive shaft as load is applied through the hydraulic cylinder, therefore the high speed shafts do not remain aligned. Some couplings fail to maintain a constant rotational velocity as the angle of the shafts increase, so it was important to select constant velocity (CV) couplings for the high speed shafts. Further, to monitor the applied torque, a torque transducer was mounted between the two couplings. Figure 4.1 shows a good representation of the arrangement. Not shown in Figure 4.1 are the oil sump tanks, filters, and heat exchangers.

To minimize as much high speed shaft misalignment as possible (and therefore reduce any potential speed variation), at no load the Convoloid gearbox was positioned with the pinion shafts unaligned. The amount of initial misalignment was calculated based upon the expected torsional deflection of the solid low speed shaft. Minor adjustments on the fixture resulted in the shafts being nearly aligned once the test load was applied.

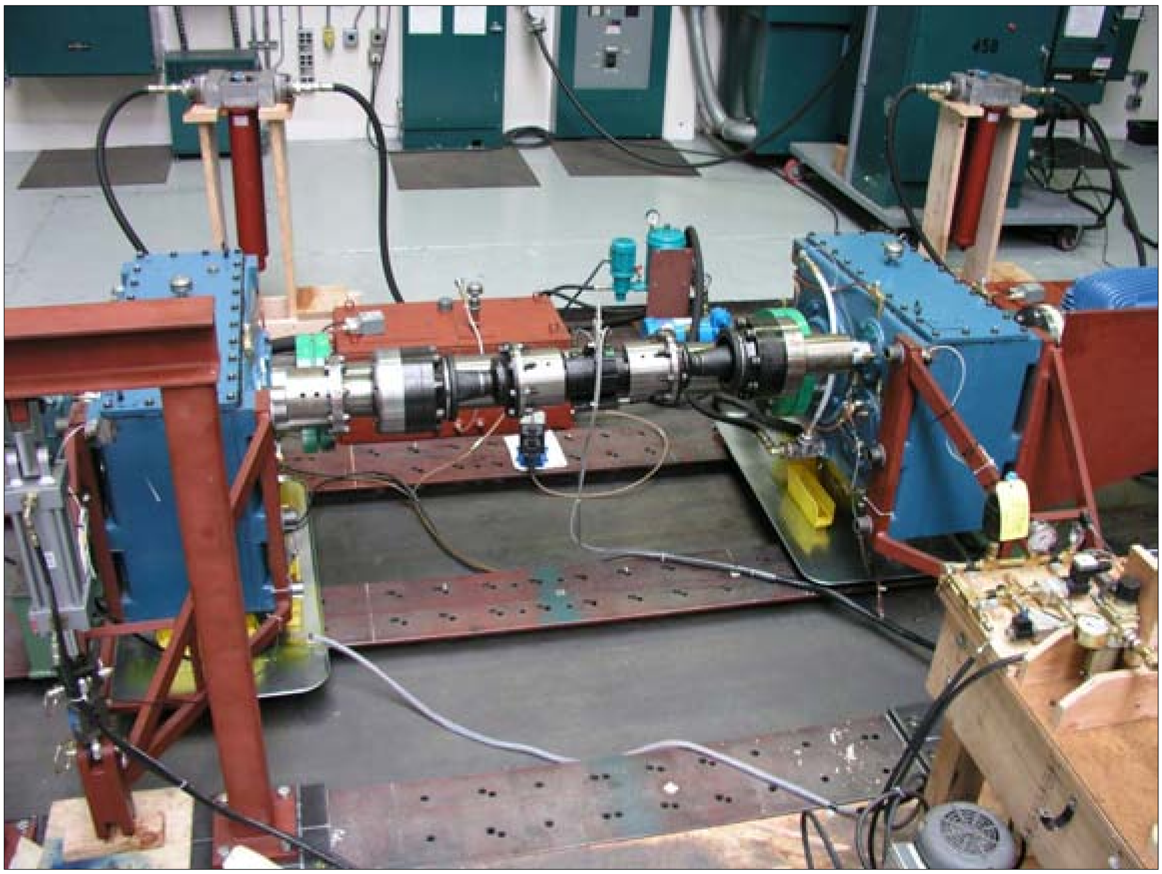

Figure 4.2. The Micon 108 test equipment: The involute gearbox is on the right, the Convoloid gearbox is on the left; also visible on the left is the hydraulic cylinder (used to introduce loading to the system)

4.1.2.2. Test Fixture Lubrication

In the test, it was necessary to control as many variables as possible. Assuring that all wear components were properly supplied with clean oil was of supreme importance. To prevent cross-contamination, two identical systems were developed for the gear drives. Test oil viscosity was chosen to be similar to that used in the field: ISO320.

The lubricant system began with the oil sump, which consisted of three chambers that allow any particles in the oil to drop out of suspension, and also minimize foaming and trapped air. A heating element in the sump was used to bring the oil to operating temperature. Oil was pumped through a high-capacity 10-micron filter and a heat exchanger. Therefore, clean oil of a consistent temperature was distributed to all bearings in the test gearbox. Individual valves for each bearing allowed manual oil-flow corrections to be made to maintain bearing temperatures.

Although all oil passes through a filter with a 10-micron element prior to entering a test gearbox, a separate kidney-loop circuit with an oil pump and 3-micron filter constantly filtered oil in the sump. Oil cleanliness of the sump lubricant was monitored constantly.

4.1.2.3. Instrumentation

A number of test parameters were monitored during the test. A detailed listing of those instrumentation points is given in Appendix E.

4.1.2.3.1. Monitored Values

Of high importance was monitoring bearing temperatures and gearbox vibrations, so that any adverse deviations could be documented. If a warning threshold was surpassed, then visible and audible alarms were set to facilitate the test's shutdown. Additionally, analysis of the captured data could be used to detail methods for earlier notification of impending gear and bearing failures.

4.1.2.3.2. Controlled Values

Motor speed and applied torque were controlled via manual inputs. Adjustments to the motor set the motor speed. A rheostat was used to establish pressure in the hydraulic cylinder to generate torque with manual verification of the desired values. During operation, motor speed remained stable; however, slight adjustments to the hydraulic pressure were required to maintain the desired torque.

4.1.2.4. Alarms

Visual and audible alarms were installed to warn of potential gearbox or test fixture failures. Included in Appendix E are specific alarm points. As mentioned, all bearing temperatures were monitored with alarms set for temperatures exceeding 190°F. Alarms also were set for vibration and lubricant supply.

4.2. Test Preparations

4.2.1. Test Gearbox Design

Based upon the existing production gearbox, the test gearboxes maintained a basic design of parallel shaft, helical gearing supported by roller bearings. Although gearing was optimized, bearings, shaft design, and the one-piece housing were retained.

4.2.1.1. Gearing Design

A four-square test imposes a constraint that the two gearboxes used in the test must have identical gear ratios. The ratio used for the test is near that of the original gearboxes: 27.00:1. Under this constraint, the gear sets were optimized to produce the highest ratings. Face widths are the same as with the original gearbox. The involute gear sets were rated per AGMA standards (ANSI/AGMA 2001), and the Convoloid gear sets were rated using the same theory used by AGMA for stresses and life. The two involute gear sets (high speed (HS) and low speed (LS) stages) are summarized below.

Table 4.1. High Speed Involute Gear Set

	Pinion	Gear
Number of Teeth	16	96
Normal Diametral Pitch (Module)	5.776 (4.397)	
Normal Pressure Angle	21.9808°	
Helix Angle	10.0°	
Center Distance	9.8425 in	

Table 4.2. Low Speed Involute Gear Set

	Pinion	Gear
Number of Teeth	16	72
Normal Diametral Pitch (Module)	3.175 (8.000)	
Normal Pressure Angle	20.0°	
Helix Angle	10.0°	
Center Distance	13.9764 in	

A full analysis using the AGMA Gear Rating Suite of software for the involute gearing under the 240% load is included in Appendix F. The two Convoloid gear sets are listed below.

Table 4.3. High Speed Convoloid Gear Set

	Pinion	Gear
Number of Teeth	13	78
Normal Diametral Pitch (Module)	4.73128 (5.3685)	
Equivalent Normal Pressure Angle	24.96°	
Helix Angle	12.292°	
Center Distance	9.8425 in	

Table 4.4. Low Speed Convoloid Gear Set

	Pinion	Gear
Number of Teeth	12	54
Normal Diametral Pitch (Module)	2.44079 (10.4065)	
Equivalent Normal Pressure Angle	30.60°	
Helix Angle	14.679°	
Center Distance	13.9764 in	

All gearing was analyzed for deflections to determine the proper amount of tooth modifications necessary to achieve a smooth, balanced load across the active tooth flank. This analysis required creating an analytical model of the gearing, supporting shafts, bearing stiffness, and housing stiffness. The variability of the housing stiffness required modifications of the housing to properly support the gearing.

An analysis of stresses for the test gearing revealed a dramatic reduction in stress levels for the Convoloid gearing versus the involute. When applying the 200% load, the contact stresses for the HS and LS involute gearing were 174 ksi and 205 ksi, respectively. This compares with the Convoloid contact stresses of 167 ksi and 186 ksi (HS and LS respectively). Similar reductions are evident when comparing the bending stresses. The complete report is given in Appendix G.

The minimum anticipated L_1 lives for the involute (low speed gear set) was 2,985 hours; the Convoloid minimum life (low speed gear set) was 16,750 hours. A complete discussion is presented in Appendix H.

4.2.1.2. Bearings

- The bearing configuration for the test gearboxes was the same as for those currently in the field. One modification was to increase the size (and therefore the rating) of the high speed bearings.
- Three different types of bearings were used in the Micon 108 gearboxes, tapered roller bearings for the high speed shaft; spherical roller bearings for the intermediate shaft; and cylindrical roller bearings for the low speed hollow shaft. The lives of the HS and intermediate bearings are much

shorter than those for the LS bearings. At a test load of 240% rated and 2,050 rpm, the expected L_{10} life of the HS bearings was expected to be 280 hours and the anticipated life for the intermediate bearings was 418 hours.

- Making changes to bearings was not possible during this phase of the project. It was expected that supplying a ready supply of cool, clean oil to the bearings would increase their life. A summary of all internal components in included in Appendix I.

4.2.1.3. Housing

- During assembly of gear drive components, it was found that the HS tapered roller bearings would not retain the necessary preload setting. This was not understood until preliminary testing (conducted in June 2005) revealed unexpected shifts in the gear mesh tooth contact patterns as soon as power exceeded 100% of rated load (108 kW). The shifts were especially evident at 150% of rated load. This problem was evident in both the involute and Convoloid drives.
- Using a "space frame," a static test of the involute box indicated that twisting of the housing was not a problem. A second test to determine whether the housing was deflecting due to gear axial forces revealed excessive deflection (0.007 in to 0.009 in). This deflection was reduced dramatically when the cover plate was taper-pinned to the housing, essentially becoming part of the load-carrying structure. After this modification, tooth contact pattern shifts were not a cause for concern.

4.2.1.4. Component Analysis

The complete report is included in Appendix I.

- Verification that individual components are capable of withstanding the test loads is good practice and included in the detailed task outline (Appendix C) under Task 3.
- Key analyses revealed a minimum safety factor of 1.03 (for the involute HS shaft extension) at the full 240% test load. Other keys were found to have safety factors much greater than 1.0.
- Experience has shown that rotating shafts supporting gears are loaded with alternating stresses that can cause fatigue failures. A critical area of stress is the intermediate shaft where loads from two gears apply; specifically, the shoulder where the gear contacts the pinion. During analysis the existing design barely met a 1.0 safety factor in this area. Safety was increased by grinding the shoulder radius.

4.2.2. Manufacturing Sensitivities of Convoloid Gearing and Their Effects on Rating Factors

The mesh action of the Convoloid tooth form is similar to the involute form in that both exhibit a conjugate action. This action creates a smooth transfer of power from one component to the other. Because of geometry differences between the Convoloid and the involute tooth forms, however, calculation methods for key factors used to rate a gear set are modified. Many rating factors are the same and react to manufacturing deviations similarly, but differences exist where factors are based upon tooth-profile geometry. Those factors with distinct differences are discussed here. The complete report is included in Appendix J.

4.2.2.1. Specific Sliding

The relationship of the sliding to rolling velocities for Convoloid gear sets are reduced throughout the mesh. This reduced “slip” of one gear tooth with respect to the other is highly desirable, reducing the risk of scoring and enhancing film-thickness characteristics.

4.2.2.2. Absolute Sliding Velocities

Values for the sliding velocity are computed from the difference in rolling velocities of the pinion and gear. The sliding velocities are slightly greater for the Convoloid; however, the relationship of the sliding ratio to the rolling ratio (specific sliding) is smaller. This slight increase in absolute sliding velocity is thought not to be detrimental, due to the relative curvature of the Convoloid gearing.

4.2.2.3. Oil-Film Thickness

Film thickness is directly related to the curvature of the elements in contact and their sliding velocities. In the traditional involute tooth form, the two mating parts have a convex curvature (similar to two cylinders rolling against each other). This contact exhibits a constantly changing radius of curvature for each element, creating a varying film thickness over the tooth profile. In gears having small numbers of teeth, this condition causes widely varying sliding velocities. In contrast, the Convoloid profile creates a situation where one mating tooth portion is convex and the complementary portion is concave (similar to one cylinder rolling inside another) and the two profiles have a constant relative curvature. Thus, the oil-film thickness can be consistent throughout the profile and can exhibit a capacity for greater film thickness.

4.2.2.4. Center Distance Changes

Within normal center distance tolerances common today on modern CNC machinery, Convoloid gearing sensitivity to center distance changes will not affect operation or rating expectations. It should be possible for Convoloid gearing to function if the center distance is closer than the calculated value. However, distinct limits relating to tooth thickness and the location of the transition zone could limit operation of the gear set. The application of Convoloid technology in planetary gear sets where the sun gear floats and is in tight contact when no load is present has not been tested.

4.2.2.5. Lead Crown (Tooth Modification)

Although Convoloid gearing can be crowned in a way very similar to involutes; early gear grinding revealed the characteristic “twisted tooth” syndrome. This condition is characterized by gears having exact leads at the pitch diameter, but with positive lead readings in the dedendum, and negative lead readings in the addendum on the same flank of the same tooth, or vice versa. The greater the amount of crown specified for a gear, the worse the condition. Subsequent investigation during manufacturing helped minimize this condition.

4.2.2.6. Bearing Capacity

Although the radial and separating forces of Convoloid gearing closely approximate those in same-sized involute gearing under the same load/speed spectrums, when Convoloid theory is applied to increase power density of a gear drive system to carry a certain load/speed spectrum, gear center distances and gear sizes are reduced as compared to the involute design (but maintain the same or lower Hertz and bending stresses), thus increasing the loads on the supporting bearing system. The bearing industry recently developed processes and procedures to significantly increase the power density of products. It is thought that these advancements will economically accommodate these anticipated increased loads, providing industry with well-balanced gear drive systems from a stress and reliability standpoint.

4.2.3. Gear Manufacturing

Manufacture of Convoloid gearing follows the same processes as with involute gearing. Creation of the gear blanks and pre-grind gear cutting follow the same methods. Variations between the gear types occur when finish grinding the gear teeth. Specifically, because gear grinding software is designed for involute gearing, allowances must be made to “convince” the software to properly grind the Convoloid tooth form. Additionally, verification of the Convoloid profile is not as straightforward as for involute gears.

Currently, the best confirmation of proper contact between Convoloid gear teeth is to use a marking compound to transfer a no-load contact pattern. Once contact was centered on the tooth top-to-bottom (Figure 4.3), lead corrections were used to move the contact towards the middle of the tooth (Figure 4.4). Investigations conducted during the second phase of testing show promise in developing similar inspection techniques as with involute gearing, enabling this type of check to be a confirmation of the method, not the sole means of confirmation.

4.2.3.1. Grinding Lead Modifications into Convoloid Gearing

When gears are subjected to high loads, lead modifications are normally applied to compensate for deflections of the gear tooth and supporting shaft. As the amount of modification increases, machine tool software must compensate for “tooth twisting” that occurs on a tooth flank. Tooth twisting is a phenomenon that typically occurs when grinding both involute and non-involute forms. It manifests itself, for example, as a positive lead error in the addendum of the gear, accurate lead at the pitch line, and negative lead error in the dedendum creating a cross lead on the same flank of the same tooth. Machine tool manufacturers of gear grinding machines have developed software corrections for situations, such that lead errors are greatly reduced and twist is very minimal.

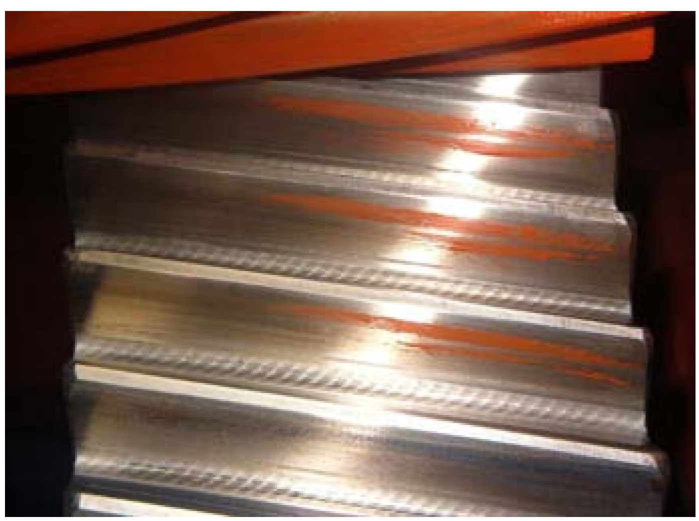

Figure 4.3. Convoloid high speed gear mesh no-load contact patch prior to lead modification

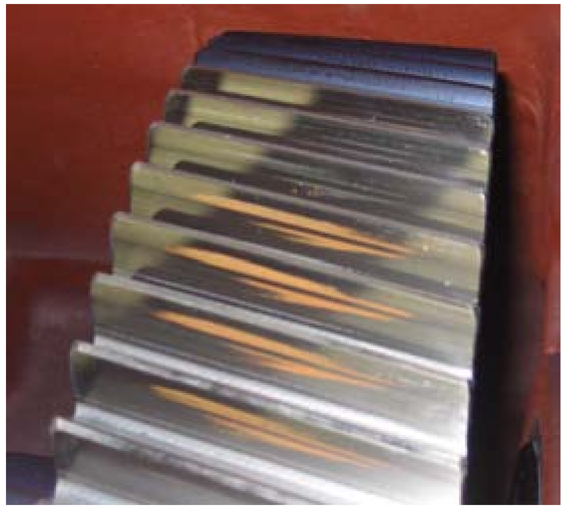

Figure 4.4. Convoloid high speed gear mesh no-load contact patch after lead modification; gearing has been assembled into the test housing

4.2.3.2. Tip Relief

Selection of the amount of tip relief for Convoloid gear teeth was approximated from the involute analysis. The Convoloid gear tooth has a greater stiffness than the equivalent involute, therefore the Convoloid tooth will not flex as much under load. This leads to a reduced amount of tip relief being required. See Appendix P and Figure P.18.

4.2.4. Test Protocols

This test method is based on an industry-proven method in which 200% of the rated gearbox load (after accounting for generator inefficiencies—219,500 lb-in rated torque; 439,000 lb-in test torque) is applied for 200 hours. To increase the number of test cycles, the high speed shaft is driven at the motor speed (1,767 rpm) instead of the operational field speed of approximately 1,200 rpm. See Appendix K, Appendix L, and Appendix M for a more complete discussion of this test method. In an attempt to reduce test time, the test load later was increased to 240% of the rated gearbox load and the motor speed increased to 2,050 rpm.

4.2.4.1. Shakedown

The shakedown procedure (reference Shakedown Protocol-r2, dated November 17, 2005)(*see* Appendix K) was performed until particulates in the oil were well within an acceptable range (approximately ISO 14/11/9 maximum for both oil systems). Application of load revealed no problems, and the load was increased in increments to 200%. The wear pattern of both drives gradually increased to cover the entire face width as the load was increased, as expected. Although the oil was extremely clean (oil patch tests confirmed instrument readings), radial marks appeared on the loaded flanks. These marks could not be felt and were deemed inconsequential.

4.2.4.2. Primary Test Protocol

Execution of the test protocol 200% load test (reference Test Protocol, dated September 7, 2005) (*see* Appendix L) was as follows. After a successful shakedown, execution of the test protocol commenced. Notably, due to a mathematical error in the torque strain gauge chart (converting from micro-strain to input torque), the gearboxes were operated at 77% of rated load for 179 hours. Upon discovery of the error, the shakedown procedure was followed up to the proper 200% torque value. The test protocol was implemented once again at the proper load and was concluded after 235 hours. All notations of load in this report refer to the actual percent of load applied.

4.2.4.3. Continued Test Protocol (Continuation of Test—Addendum A of Test Protocol)
Although the original test protocol was complete, testing was continued following the similar procedures but at increased loads (240% of rated load) and speeds (up to 2,700 rpm) until a definitive gear failure occurred. Loading was chosen after load analysis on all gearing and components was complete. (*See* Appendix M.)

4.2.4.4. Start-Up Procedure
A specific start-up procedure was followed to ensure proper lubrication and to prevent improper overloads (Appendix N).

4.3. Testing
See Appendix O for detailed discussions of the testing.

4.3.1. Early Test Observations—200% x 200-Hour Test Protocol
Note that the original test load was 77%, and not 200% of rated load as was planned. Much of the early test observations were recorded during this time. Initially, the Convoloid gearbox seemed noisier than the involute gearbox. Once the loading was increased to the true 200% torque, the noise decreased to less than that of the involute. The Convoloid gearing also showed early signs of micropitting around the transition zone. As this wear progressed on the high speed pinion, contact improved across the face width. About halfway through the test, advancement of the micropitting retarded to a point when minimal increases were noticed. Noise from the Convoloid gearbox continued to decrease as the test progressed.

In contrast, the involute high speed pinion began to show signs of micropitting near the root, and this wear progressed until the end of the test. Noise and vibration increases accompanied the resulting apparent degradation in tooth profile. Although the oil for both gearboxes was exceptionally clean, metal particles were evident in the involute samples taken towards the end of the test.

The Convoloid gearbox did not seem to be negatively affected by the application of increased load. As the test progressed, the Convoloid gearing seemed to “wear-in,” as opposed to degrading. After the areas of high stress were removed by micropitting, minimal additional wear was noted. The involute gearing began to wear and micropitting seemed to progress. It also appears that the profile degraded, which caused increased noise.

4.3.1.1. Efficiency
Two methods were used to determine the losses within the test system. From these losses, a composite efficiency for both gearboxes can be determined. The drives only vary in the gear geometry, therefore a valid assumption can be made that there is a negligible difference in efficiency between the two. Thus, the efficiency of each gearbox is assumed to be about half the total losses. The total losses were approximately 5.4%, giving an average efficiency of 97.3% for each gearbox. A determination for the efficiency for each gear mesh can be approximated by removing the power losses due to non-gearing components (bearings and contacting oil seals). Without the losses of these components, the efficiency of each gear set is 98.9%. (See Appendix T for a detailed discussion.)

4.3.1.2. Sound Comparison

Due to the different tooth counts, there were distinctly different sounds coming from the two gearboxes. The involute gearbox had a higher pitch due to having more teeth in the high speed gear set; In comparison, the Convoloid gearbox had a decidedly lower pitch. Based purely on subjective impressions, at low load levels the sound level of the Convoloid gearbox was markedly louder than the involute. When loads were increased, however, the Convoloid gearbox became much quieter than the involute. As testing progressed, the noise level of the involute gearbox steadily increased but no increase was noticeable with the Convoloid drive.

Data taken with AM audio-spectrum analyzer confirmed what was heard by operators. Figure 4.5 is a screen capture showing spikes at the high-speed mesh frequencies of the two gearboxes.

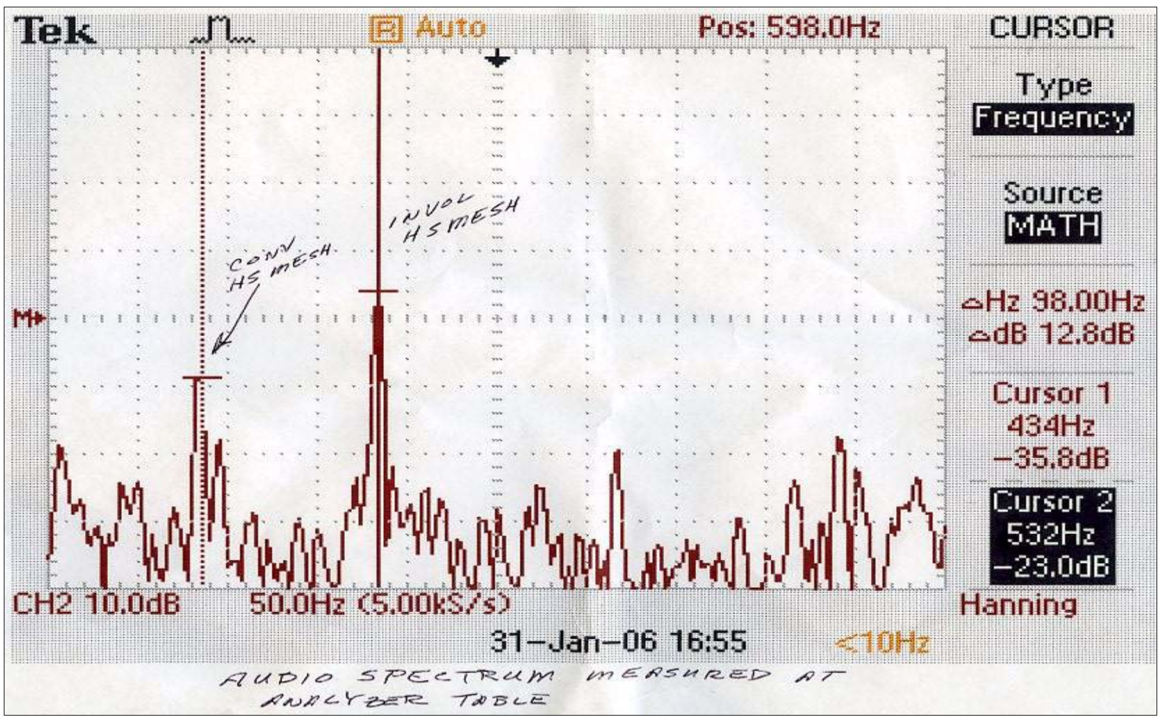

Figure 4.5. Audio (sound) spectrum at a distance of 10 ft after approximately 300 hours of operation

4.3.2.1.1. Vibration

Accelerometers were attached to both gearboxes in an attempt to gather vibration data. One finding of significant interest is that the Convoloid total vibration value continually decreased as the test progressed, and the involute vibration slightly increased. In comparison with the start of the 200% load test, the Convoloid vibration reduced by nearly 10%. Conversely, the involute gearbox experienced nearly a 6% increase in vibration.

Prior to ending the test, a spectrum analyzer using a magnetic pick-up gathered information on the various frequencies of both gearboxes. The nominal high speed frequency was 29.5 Hz (1,768 rpm / 60 sec/m). Using this value, the Convoloid high speed gear frequency was 383 Hz (29.5 x 13 teeth) and the involute high speed gear frequency was 471 Hz (29.5 x 16 teeth). For the respective gearboxes, distinct peaks occur at the fundamental and subsequent harmonic frequencies. The Convoloid fundamental frequency was much smaller than the involute fundamental. In the first, second, and third harmonics, however, peak levels are comparable. Appendix R provides a detailed discussion of this topic.

4.3.2.1.2. Oil Cleanliness

Throughout the test, the ISO cleanliness level for 6 micron (6u) particles in the Convoloid gearbox was 7, according to the online particle counter. The ISO cleanliness level for 6u particles in the involute gearbox was 11. These cleanliness levels were confirmed by daily patch tests for each gearbox. Patches were made into slides and have been preserved. In most of the patches debris was difficult to determine. Towards the end of the test, however, a small number of metal particles were observed in samples taken from the involute gearbox. These were not deemed detrimental to continued operation.

4.3.2.1.3. Oil Temperature

The oil sumps and all individual bearing temperatures were monitored and in no instance did any bearing exceed 170°F. The hottest bearing in the Convoloid gearbox was the outside low speed bearing with a maximum temperature of 167°F which, in the worst case, was 1°F above its sump temperature. The hottest bearing in the involute gearbox was the inside intermediate bearing with a maximum of 163°F which was also only 1°F above that sump temperature. Both reservoirs were controlled to a maximum of 161°F and a minimum for 145°F. Variations in the oil temperature primarily are due to the cycling of the heat exchanger cooling fans.

Temperatures for the involute gearbox tended to be lower than the Convoloid due to the high air flow (approximately 700 fpm) blowing from the motor’s fan. Experimentation with external fans blowing on the Convoloid gearbox caused dramatic drops in oil sump and bearing temperatures. Similarly, when the air flow from the motor was blocked temperatures dramatically rose in the involute gearbox. Both gearboxes generated enough heat to cycle their respective heat exchanger fans, therefore this variation was not deemed detrimental to the test.

4.3.2.1.4. Micropitting

Micropitting is a microscopic removal of material that produces a “frosted” or matte-gray surface. The topic of the failure mode of micropitting is debated fervently in the industry, and many respected experts in the field have differing views. Few that agree as to the underlying causes or as to the effects of micropitting on overall gear tooth life. It is agreed that micropitting occurs, however, and that in some instances it can lead to macropitting and tooth breakage.

In this test, it appears that micropitting began in areas of high stress and at the boundaries of contact. High stress can be caused by excessive material in the tooth profile, edge and tip loading due to bending of the supporting shaft and the tooth itself, and high points within the contacting surface due to grinding marks and surface finish. In normal operation, gear teeth are separated by an oil film caused by an increase in lubricant viscosity due to very high elastohydrodynamic pressures. At the boundaries of contact, oil pressure abruptly drops causing a much thinner film, potentially allowing metal-to-metal contact. Solutions to these gear-design challenges required detailed study and careful manufacture.

4.3.2.1.5. Increase in Test Load to “True” 200% of Rated Load

The high speed pinion in the Convoloid gearbox experienced micropitting early in the test when subjected to only 77% of the rated load. The wear area directly corresponds with areas of positive—or excess—material. As the test progressed, the other components also exhibited this wear forming a narrow line along either side of the transition zone, at the edges, in the root, and at the tip. Near the transition zone, this line of wear broadened under the 200% load until after approximately 100 hours of operation. Further wear abated and did not seem to spread after this point. The initial micropitting along the edges, root, and tip did not continue to grow as the test progressed (*see* Figure 4.6).

Figure 4.6. Convoloid high speed pinion at conclusion of 200 x 200 testing; note that although there is extensive micropitting on both sides of the transition zone, it is not rapidly progressing (235.8 hours at 200% of the rated load)

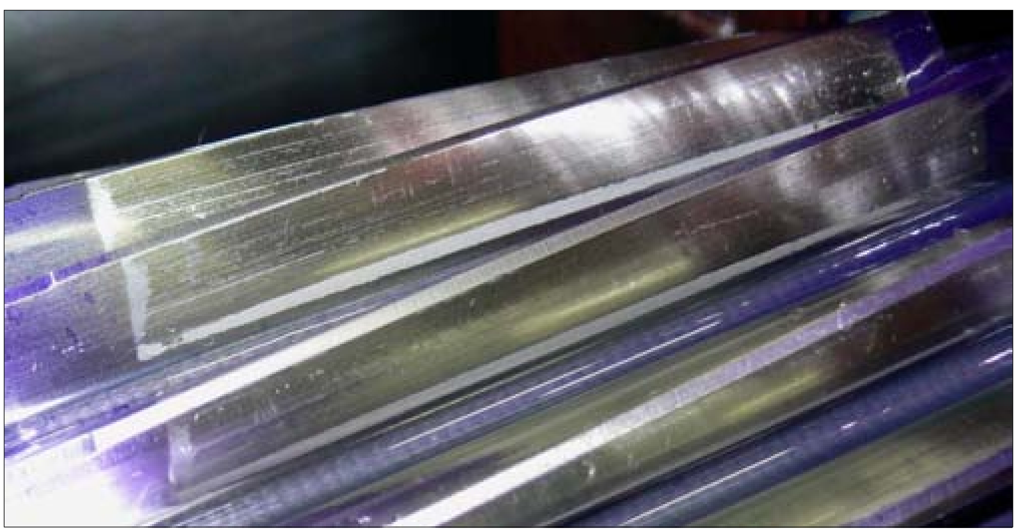

Figure 4.7. Involute HS pinion at the conclusion of 200 x 200 testing; note the extensive micropitting in the dedendum and at the edges, some micropitting is visible along the tip, this wear is progressing (235.8 hours at 200% load)

The involute high speed pinion developed very light micropitting late in the 77% load test. With the increase in load to 200%, wear progressed steadily until the test was stopped. A broad line of wear in the dedendum along the entire effective face width, at both ends and, to a lesser extent, at the tip is indicative of progressive micropitting (*see* Figure 4.7). The loss of profile due to micropitting caused a noticeable increase in noise from the involute gearbox. Note also the faint wear at the top of the grinding marks. No indications are apparent to predict any impending catastrophic failures with either gearbox.

4.3.2.1.6. Post-Test Action

To determine the extent of micropitting around the transition zone in the Convoloid high speed pinion, the pinion was removed from the gearbox and sent to the North Kingston, Rhode Island, facility of Brown & Sharpe for inspection on a Leitz Precision Measuring Machine. Using a 0.50-mm diameter inspection probe, the plotted profile of an untouched portion of the active flank showed excess material (0.0004 to 0.0007) at the extremes of the transition zone, and extending into the addendum and dedendum of the pinion. The plot of a worn portion of the tooth showed distinct areas of material removal around the transition zone.

4.3.2.1.7. Preliminary Conclusions

Profile errors found during the post-test inspection of the Convoloid high speed tooth profile and improper tip relief (as seen on the high speed gear in Figure 23) caused greater than calculated stress loads across the flank of the gear teeth. In contrast, wear patterns on the involute gear meshes confirm that the teeth were manufactured to a high-quality, AGMA 12 accuracy. Variations in the Convoloid tooth form do not conform to as high an accuracy grade; therefore this test is a comparison of an “A” involute gearbox with a “B” Convoloid gearbox. The Convoloid gearbox has functioned as well as the higher precision involute gearbox.

Profile appears to be very important for the Convoloid gearing. Variations in the profile seem to “heal” or repair themselves, however, during a wear-in process. What affect this micropitting has and whether the wear will progress to macropitting is unknown. Conversely, once micropitting begins on the involute gears the wear appears to be progressive, and it can lead to degradation in tooth profile causing increased noise and vibration. A detailed discussion of the analysis of vibrations within the test fixture is given in Appendix R. Additional observations by the test supervisor are included in Appendix S.

Although analysis of the individual efficiencies of the gearboxes is not practical in this test due to existing instrumentation, the total amount of power lost (as supplied by the motor) is in accordance with the average efficiency of two standard gearboxes. See Appendix S for the derivation of gearbox efficiencies.

4.3.2. Test Observations—Extended Testing

Having completed the testing protocol (200 hours at 200% load and 50% faster speed) during December 2005, it was determined that the test results were not definitive. Extensive discussions resolved that the test should continue until a failure occurs.

To reduce the length of the test, investigations were conducted to determine whether the applied load and the motor speed could be increased. Analysis of the limiting factors (gear tooth life, key stress, and shaft strength) confirmed that and increase of load to 240% of the operating torque was viable. Speed of the test motor was increased to as much as 2,700 rpm, which increased the number of load cycles occurring in a shorter period. An addendum to the test protocol was developed (Appendix M). For reference, note that 240% of the rated load is 515,000 lb-in torque versus the standard 214,500 lb-in.

Testing resumed on January 23, 2006, and followed the test protocol addendum. Once the oil and bearing temperatures stabilized with a 200% load and 1,768 rpm, the load was increased to 220% and then to 240%. No abnormal increases in temperature were observed and sound levels did not noticeably increase.

Motor speed gradually was increased to apply as many load cycles as possible in the shortest amount of time. When the speed reached 2,350 rpm, the rubber coupling cover for one of the constant velocity couplings ruptured. Upon consultation with the manufacturer, an operational speed of 2,050 rpm was chosen. After the coupling was repaired, testing resumed.

After 67 hours of operation at the 240% load (302 total test hours), micropitting on the involute high speed pinion teeth (Figure P.28) appeared to have spread in the pinion root and along one edge (*c.f.* Figure P.20). Wear on the Convoloid teeth (Figure P.29) did not appear much changed from the end of the previous test (*c.f.* Figure P.24).

Vibration levels in both gearboxes appear to change as the test progressed. Vibration in the involute gearbox gradually increased. Conversely, vibration in the Convoloid gearbox slowly decreased. Spectrum analysis of these vibrations reveals a substantial difference in the two gearboxes. High speed mesh frequencies at 2,000 rpm are 533 Hz for the involute and 433 Hz for the Convoloid. Figure 4.8 and Figure 4.9 compare the high speed mesh frequency for the involute and Convoloid respectively. When measured at the respective gearbox, the fundamental frequency for the high speed mesh is 19.2 dB higher with the involute gearbox.

Additionally, an audio sound spectrum taken from approximately 6 feet away shows a very distinct difference in noise level between the two gearboxes (Figure 4.10). The involute gearbox is 12.8 dB higher than the Convoloid gearbox, with the primary noise occurring at the high speed gear mesh frequency.

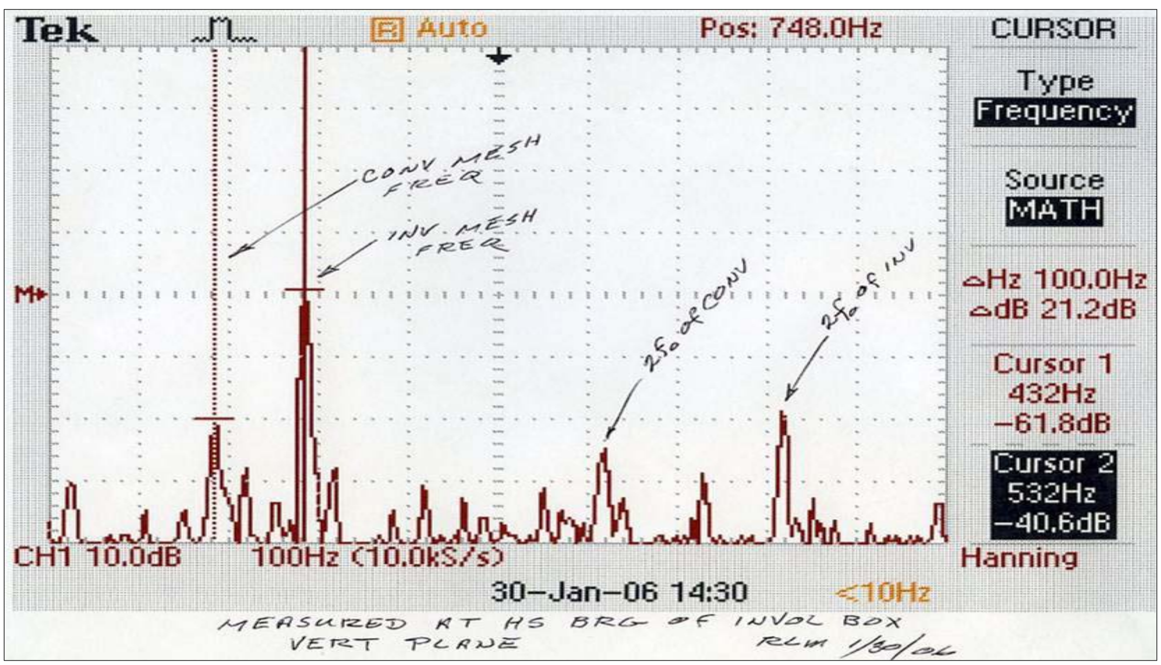

Figure 4.8. Vibration spectrum for involute gearbox as measured at the high speed bearing; value for the involute high speed gear mesh (533 Hz) is shown in the black box (Cursor 2); transmitted vibration from the Convoloid gearbox (433 Hz) is listed as Cursor 1; secondary peaks are harmonics

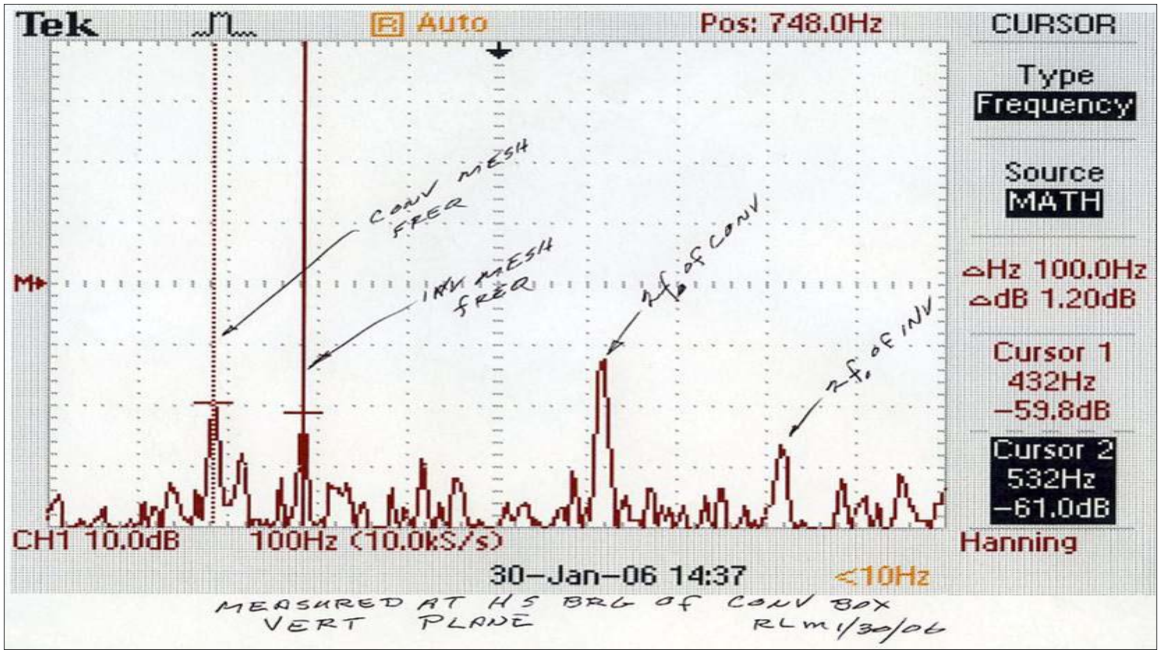

Figure 4.9. Vibration spectrum for Convoloid gearbox as measured at the high speed bearing; value for the Convoloid high speed gear mesh (433 Hz) is shown in as (Cursor 1); transmitted vibration from the involute gearbox (533 Hz) is listed as Cursor 2

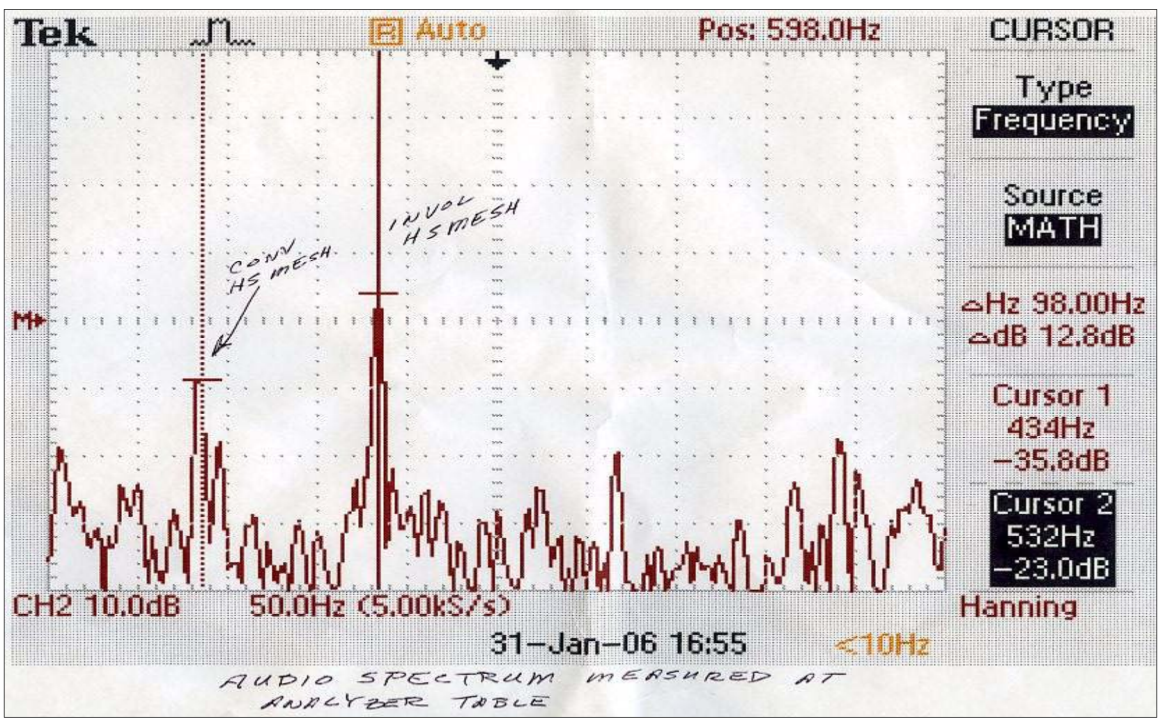

Figure 4.10. Audio (sound) spectrum at a distance of 10 feet after approximately 300 hours of operation

4.3.3. Cessation of Test

4.3.3.1. Bearing Failure in Micon 108 Test

On March 1, 2006, noise levels and unexplained fluctuations in an intermediate shaft bearing temperature in the Convoloid gearbox prompted the test monitor to shut down the test. Significant damage was present on the rollers and cage (Figure 4.11). Pieces of the cage and roller material passed through gear meshes and other bearings (Figure 4.12). Significant quantities of metal were recovered in the gearbox sump, oil reservoir, and primary filter. A day prior to the bearing failure, a magnet inspection of both gearbox sumps revealed no metal particles. After the failure, extensive quantities were found inside the gearbox housing.

Figure 4.11. Failed spherical roller bearing in Convoloid gearbox after 763 total test hours; note pitted edges of the rollers and deformation of the cage; metallic particles are present at the bottom of the housing bore

Figure 4.12. Pieces of the failed bearing

Analysis of temperature and vibration data taken during the final day of operation, show a distinct increase in vibration levels about 8 hours before shutdown (Figure 4.13, Figure 4.14). An observation by the operator that was written in the test log at the time of the increased vibration indicates that there was "a change in the sound" of the Convoloid gearbox. However, testing continued until a more dramatic failure occurred.

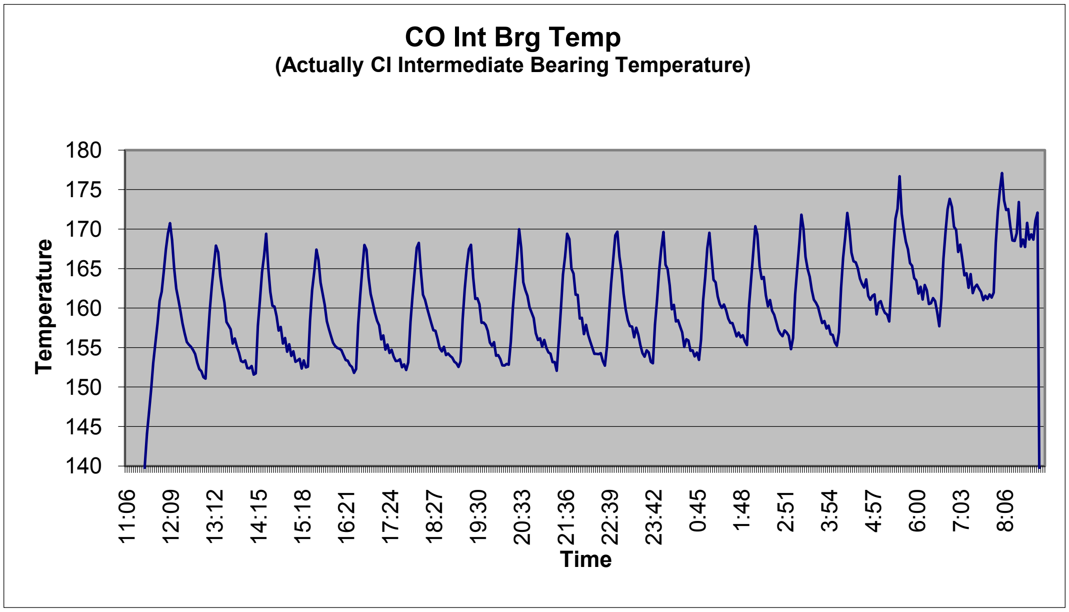

Figure 4.13. Bearing temperature chart for the failed Convoloid intermediate bearing; note the rise in temperature after 2:18 a.m.; the jagged "saw-tooth" shape is due to cycling of the heat exchanger fan

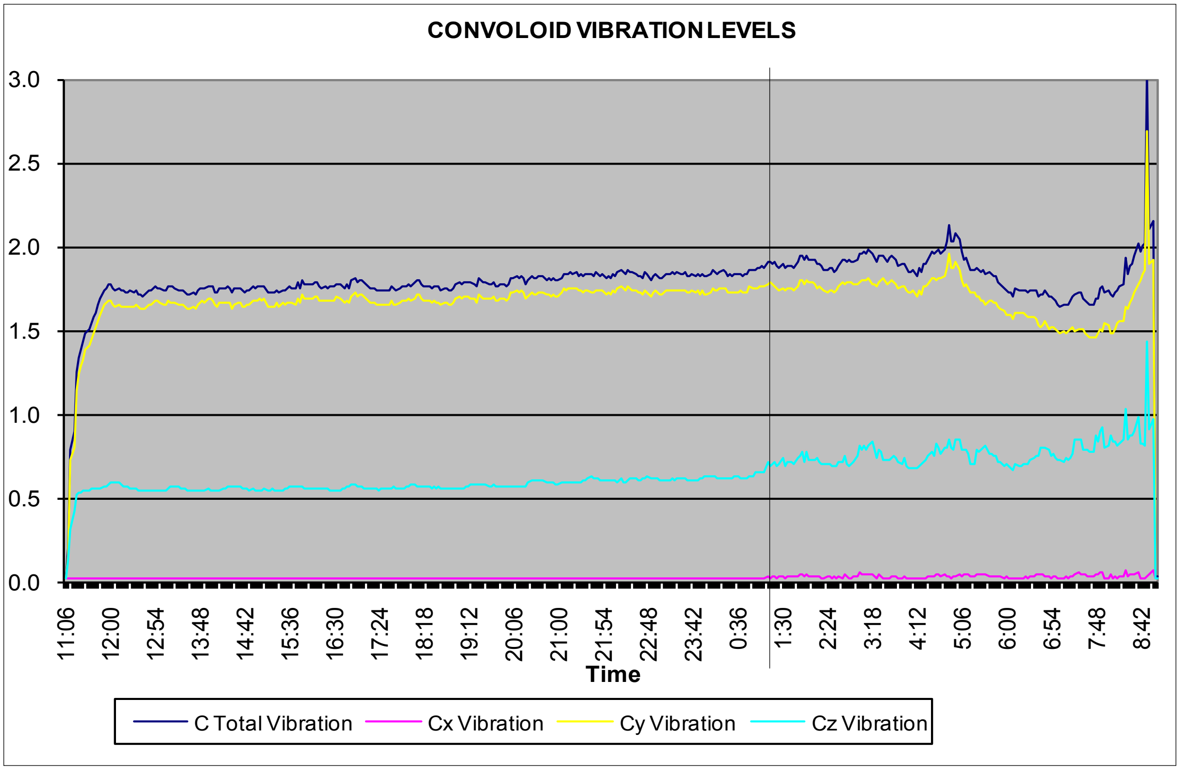

Figure 4.14. Vibration levels for the Convoloid gearbox for the last day of operation; note the increase in vibration at approximately 1:00 a.m.

4.3.3.2. Condition of Gearing at End of Test

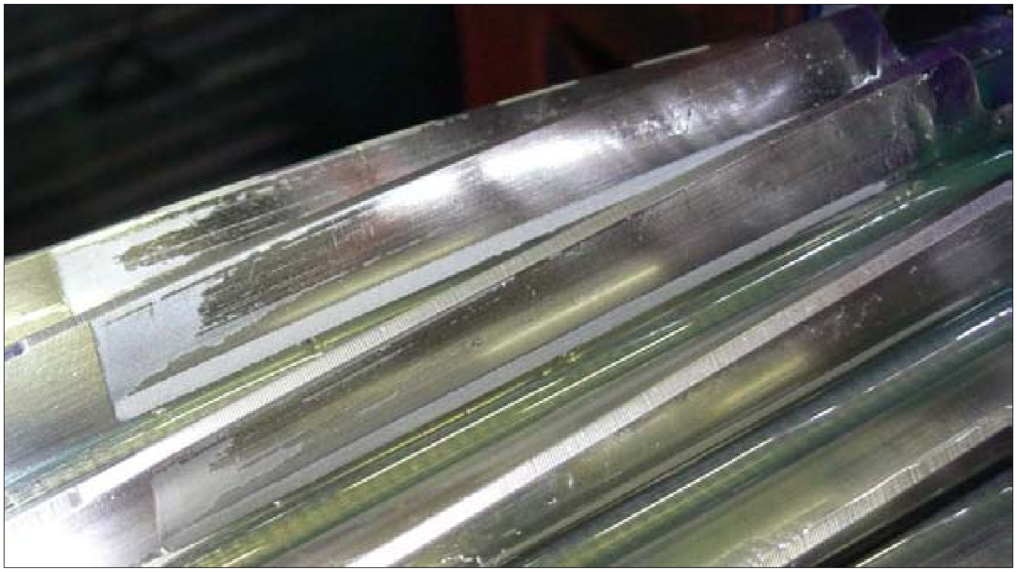

Figure 4.15. Involute HS pinion at the end of test; wear due to micropitting is extensive along one side (235 hours at 200% load; 528 hours at 240% load)

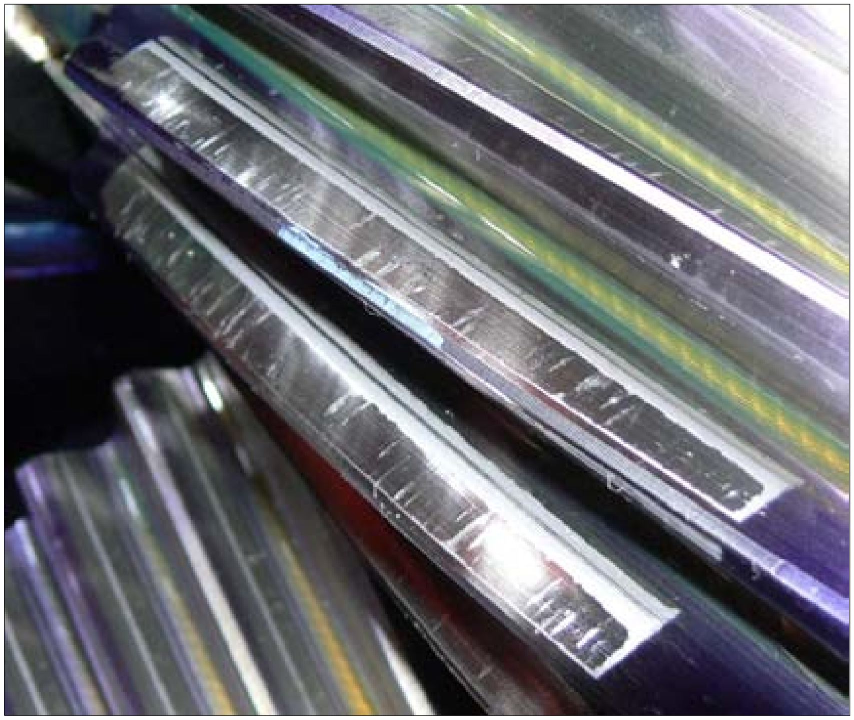

Figure 4.16. Convoloid HS pinion at the end of test; wear due to micropitting is expanded slightly during the test, but no progression occurred on the edges (235 hours at 200% load; 528 hours at 240% load)

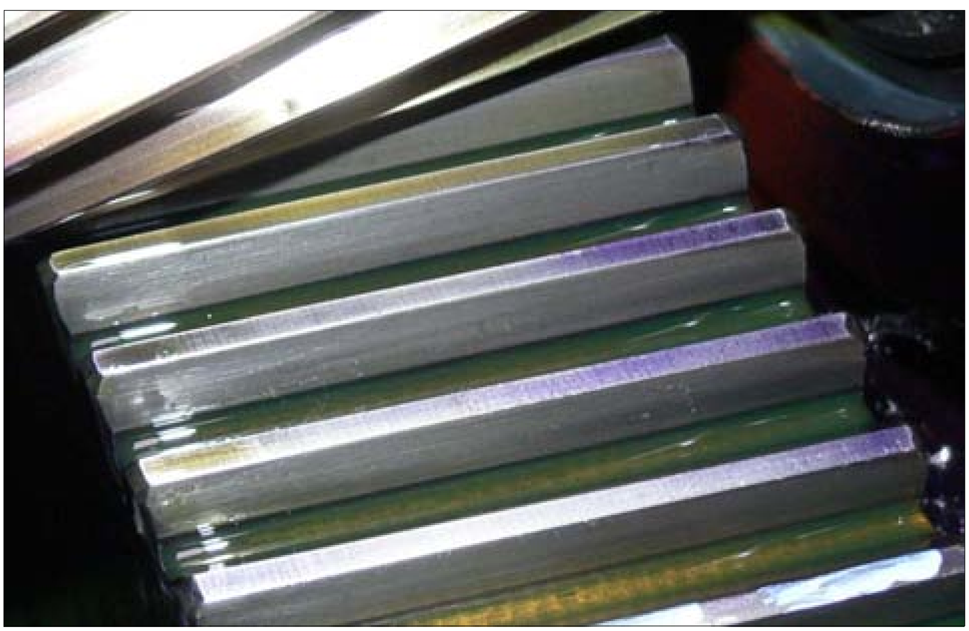

Figure 4.17. Involute HS gear at the end of test; no wear is readily visible (235 hours at 200% load; 528 hours at 240% load)

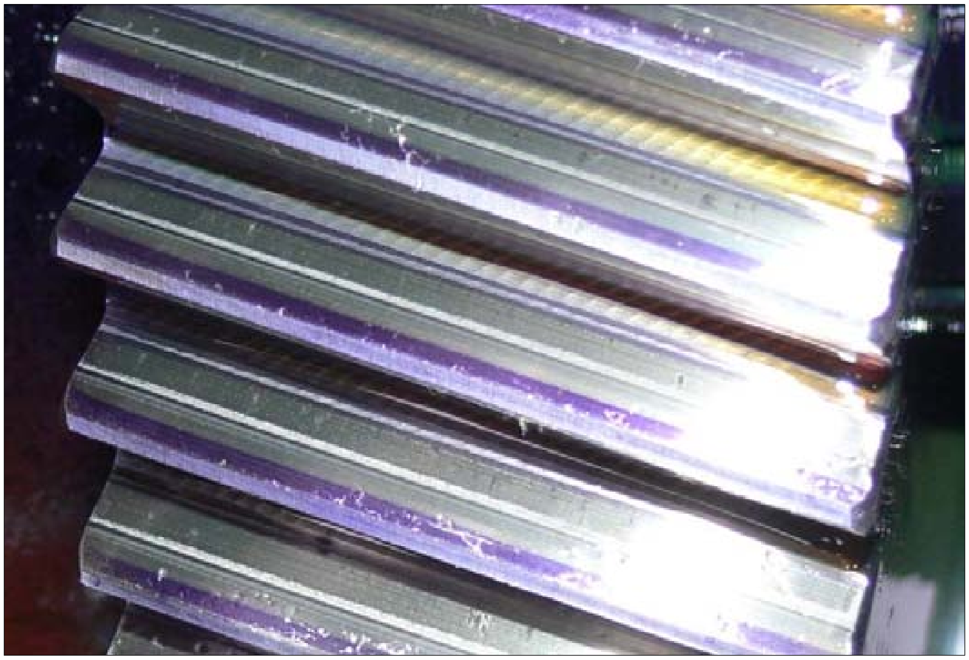

**Figure 4.18. Convoloid HS gear at the end of test;
some wear due to micropitting is visible on the dedendum side of the transition zone; a large section of the addendum still shows marking dye—indicating too much tip relief in the profile
(235 hours at 200% load; 528 hours at 240% load)**

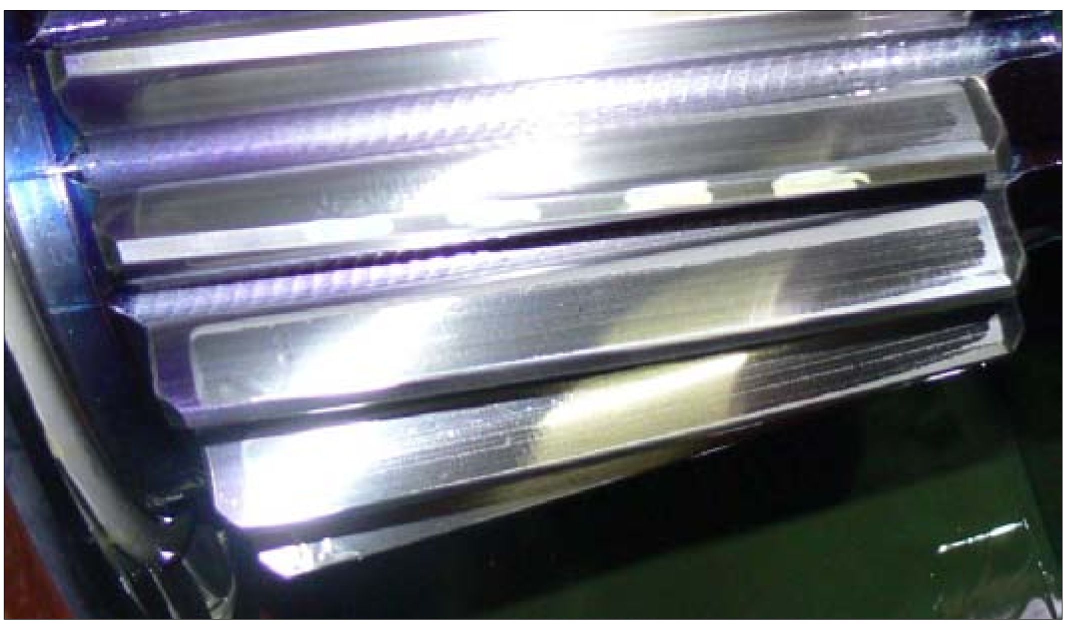

Figure 4.19. Involute LS pinion at the end of test; similar to the HS pinion, micropitting wear is concentrated along one side and in the dedendum (235 hours at 200% load; 528 hours at 240% load)

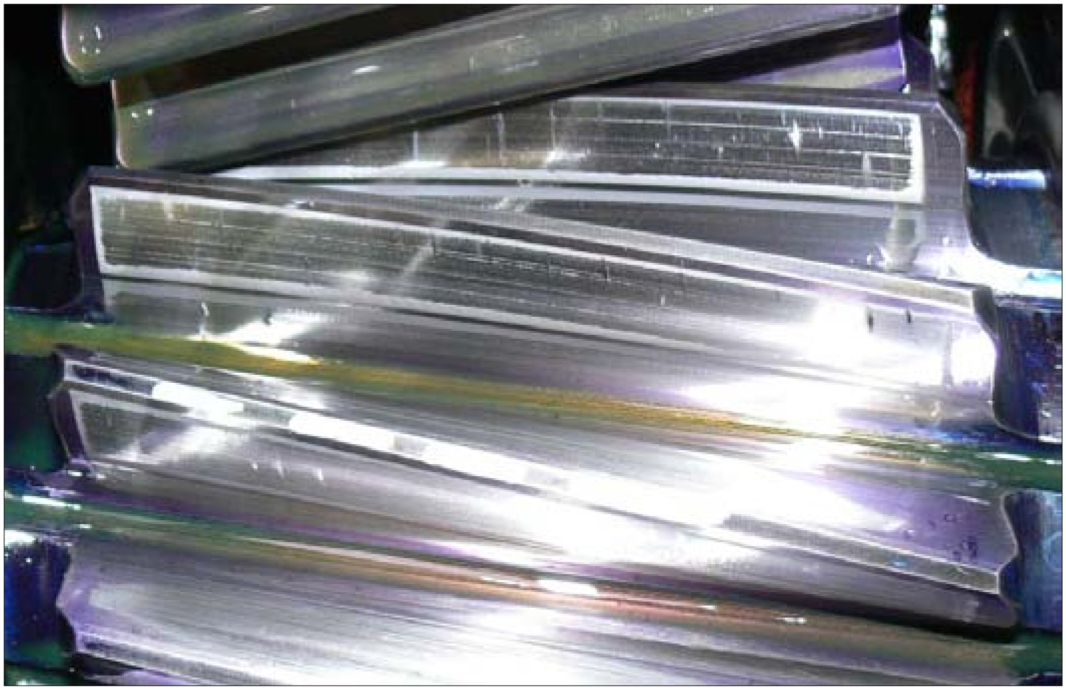

Figure 4.20. Convoloid LS pinion at the end of test; wear due to micropitting brackets the operating surface, except for in the dedendum due to proper tip relief on the mating gear (235 hours at 200% load; 528 hours at 240% load)

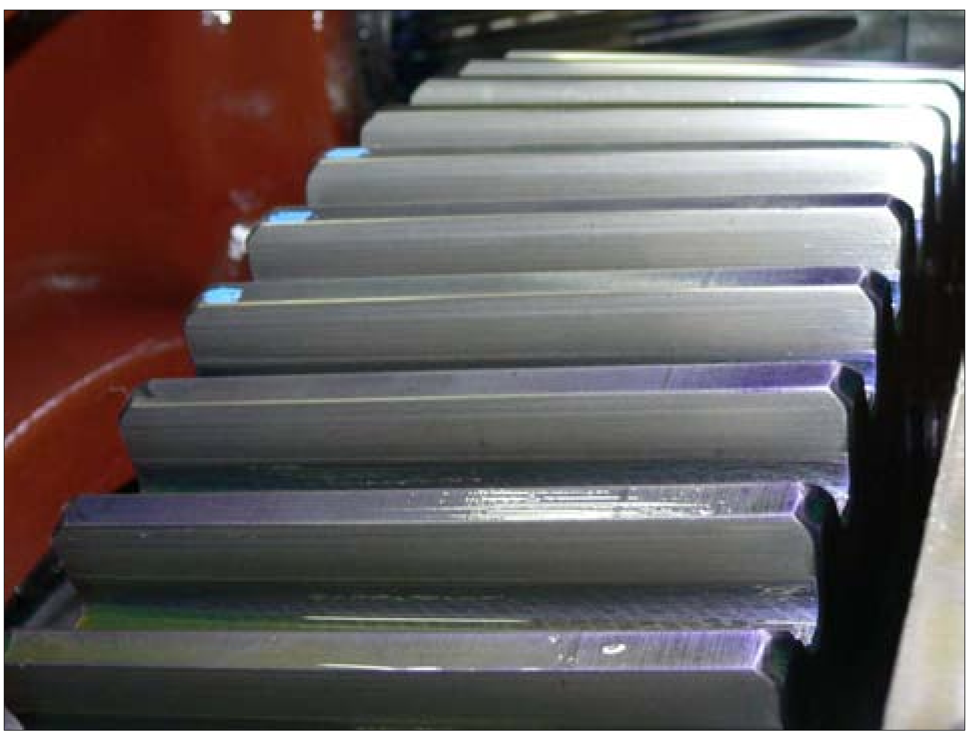

Figure 4.21. Involute LS gear at the end of test; no wear is visible (235 hours at 200% load; 528 hours at 240% load)

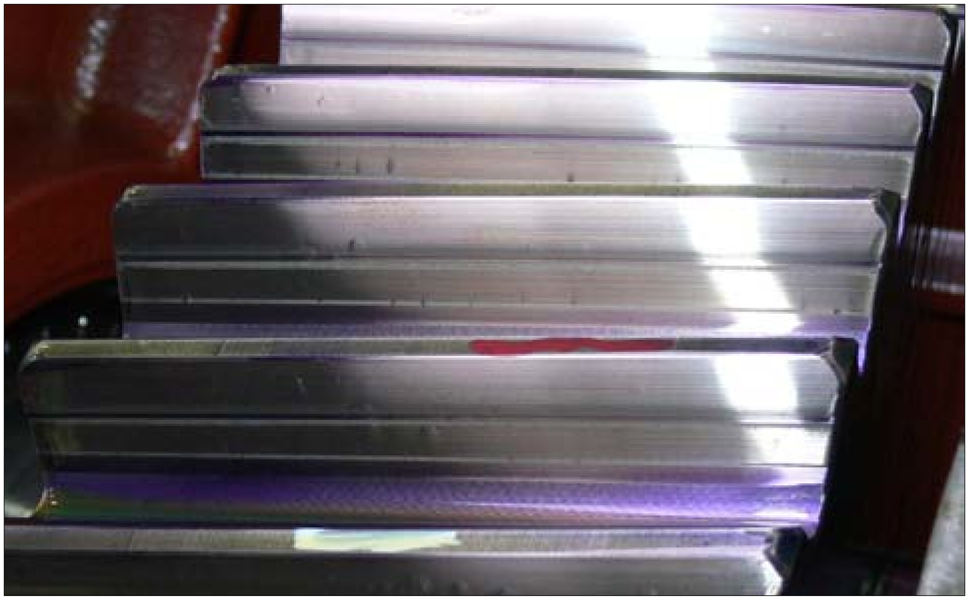

Figure 4.22. Convoloid LS gear at the end of test; slight wear due to micropitting is visible in the dedendum (235 hours at 200% load; 528 hours at 240% load)

4.3.4. Variations During Test

A few situations where variability could have been introduced to the test must be noted.

4.3.4.1. Incorrect Torque Application

Fortunately, the mathematical error in calculating the applied torque from the Lebow transducer caused less (rather than more) torque to be applied. Also, because this test ultimately was to compare the Convoloid design with an involute, both gearboxes were loaded equally. Thus the comparison remains valid.

4.3.4.2. Bearing Lubrication Method

Lubrication was supplied to each bearing via a manually operated orifice valves. An operator was required to constantly monitor bearing temperatures to make certain proper oil quantities were being delivered. Although alarm conditions exist for excessive temperatures, smaller variations could have caused early bearing wear should the test have been continued. The failure of the intermediate bearing in the Convoloid gearbox appears to have been caused by metal fatigue and was not attributed to lubrication.

4.3.4.3. Lubrication Filtering Before Heat Exchanger

The presence of “smears” on the active surface of the gear teeth (easily visible in Figure 4.23) indicates that some small particles went through the mesh. No damage could be felt due to these marks, and old marks tended to fade as the test progressed. Oil flowed through the heat exchangers after the moving through filters, therefore it is thought that particles flushed from the exchangers could have caused the marks. These particles would have been soft aluminum, which would explain the lack of damage to the tooth surface. Any particles also would have caused variations in the orifice valves feeding the bearings.

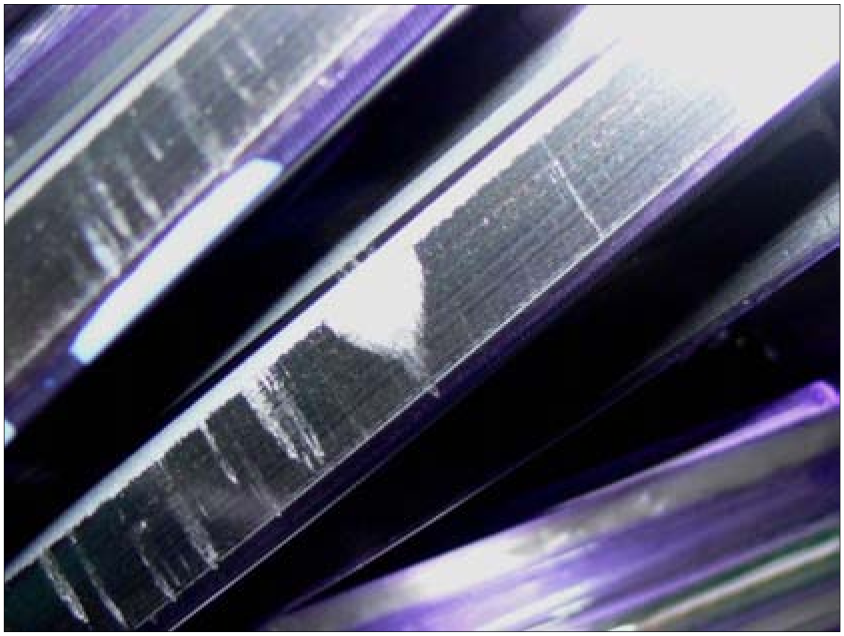

Figure 4.23. Close-up of oil-debris damage on the Convoloid HS pinion; this mark began to fade at the next inspection (91.2 hours at 200% load)

4.3.4.4. Inadequate Oil Flow Out of the Test Gearboxes

Operators constantly had to balance the amount of oil flow to prevent overfilling the gearboxes due to an inadequate draining of the gearboxes. This was understood early in the test; however, modifying the test fixture to raise the gearboxes to provide a smoother flow into the oil reservoirs would have substantially delayed testing and increased costs.

4.3.5. Load/Power Summary

Table 4.5 shows the loads, speed at the generator, and power summary for the Micon 108 comparison test.

Table 4.5. Load/Power Summary for Micon 108 Test
Original Rating 108 kW (145 hp), 1,200 rpm at Generator

% Rated Torque	Speed at Generator (rpm)	Circulated Test Power (kW)	(HP)	Test Time (hr)
77	1,800	94	126	179
200	1,800	242	324	235
240	2,350	386	518	6
240	2,000	432	579	125
240	2,050	443	594	420

It should be noted that the test loads included system inefficiencies. Measurements were taken which indicated a total gear inefficiencies and lubricant churning losses to be about 5.2%. Knowing the approximate losses from the involute gearbox through conventionally accepted calculations, it was inferred that the total losses through each gearbox—involute and Convoloid—were approximately equal at 2.6%. Gear mesh efficiencies were approximately 98.9%. *See* Appendix T.

When test speeds were increased to 2,350 rpm (high-speed shaft speed) the CV joint lubrication covers (made of rubber) failed, thus a decrease in motor speed to 2,050 rpm was necessary. A majority of the test time occurred at this speed. After 965 hours of total test time, the spherical roller bearings of the intermediate bore in the Convoloid gearbox failed. Damage to nearby gearing caused testing to be stopped.

4.4. Test Results

Testing of the existing gearboxes was halted pending analysis of the bearings and feasibility of correcting damage due to debris in the gear mesh. Initial plans for replacing the failed bearing and continuing testing were dismissed due to the amount of damage sustained by the Convoloid gear teeth. Any further testing would require replacement of the damaged gearing, thus negating any previous test results. It was emphasized that completion of a laboratory test that shows a definitive comparison of the two gearing types is imperative. This is necessary before any field testing can be started.

A field test also is necessary. Due to design modifications in the field, a change in ratio is required for the field test gearboxes. To maintain correlation between the laboratory and field tests, gearboxes used in both tests must be of the same configuration and ratio, therefore gearing changes are required. Additionally, because the failure mode of the test gearboxes was in the bearings, a new bearing configuration is required. It was agreed that turbine gearboxes in the field should have filtered and cooled lubrication oil and that the oil should be applied directly to all bearings.

4.5. Post-Test Analysis

4.5.1. Housing Stiffness / Bearing Redesign for Micon 108

The bearings are critical to the updated gearbox configuration. Due to failure of the spherical roller bearings in the intermediate shaft of the laboratory test, a new bearing design method was required. Before any bearing selection could be conducted, however, a test was conducted on the gearbox housing to determine its stiffness at the intermediate bearing location.

Mr. Richard L. Meredith (DC Energy) set up one of the test gearbox housings with a hydraulic ram inside to measure deflection based upon an applied load (Figure 4.24). A dial indicator was used for measurement. After the equipment was installed, the housing cover was assembled to the gearbox as was done with the test (i.e., with a bolted connection and all four taper pins installed). Application of hydraulic pressure created a force that is trying to separate, or spread the housing. At the nominal rated load (108 kW) the deflection due to the resultant gear thrust forces (~3,525 lb thrust load) was approximately 0.0055 in. As the load was increased to the new design load, deflection increased to approximately 0.017 in.

These values greatly exceeded expected values and negated any bearing selection that can employ opposed roller bearings for the intermediate shaft. Instead, a double-row bearing design was developed. For this type of configuration, a double-row tapered roller bearing is employed on one end of a shaft and a cylindrical roller bearing is used at the other end. The tapered roller bearing resists all gearing thrust forces, allowing the cylindrical roller bearing to carry only radial load. A similar arrangement is intended for the high speed bearings. Investigation into limitations of the low speed cylindrical bearings found that the existing bearings are sufficient to properly support expected loads.

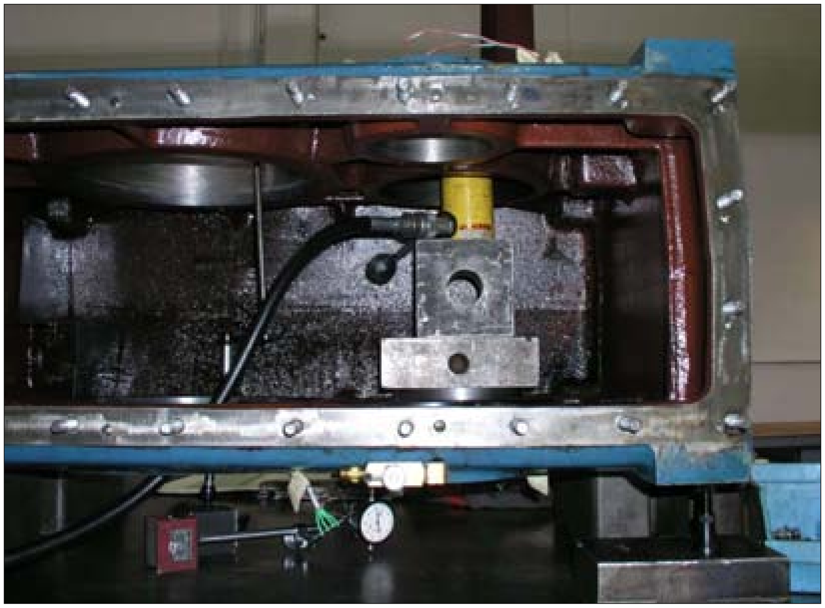

Figure 4.24. Housing stiffness test configuration

4.5.2. Convoloid Profile Modification (Tip Relief) Study

Determination of the proper Convoloid tooth profile is critical for minimizing localized stresses and premature wear. A full discussion is included in Appendix T. In summary, deflections of Convoloid gear teeth appear to be 10% to 20% less than an equivalent involute tooth when subjected to the same loads. The exact amount of deflection cannot be assumed due to the approximate nature of the analysis. As a comparison, however, this analysis has definitely shown that the Convoloid tooth form is much stiffer than the involute.

Inspection of the Convoloid high speed pinion allows for a good discussion on tip relief. Figure 4.25 shows the pinion after 154 hours operation at 77% of rated load. A little bluing is visible at the tooth tips (probably due to the chamfer) and on each end (due to crowning), but contact appears smooth at the tip with no signs of wear. When the load was increased to 200%, slight wear appears at the tip (Figure 4.26). A further increase to 240% reveals distinct tip wear (Figure 4.27). It therefore appears that the applied tip relief could be appropriate for the light load, but it is not enough for a heavier loading.

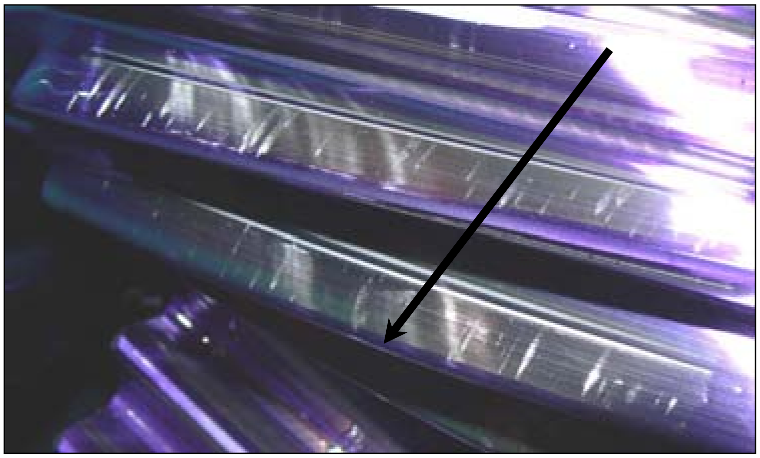

Figure 4.25. Convoloid HS pinion (77% load, 154 hours)

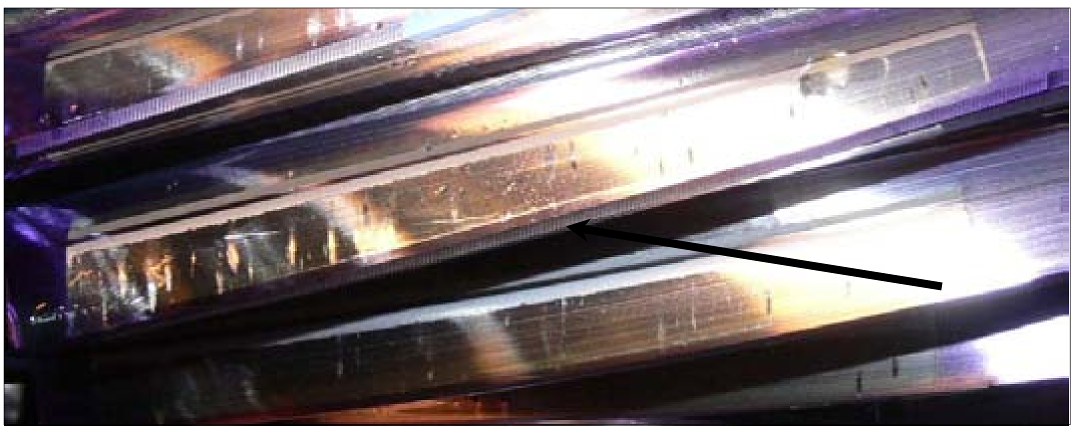

Figure 4.26. Convoloid HS pinion (200% load, 235 hours)

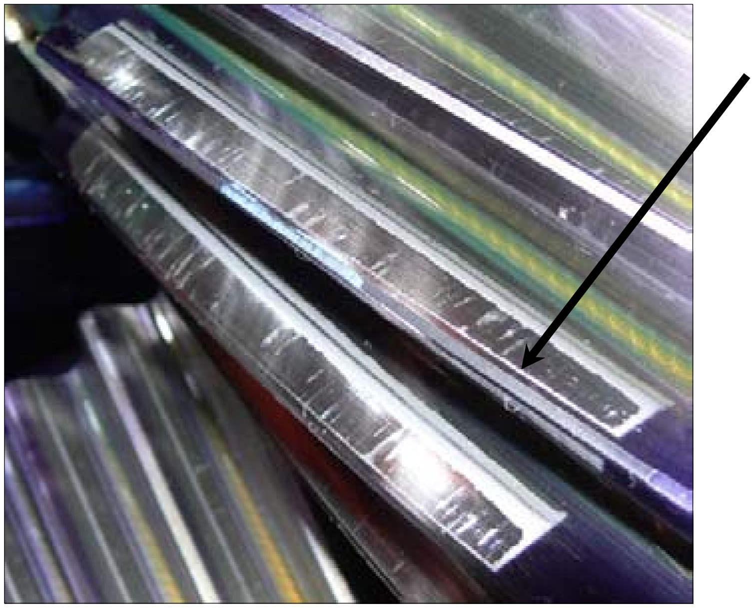

Figure 4.27. Convoloid HS pinion (240% load, 763 hours)

The computer analysis indicate that deflection of the gear teeth appears to be not just a function of the load—both tooth size and number of teeth make a difference. Additionally, trying to relate the amount of correction to an equivalent involute gear might be difficult, although for larger numbers of teeth the relationship could be more consistent. Determination of the proper amount of tip relief necessary to counter tooth deflection for Convoloid gears requires additional study. At the time of this writing, using an approximate relationship for the amount of tip relief could be related to an equivalent involute.

4.6. Test Conclusion

4.6.1. Test Limitations

Attempting to qualify the Convoloid tooth form in a production wind turbine gearbox is not cost effective due to factors including:

- Flexibility of housing creates variations that cannot be controlled;
- Assembly/disassembly of components within the full-scale gearbox with a single-piece housing design was laborious and very time consuming;
- Cost of individual parts was very high; and
- Size of the test fixture is cost prohibitive for numerous tests.

When test experiences required modifications to the Convoloid profile, the entire test fixture had to be disassembled. Subsequently, there was a hesitancy to make multiple changes.

4.6.2. Conclusions

Upon analysis of the failed bearing and the resulting damage to the Convoloid components, it became clear that additional testing in this test environment would be cost prohibitive. An alternative to conducting profile development on the existing test apparatus is to build a much simpler and less expensive test gearbox. This configuration (referred to as the 3-Hole Test) features a pinion driving an intermediate gear which in turn is driving another pinion (*see* Figure 4.28).

Two gearboxes were built (one Convoloid and one involute) and arranged in a back-to-back test configuration. An advantage of this setup is that the ratios of the two gearboxes need not be exact because the second pinion will have the same number of teeth as the first pinion, giving a relative ratio between the input and output shafts of 1.00:1. Gear designs for both can be optimized for test loads, thereby offering a true comparison. Additionally, more tests can be conducted (two for each gearbox) giving more data points and lessening the chances for statistical aberrations.

The project situation meeting held May 18, 2006, featured a summary of the project to date including results of the testing conducted in Seattle, Washington; expected field test configurations and gearbox quantities; expected cost of energy analysis including the longer durability of a complete gearbox system design and additional energy production allowed by higher-capacity gears and bearings; and discussions of the benefits of a 3-hole type test. The 3-Hole Test discussion addressed topics including:

- Long test runs are required to get a surface durability failure with the Micon 108,
- Assembly and disassembly of the Micon 108 design is cumbersome and labor intensive,
- A "3-Hole Test" is smaller and much easier to work with,
- Gear change-outs are much easier due to a split-housing design,
- The idler gear simulates a planet gear with reverse loading,
- Cost of the gears is much less than those used in the current test,
- Time to surface durability failures is reasonable due to the smaller size,
- Bearings can be sized to prevent bearing failures, and
- Conducting a number of tests increases reliability of test results with Weibull charts or distribution plots.

The overwhelming sentiment of those attending the meeting was that the 3-Hole Test would be of greatest benefit to the industry in that more information would be realized for a specific cost. Following as a close second was the COE analysis utilizing site-specific wind-load bin arrays applied to the proposed system redesign. Careful budgetary projections will allow for the 3-hole testing program, however the engineering, manufacturing, and deployment of the field test units have been suspended.

Preliminary design work commenced on the 3-Hole Test gearbox upon the conclusion of the situation meeting. Design power was limited to 180 hp in an attempt to keep the test gearbox small enough that testing time would not be too long. The test should be able to run unattended and incorporates a "dry sump," in which the gearing does not dip into the lubricating oil. All bearings and gear meshes are directly supplied with filtered and cooled oil. Figure 4.28 is an early representation of a test gearbox.

4.6.3. Analysis of Test Results—Micon 108

The 4-square test rig was shut down periodically, sometimes more than once a day, to examine the gear teeth for failure examinations. Photos were taken to capture detail; these can be found in Appendix O and Appendix P. The following observations were made, taking the entire test run in context.

- Stress concentration factors which have a major bearing on tooth bending stresses were confirmed to be very adequate for both the involute gearbox (expected due to the long history of theory versus practice for this tooth form) and the Convoloid gearbox. Early in 2003, Genesis Partners LP completed 4-square testing of Convoloid gearing where bending performance was not satisfactory. A comprehensive finite-element analysis (FEA) of the Convoloid form was executed with the result of much improved protocols and values for this important gear parameter. This information was used in the design of the Convoloid gear pairs for the Micon 108 gearboxes.

- Tip relief calculations and performance worked very well for the involute gear pairs. This performance could be expected for involutes for the same reasons indicated above. The Convoloid was another matter, however. The much lower bending stresses of the Convoloid designs—and thus lower load intensities—affected a recalculated protocol for "LVR" design protocols using that data. The test results were "fair" in this regard. This area is one for future study and development in the refinement of Convoloid technology.

4.6.3.1. Micropitting

The involute gearing showed micropitting early in testing that progressed until the test was shut down. Some of this could be due to slight lead (helix angle) variations as well as hard contact in the root of the high speed pinion. The Convoloid gearing, especially the high speed gear set, showed early signs of micropitting around either side the transition zone. As the test proceeded under the full 240% load, the thickness of the micropitting wear (0.020 in to 0.030 in) did not grow after the initial 100 hours or so (see Appendix P for photos). This test result was studied carefully—including conferences with fluid dynamics experts—and new design approaches were developed for future Convoloid designs. One cause for the premature wear was confirmed later with gear inspection findings of excessive material around the transition zone. This manufacturing deviation caused high localized contact stresses that resulted in the premature wear. Once material was removed, no further expansion of the micropitting occurred. Investigations into lubricant interaction with contacting gear tooth flanks have also led to slight modifications in the Convoloid tooth form to improve tooth contact under load.

4.6.3.2. Macropitting

Although for themajority of the test, the applied power was at or above 200% of the rated power, there was no evidence of macropitting for either the involute gearing or the Convoloid.

4.6.3.3. Spherical Bearing Failure

Early in the design stages of both the involute and Convoloid gearboxes, a bearing analysis was made to assess the load-carrying capacities of all bearings based on the elevated loads anticipated. These loads mainly were due to gear forces in the tangential, radial, and axial directions. The spherical roller bearing at the intermediate shaft showed a 3,532-hour life at the 200% rated load level. It was thought that this calculated life could be significantly lengthened because all bearings in each gearbox were to have individual lubrication lines with individually controlled oil flows to keep temperatures in reasonable limits. Temperature sensors also were strategically placed to help this effort.

The results, however, were not anticipated. At 965 hours of accumulated test time, a catastrophic and sudden failure of the Convoloid spherical roller bearing occurred. Approximately 25 minutes before failure occurred, the test operator noticed a slight rise in the oil temperature of this bearing. Oil flow to that bearing was increased and nothing suggested an oncoming failure because the temperature rise was small and the oil flow increase appeared to rectify the problem. See Appendix O for a complete discussion.

Factors contributing to the failure included:

- A substantial amount of test time was run at 240% load and not at 200% load;
- The ability to control the oil temperature to that particular bearing was little help in forestalling the failure; and
- The helix angle of the Convoloid intermediate gear set was slightly greater (14.7° versus 10°) than that of the involute gear, therefore thrust forces were greater and, as a result, the Convoloid bearing failed before the involute bearing had the chance to fail.

4.6.3.4. Conclusion

Although both gearboxes performed very well under the high accelerated test loads required for an accelerated test, much was learned with both designs. To increase the capacity (and design) of bearings in both gearboxes would require extensive redesign, housing machining, gear-design changes, and other component redesigns and manufacture. A full treatment of the rationale for deciding to proceed with the 3-Hole Test is provided in Section 5.

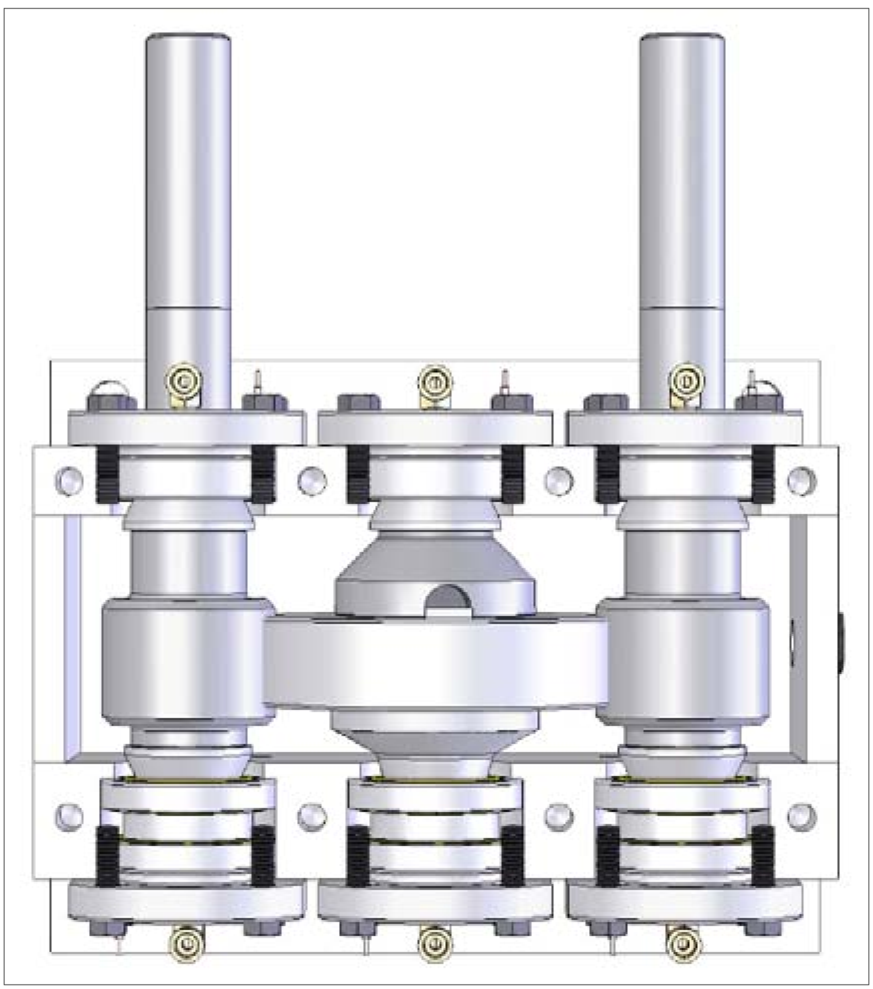

Figure 4.28. Preliminary layout of 3-Hole Test gearbox

5. Change of Test Program Venue

5.1. Meeting of NREL and Genesis Partners' Personnel

The failure of a spherical bearing in the Convoloid intermediate shaft, described in Section 4, indicates some of the cost and time implications of fixing that support system and redesigning all bearings in both systems. Such a project would be necessary to continue testing at the levels used prior to the bearing failure. On May 18, 2006, a meeting was held in Golden, Colorado, to assess the implications of continued testing of the Micon 108 gearboxes versus redesigning a smaller 4-square rig (called a "3-Hole Test") and continuing the Convoloid gearing assessment project using this new approach.

5.2. Analysis of Change of Test Program Venue

The following criteria can be used to justify continuing the Micon 108 testing.

- Substantial data on the performance of both gearboxes has been gathered.
- The test rig itself and electronic data–collection system are working well.
- Continued testing, with additional refinements in gear geometries and component upgrades—especially bearings—will provide significantly more data in "real-world" wind turbine gearboxes.
- The Micon 108 design demonstrates that there is freedom in gear interfacing components—especially bearings—where hefty safety factors can be applied to enhance the long-term stability and reliability of the test rig.

6. Project Initiation—3-Hole Test

As with the Micon 108 test, the focus of the 3-Hole Test is to refine the Convoloid tooth form from theory to practical application and then qualify the theoretical calculations by developing well-controlled test results. During testing, involute geometry is used as a control; this also acts to qualify existing gear-rating theory, specifically ANSI/AGMA2001 and ISO6336.

6.1. Description of Test

The basic test concept is identical for the Micon 108 test. A four-square (back-to-back) arrangement was developed using a hydraulic torque actuator for load application and direct lubrication to the gears and bearing with a dry-sump in the gearboxes. A simple gearbox design facilitated easy part changeover.

6.1.1. 3-Hole Gearbox Design

The concept for the 3-hole gearbox was to build a much simpler,-less expensive test gearbox that would enable both the Convoloid and involute gear sets to be optimized for the applied test loads. As shown in Figure 6.1, the original concept includes a pinion driving a gear that in turn drives another pinion. The pinions have the same number of teeth, therefore the ultimate ratio is 1:1. Thus, exact tooth counts are not necessary and gearing can be optimized without limitations.

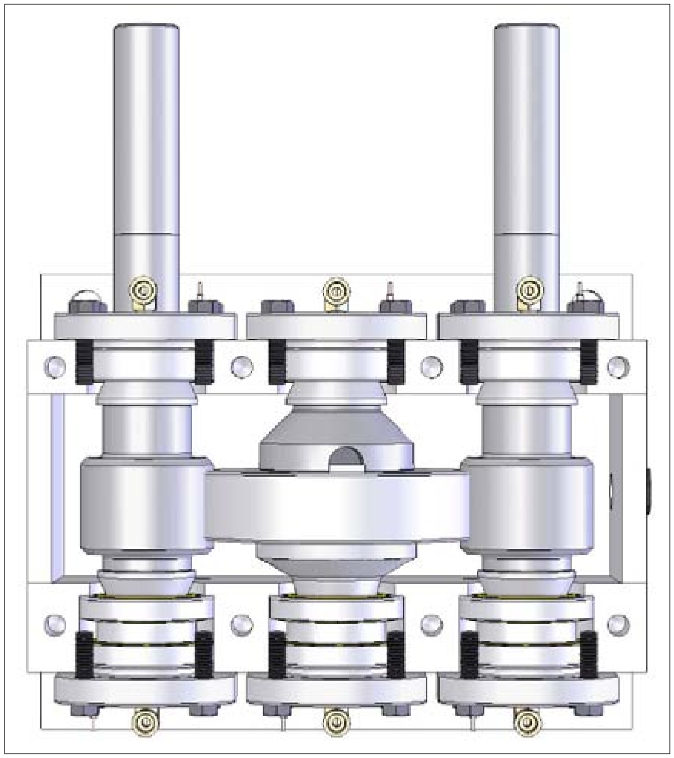

Figure 6.1. Preliminary layout of 3-Hole Test gearbox

An additional advantage to this type of testing is that, in each gearbox, the two gear sets act in different manners. If the torque is applied through the left-side pinion (*see* Figure 6.1), for example, then this pinion drives the intermediate gear. As the load passes through the gearbox to the right side, the gear drives the right-side pinion. In the gearing industry, much discussion has centered on the difference in effects of a "reducer" gear set (pinion drives a gear) versus an "increaser" gear set (gear drives a pinion). Specifically, the sliding action between the gear teeth determines how well oil can lubricate between the contacting flanks. Some gear-design professionals adamantly recommend different gear geometries for reduction gearing versus increaser gearing for this purpose; others in the industry are not convinced of the necessity. This configuration tests these theories for the involute gears.

6.1.1.1. Conceptual Development

To keep the size of the test manageable, a gearing center distance of 4.00 in was chosen. As the center distance increases, the amount of applied load increases (because load capacity is proportional to the cube of the center distance). All gearing are expected to have the same materials and heat treatments as production wind turbine (WT) gearboxes (high-quality 8620 steel to AGMA 2001 Class 2 specifications).

Initial difficulties arose when ratings of both gear sets were much lower for the common intermediate gear due to reversed bending. Future testing will investigate this loading, however the initial testing was intended to qualify the surface durability of the tooth forms and to minimize micropitting. In concept, the intermediate gear could be modified to cause the pinions to contact in different areas of a much wider face width. To simplify lead grinding for crown, however, the intermediate gear was changed to two separate pieces (Figure 6.2).

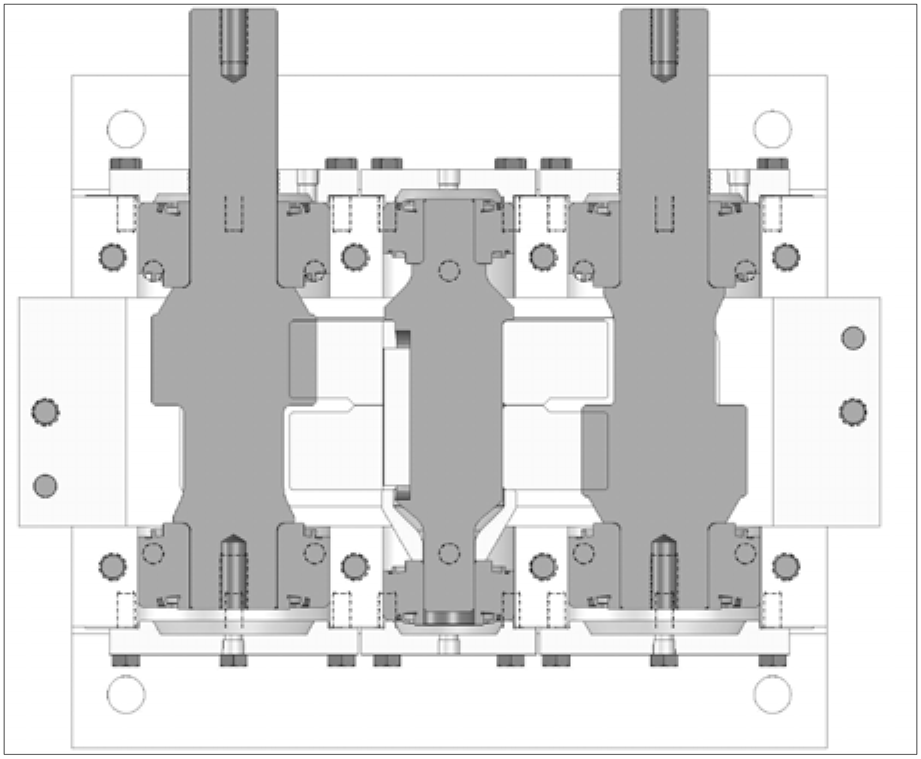

Figure 6.2. Section view of revised gearbox design

After experience with improper bearing (and, thus, gear) support in the Micon 108 test, a robust housing design using an industry-proven design incorporating the readily available tapered roller bearing was chosen. A stiff two-piece design with a split at the bearing bore center lines allowed for easy part changes and reassembly. Within the given space of the 4.0-in center distance, as much bearing as possible was chosen to be assured of no bearing failures. Lubricating oil is supplied to each bearing and to both sides of the gear meshes. The bottom of the gear housing is open, allowing oil to drop into a reservoir. Each gearbox has separate lubrication systems.

Analysis of related components and an unexpected failure in the intermediate shaft required changes in the intermediate bearing selection and a strengthening of the shaft. The final design is shown in Figure 6.3.

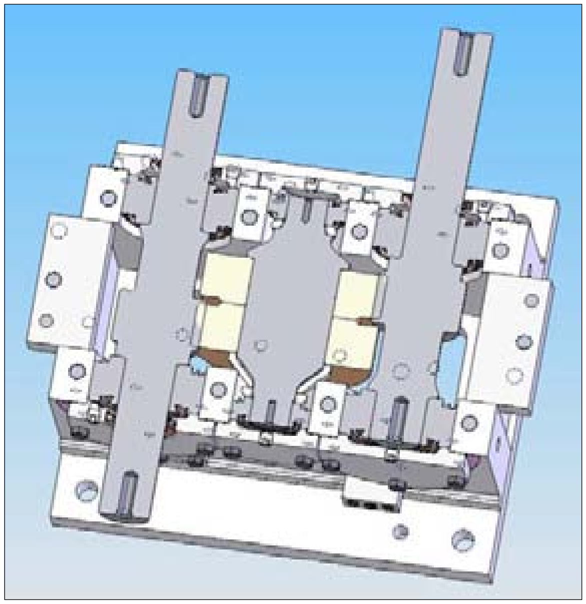

Figure 6.3. Final design for the 3-Hole Test gearboxes (section view)

6.1.1.2. Gearing Design

Gearing was designed to achieve expected L_1 lives of approximately 200 hours with the original test load (6,300 lb-in). When calculating the average lives at this load (L_{50} > 24,000 hours), it was realized that too much time would possibly pass between subsequent tests. An increase in load to 7,000 lb-in reduced the average life to approximately 6450 hours (L_1 = 11 hours). Additional increases were planned, but early gear failures (surface contact, macropitting) suggested maintaining the current loading. The involute gear set is summarized below.

Table 6.1. Involute Gear Set

	Pinion	Gear
Number of Teeth	**16**	**35**
Normal Diametral Pitch (Module)	6.000 (3.629)	
Normal Pressure Angle	20.0°	
Helix Angle	18.6974°	
Center Distance	4.00 in	

A full analysis using the AGMA Gear Rating Suite of software for the involute gearing under the 7,000 lb-in test load is included in Appendix A. The calculated lives for the Convoloid gears were much longer than with the involute, with an average life of more than 185,000 hours and an L_1 life of 320 hours. The Convoloid gear set is as follows.

Table 6.2. Convoloid Gear Set

	Pinion	Gear
Number of Teeth	11	24
Normal Diametral Pitch (Module)	4.8917 (6.1925)	
Equivalent Normal Pressure Angle	23.72°	
Helix Angle	26.5732°	
Center Distance	4.00 in	

All gearing was analyzed for deflections to determine the tooth modifications necessary to achieve a smooth, balanced load across the active tooth flank. This analysis required creating an analytical model of the gearing, supporting shafts, bearing stiffness, and housing stiffness. Some modifications to the HS pinions and the intermediate shaft were required due to excessive shaft deflections under load.

6.1.1.3. Housing Design

The housing was designed with two halves to make it easier to change out gear components. The base of the lower housing half has a large opening so the oil drops freely into the reservoir. The two halves are pinned together during machining operations, maintaining bore integrity and alignment during changeovers.

6.1.1.4. Component Analysis

- Due to the high loads being circulated through the gearboxes, analysis of the individual components is necessary, specifically keys, bearings, and shaft stresses.
- The original conceptual design featured a single gear that passed the load from one pinion to the other. A key was not necessary because no load was passed to the shaft. Upon changing to two gears, the key had to carry the entire load. Analysis of the key prompted a redesign of the intermediate shaft that increased the gear mounting diameter. The final design yielded an appropriate key safety factor for all test loads.
- Bearings were replaced whenever a component required replacement. Most components were loaded from both directions (first one flank and then the other), however, so the bearings had to be able to withstand twice the number of expected gearing cycles.
- The original 3-hole design incorporated bearings that were as large as possible to assure long life. For the larger intermediate bearing, however, the shaft size was diminished and weakened. Shaft stress calculations at the bearing shoulder revealed enough capacity for early test loads (6,300 lb-in), but when using the higher loads necessary to reduce test time (7,000 lb-in), one of the shafts broke. The calculated safety factor at the higher load was less than 1.0.
- A change in bearing (slightly smaller and with lower load capacity) enabled the shaft to be strengthened, increasing the safety factor to 1.59. Although the intermediate bearing had a lower rating than the original, during testing no wear was found on any of the bearings and no additional shaft failures occurred.

6.1.2. Test Equipment and Fixture Design

6.1.2.1. Test Fixture

The purpose of the test fixture is to properly support the gearboxes, supply load to the gearing, rotate the gearing, and supply clean lubricant of a specified temperature. During operation, the data gathered to monitor the test should include torque application, temperatures, vibrations, and lubricant cleanliness.

During testing, a large amount of torque was applied to the system. This torque acts to “twist” one test gearbox with respect to the other. It therefore was imperative that the mounting surface of the fixture be stiff enough to maintain gearbox shaft alignment under test loads. Additionally, to create a compact testing area the fixture was designed with internal reservoirs to accommodate all the oil required for the test. Further, separate reservoirs are included to prevent cross-contamination between the two gearboxes. Each reservoir had two baffles to allow particles to settle and to reduce foaming.

The intent of the design was to create a platform for mounting not just the gearboxes but the drive motor and lubricating pumps. This platform, along with the rest of the test equipment (heat exchangers, filters, and computer control system), then fit into a shipping container allowing for unobtrusive placement outside of normal manufacturing space. This was necessitated by requests to free up the main test platform and reduce noise.

Lubrication was straightforward, with oil being directly applied to each bearing through the bearing caps and to each gear mesh via oil jets. A "dry-sump" design enables all lubricating oil to fall out of the gearbox and into the fixture tank. As with the previous test, each gearbox includes a separate lubricant system with separate filters and heat exchangers.

The fabricated fixture shown in Figure 6.3 met all design intents. Each lubrication system (one for each gearbox) uses two oil pumps. One pump directs oil from the reservoir through a 3-micron filter into a distribution manifold that, in turn, feeds oil to each bearing and to both sides of the gear meshes. Oil flow is controlled by small orifices in the manifold, thereby ensuring oil flow to all components. A second pump circulates oil from the reservoir through a heat exchanger, past a heater, and back into the reservoir, thus maintaining a consistent oil temperature.

After initial alignments, the test gearboxes were pinned to the top of the fixture. It was found that part changeovers were accomplished more easily if a gearbox was removed from the test fixture. Tapered pins provided assurance that the shafts would be aligned when the gearbox was remounted to the fixture.

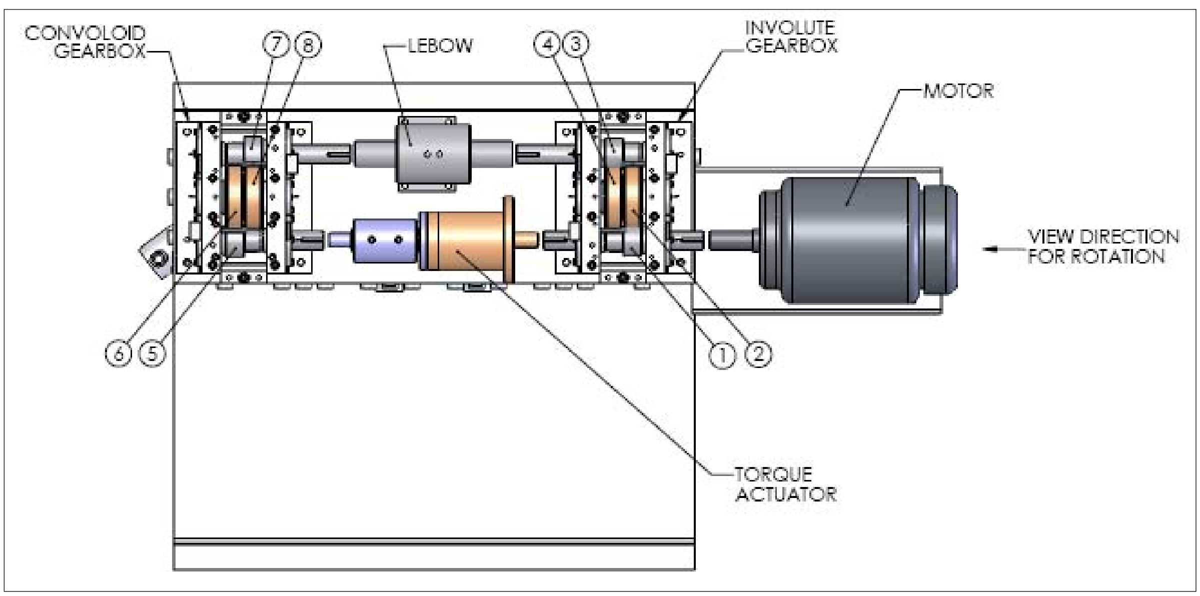

Figure 6.4. Top view of the 3-Hole Test fixture configuration

6.1.2.2. Load Application

Similar to the Micon 108 test, rotation was provided by a directly mounted motor. To mount the gearboxes to the solid top of the test fixture, a different method was needed to apply torque. By using a hydraulic rotary torque actuator with a rotating union, torque could be applied in either direction without affecting shaft alignment—allowing use of standard couplings. Being able to change the torque direction was necessary for testing both flanks of the gear teeth. Torque was monitored using the same Lebow transducer from the Micon 108 test.

A standard 3-phase motor with a variable frequency drive (VFD) was chosen to drive the system. Using the design power of 180 hp (at 1,800 rpm) and a total 6% loss in the system, 10.8 hp is required. To apply as many load cycles as possible, the test operated at 2,200 rpm—thus requiring about 13.5 hp from the motor. Because of the winding losses due to the increase in speed, and to keep motor temperatures down, a 30 hp motor was used.

Operation of this 3-Hole Test allows for two different mesh conditions (a reduction gear set and an increaser gear set) in each gearbox (involute and Convoloid). Therefore, four gear meshes were tested at a time. Subsequent testing also can be conducted by reversing the direction that torque is applied. Thus, sufficiently descriptive and unique annotations were required for each component as well as each tooth flank. Referencing Figure 6.4, gearing component designations could follow the descriptions in Table 6.3.

Table 6.3. 3-Hole Test Component Naming Specifications

	Involute Gearbox		Convoloid Gearbox
1	Involute pinion 1	5	Convoloid pinion 1
2	Involute gear 1	6	Convoloid gear 1
3	Involute pinion 2	7	Convoloid pinion 2
4	Involute gear 2	8	Convoloid gear 2

As an example, for a given direction of torque, gear sets 5, 6, 3, 4 could act as increasers, and gear sets 1, 2, 7, and 8 could act as reducers. This changes as the torque reverses directions to gear sets 5, 6, 3, and 4 acting as reducers, and gear sets 1, 2, 7, and 8 acting as increasers. Determining whether a gear set is acting as an increaser or a reducer is based upon the motor direction and which gear tooth flanks are loaded.

6.1.2.3. Instrumentation

Test instrumentation is similar to that of the previous test; however, additional controls for torque application were added to facilitate unattended operation. A substantial cost for the previous test was a requirement of having an operator monitor the test at all times. This was necessitated by constant variations in lubricant flow and fluctuations in the torque control required constant attention.

For the 3-Hole Test, alarm points were inserted in the control software to enable the system to carefully shut down when appropriate. Key among these shutdown parameters were high bearing temperature, high vibrations, and detected particles in oil flowing out of the gearboxes. Further, external conditions were monitored (test-fixture compartment ambient temperature, computer compartment ambient temperature, electrical power, smoke detector) so that the system could be automatically shut down in the event of an emergency. A detailed list of monitored parameters is given in Appendix A.

6.1.2.3.1. Monitored Values

Monitoring bearing temperatures and gearbox vibrations so that any adverse deviations could be captured was very important. If a warning threshold was surpassed, then visible and audible alarms were triggered in addition to the automatic shutdown of the test. As determined from the previous test, temperature could not necessarily be considered a dependable variable. Also, total vibration did not reveal a problem until major damage occurred. One aspect of vibration did directly correlate to the beginning of damage, however (Figure 6.5, see vertical line that corresponds to test documentation made by the operator of a “change in sound” in the test). In one direction the levels were generally small; however, once damage started, this very small component of the total vibration dramatically changed. Analysis of the starting value of the small vibration component was used to establish an alarm value.

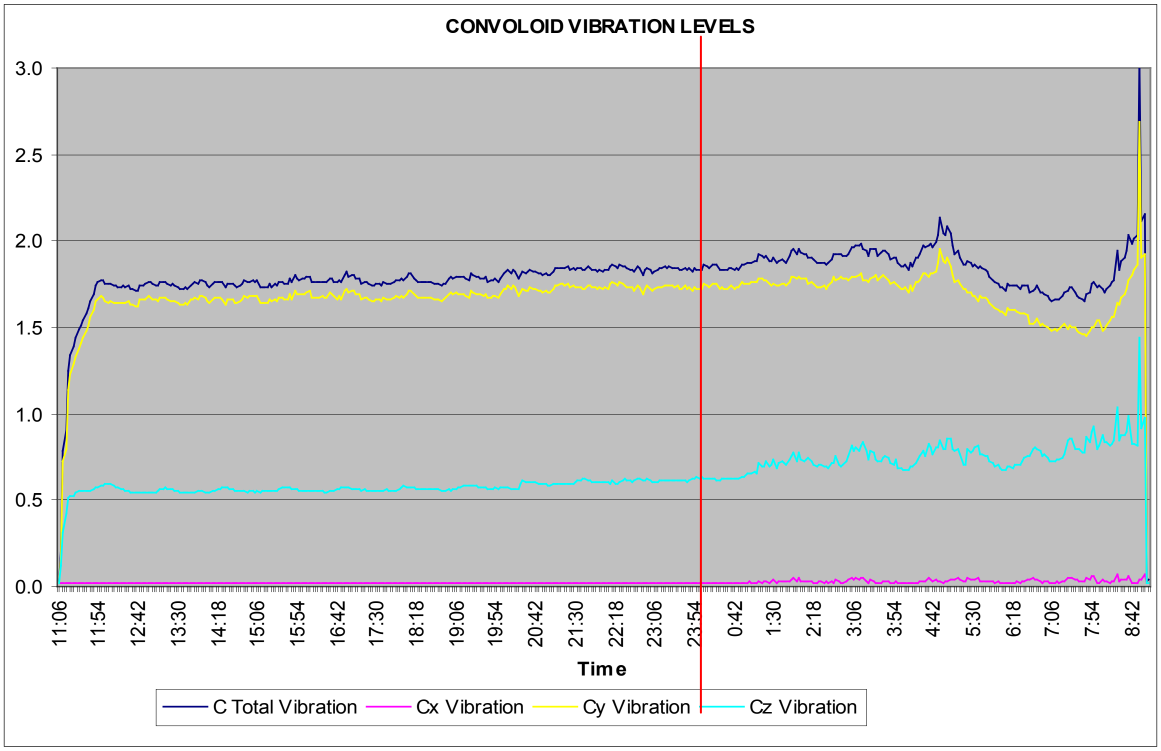

Figure 6.5. Micon 108 vibration levels at time of bearing failure

6.1.2.3.2. Controlled Values

In contrast with the earlier test, motor speed and applied torque was computer controlled and monitored. Any variations in the rheostat setting were adjusted, keeping the system much more consistent than when corrected manually.

6.1.2.4. Alarms

In addition to external visual and audible alarms installed to warn of potential gearbox or test fixture failures, e-mail and telephone notifications were added due to the unattended operation of the test fixture. Included in Appendix A are specific alarm points. As noted, all bearing temperatures were monitored with alarms set for temperatures exceeding 190°F. Alarms also were set for vibration and lubricant supply. Another important alarm condition was related to the presence of any metal particles in the oil falling out of the test gearbox and back into the reservoir. This oil was funneled past a chip detector (Figure 6.6) designed to collect metal particles with a magnet. When particles are present a gap is bridged, triggering the alarm.

Figure 6.6. One of the chip detectors after a surface fatigue failure

6.2. Test Preparations

6.2.1. Test Equipment

6.2.1.1. Gear Manufacturing

Manufacturing of the gearbox housings, the involute gearing, and the Convoloid gear blanks was accomplished with no problems. Twist correction protocols are inherent in most gear tooth grinding machine software, and depend on basic involute data to effect the necessary machine/wheel movements to mitigate twist. Convoloid forms, being non-involute, do not respond as well to these twist correction inputs as do involute tooth forms. Verification of the tooth contact was used to apply slight machine profile corrections resulting in good no-load contact. Upon installation of the gear in the test gearbox, and painting it with machining dye, the light-load contact patterns were confirmed (Figure 6.7, Figure 6.8).

Figure 6.7. Representative involute no-load contact

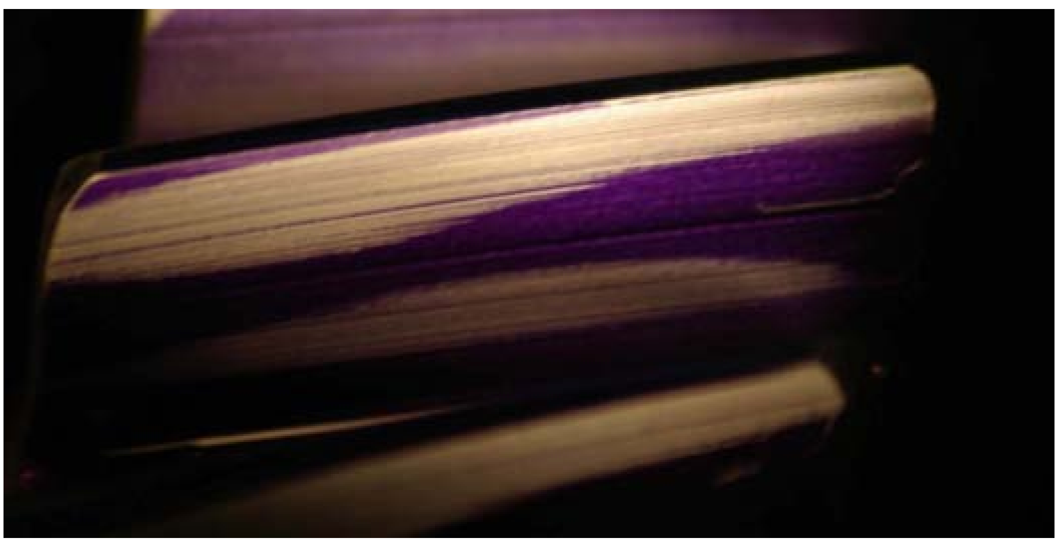

Figure 6.8. Convoloid no-load contact

In the ideal case, a no-load contact shows marking centered on the tooth flank, and the edges of the marking gradually “feather away” leaving no hard edges. For the Convoloid gearing, there should be an empty area in the center of the flank around the transition zone with contact above and below—again with feathered edges. The original Micon-108 test had very hard contact around the transition zone. It was in this area that micropitting appeared very early in the test and at a relatively low load.

The intent of the initial testing is to determine the proper tooth geometry to minimize or eliminate micropitting around the transition zone. The test is designed to test one flank at a time for each gear element, therefore if good contact can be achieved on one flank then the hardened test parts can be ground and the test begun with no detriment to viability of the test. Whether micropitting appears with the updated geometry should be evident early in testing. Should micropitting occur, the gearing geometry is evaluated and the parts are reground. When a test can be run without premature wear (micropitting), then long-term testing can begin.

6.2.1.2. 3-Hole Gearbox Housing

Manufacture of the gearbox housings was straightforward. The design consisted of fabricated steel upper and lower housing halves to provide exceptional bearing support. The housings (Figure 6.9) were stress relieved to prevent any unexpected distortions during testing. The two halves were pinned and bolted prior to final machining to assure bearing alignment. Housing stiffness for this test is much higher than in the previous test. In contrast to the one-piece housings for the Micon-108 test, these split housings allow easy replacement of gearbox components (Figure 6.10).

Figure 6.9. The 3-Hole Test gearbox during final machining

6.2.1.3. Test Fixture

The test fixture (Figure 6.11) consists of the two test gearboxes, the main test fixture (with drive motor, Lebow torque transducer, rotary torque actuator, and lubricating pumps), electrical control assembly, and the computer control.

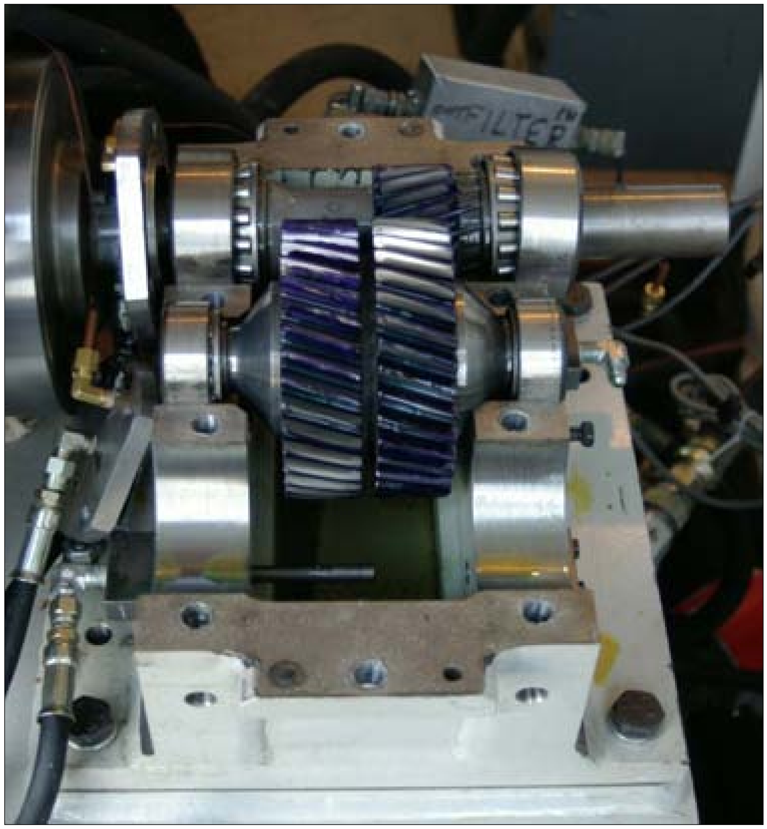

Figure 6.10. The 3-hole split housing enables easy part changes

Figure 6.11. The 3-Hole Test fixture

The test fixture base was designed to hold all the oil for the test in separate baffled reservoirs. It was necessary to have a separate lubrication system for each gearbox to prevent cross-contamination between the gearboxes. Two separate pumps were fitted for each system—one for circulating oil taken from the reservoir through the heat exchanger and heater; and, one for supplying filtered oil to the oil manifold for distribution to each bearing and to both sides of both gear meshes. A detailed discussion of the test fixture configuration and shakedown is included in Appendix A.

6.2.1.4. Testing Shutdown Criteria

The occurrence of a fatigue failure is an obvious reason to halt testing. During testing, however, other conditions can occur that require ceasing operation immediately. Any change in sound, vibration, oil cleanliness, or any other abnormal operation is acceptable cause to shut down the operation of the load cell and motor (lubricant pumps should remain engaged until all motion is stopped). A thorough inspection of all components should be conducted before continuing the test, and all variations should be noted in the test notebook. A list of computer monitored conditions follows; however, there can be additions or deletions as needed. Each of these conditions has warning levels and shutdown levels enabling the test to run unattended.

- Loss of oil pressure (pumps failed)
- Oil differential pressure too high across filters (filters clogged)
- Chip detector goes low (ferrous material in the oil)
- Bearing temps are greater than 190 º F
- Oil flow too low
- Oil pressure too high at manifold (manifold orifices clogged)
- Vibration over limits in any of three planes
- Particle counters indicate high particulate count
- Torque loss
- Power loss to test container
- Bulk oil over temperature
- RPM of drive system out of test limits
- Visual observations from the webcam indicate shutdown is needed

6.2.2. Test Protocols

6.2.6.1. Micropitting Test Protocol

Reference: Micropitting Test Protocol, dated April 9, 2007 (*see* Appendix A).

Initial testing in the 3-hole gearboxes was intended to qualify profile corrections in the Convoloid tooth form. This testing is in preparation for the main extended comparison test. Previous test gears developed micropitting early in the test cycle. Although this failure mode is not in itself catastrophic, it can lead to macropitting and, in extended operation, to complete failure of the gear. Before extended testing can occur the micropitting phenomena must be solved.

6.2.6.2. Extended Comparison Test Protocol
If premature micropitting wear is prevented, then extended testing for comparisons of the surface durability is possible. The expected failure mode is surface fatigue on the contacting gear tooth flanks which results in macropitting. A failure for this test was defined as follows.

> The definition of failure for this test is the development of an area of macropitting over more than 1.5% of the contact face width i.e. active tooth flank. The two gear sets (involute and Convoloid) are of a different module (tooth size), therefore the actual area varies slightly. The amount of area that is assumed to be a macropitting (fatigue) failure for the involute is an area of 0.0070 in^2 (4.52 mm^2) which can be described by an area bounded by a circle of an approximate 0.10 in (2.4 mm) diameter. For the Convoloid gearing, the area is slightly larger {0.0117 in^2 (6.55 mm^2)} with a diameter of 0.12 in (3.1 mm).
>
> Extended Comparison Test Protocol, dated April 9, 2007 (*see* Appendix A).

Torque values for testing originally were expected to be 7,880 lb-in, however this was reduced to 6,300 lb-in. After initial operation at this value, loading was increased to 7,000 lb-in where it remained for the remainder of the test.

7. Testing—3-Hole Test

7.1. Testing
Appendix A provides detailed discussions of the testing. To date, the majority of testing occurred during shakedown of the test fixture. As such, some gear failures could not be attributed to traditional surface fatigue (the expected failure mode). Also, most testing has involved two involute gear sets (versus one involute and one Convoloid) due to manufacturing delays and the micropitting protocol. The number of cycles and load amount are being logged for each loaded flank, so all test cycles can be tracked. All gearing failures have been documented to help generate a baseline for future Convoloid comparisons. The definition of gear surface fatigue failure is discussed in Section 6.2.6.2 (above).

7.1.1. Early Test Observations—Micropitting Test
Before starting any extended testing, the premature micropitting wear that appeared on the Convoloid gear teeth in the Micon 108 test had to be eliminated. In the earlier test, micropitting occurred early at lower test loads, therefore initial testing was focused on proving the tooth profile corrections, and resolved this situation. The Micropitting Test Protocol is included in Appendix A.

Execution of the protocol began during shakedown of the test fixture under constant supervision. When the test fixture was up to temperature, low oil pressure in the distribution manifold that feeds oil to the bearings and gear meshes prompted inspection of the lube pump bypass valve leading to a loss of lubricant to the Convoloid gearbox. Although operation was stopped, severe scuffing damaged the gear teeth (Figure 7.1).

Figure 7.1. Severe scuffing failure due to loss of lubricant

Some teeth suffered worse damage than others due to how the load was applied, the direction of gearing rotation, and how the oil was supplied to the tooth mesh. If the gear mesh was loaded (as in Figure 7.2), then loss of oil pressure would starve the contact area of lubrication. No oil would could flow upward, against gravity. The upper oil nozzle only acts to cool the gear teeth after emerging from the mesh. However, the other gear mesh received a little more oil, due to gravity allowing oil to drip into the mesh.

When oil failed to lubricate the gear teeth, localized temperatures raised dramatically causing minute welding and tearing of the tooth flanks. Fortunately, some information can be gleaned from the undamaged portions of some teeth in the gravity lubricated mesh.

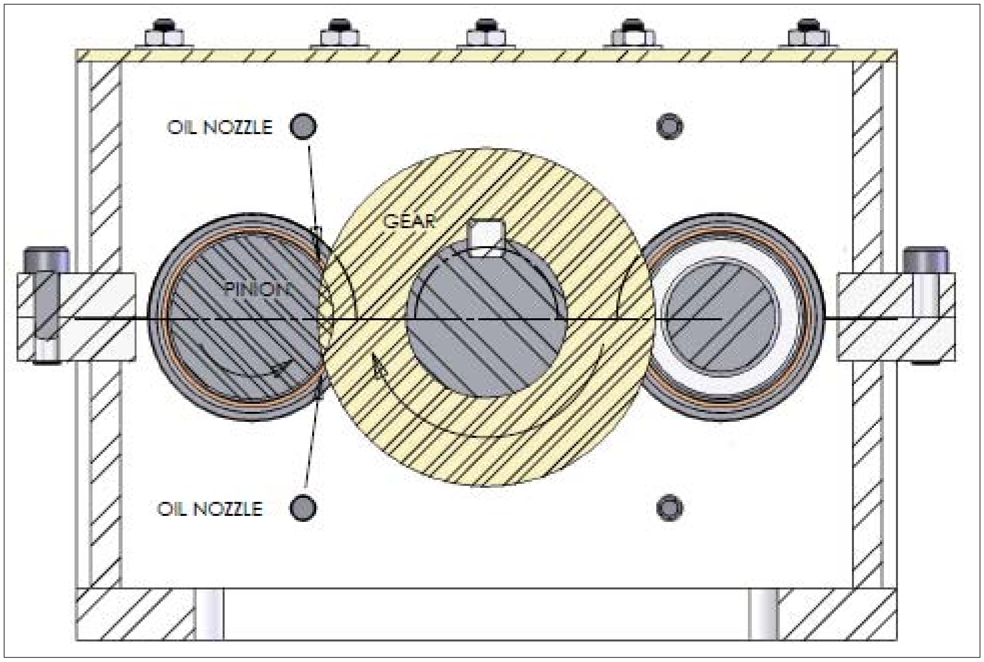

Figure 7.2. Cutaway of the 3-hole gearbox showing the gear mesh lubricating method

Figure 7.3. Close-up of Convoloid gear tooth with no micropitting and partial scuffing damage due to loss of lubrication

The loaded tooth flanks show some scuffing damage (Figure 7.3, Figure 7.4), but other areas still show grinding marks. Due to early micropitting around the transition zone in the Micon 108, slight modifications were made to the tooth profile algorithm in this area. Inspections of the undamaged portion of the gear teeth do not show any early wear from micropitting around the transition zone giving a (preliminary) verification to the changes. Enough torque (6,300 lb-in) had been applied to generate wear if similar geometry conditions to the Micon 108 test had existed. However, not enough test time passed to allow for a definitive assessment. Only one full Convoloid gear set had been finished ground, pending qualification of the final tooth profile geometry. Continued micropitting testing on the Convoloid gearing therefore had to be postponed until new parts were ground.

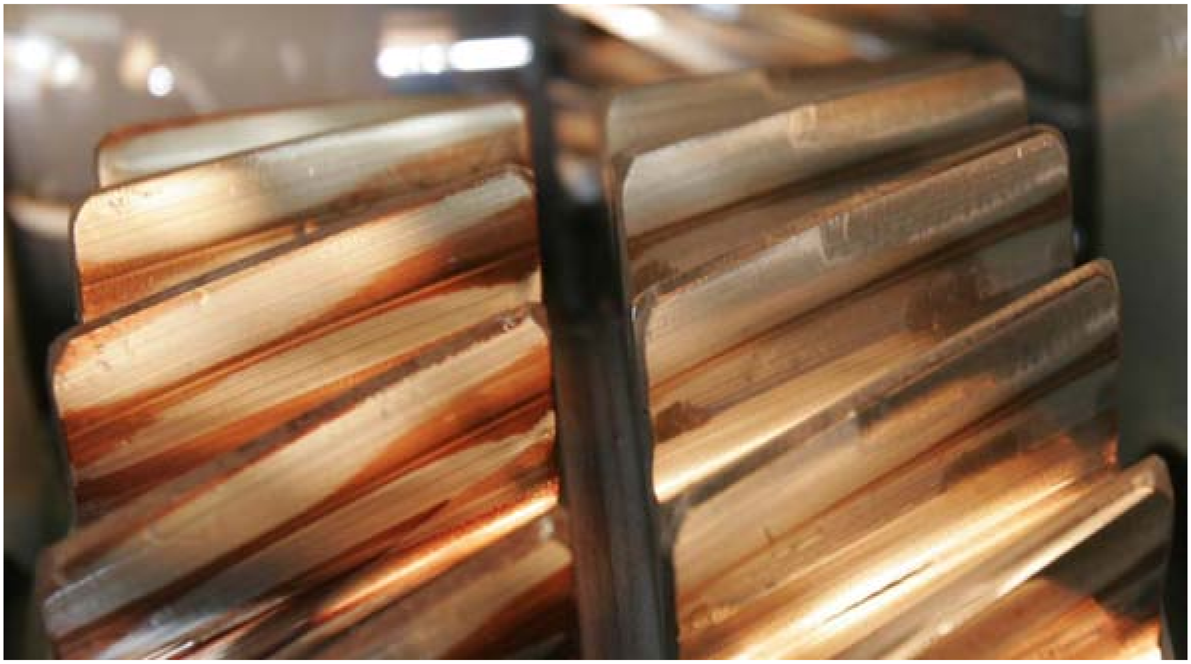

Figure 7.4, Convoloid gearing showing some scuffing and discolored marking dye; the left gear was not loaded

7.1.2. Test Observations—Extended Testing

Operation of the test fixture was required to properly shake down the equipment. Because the calculated average life was greater than 8,000 hours, applying load cycles whenever possible was thought to help achieve usable results as soon as practical. Therefore involute gearing was installed in gearbox 2. All test cycles were documented and it was intended that any information gathered from the involute gearing would act as a baseline for the Convoloid testing. Further comparisons of the involute gearing with gearing-life calculations based on industry standards will be meaningful to the industry.

As with previous testing, the no-load and full-load contact between all gearing were documented to verify proper gear mesh (Figure 7.5, Figure 7.6). The slight presence of marking dye at the corners and in the root and tip indicate that there is no hard contact in the tooth mesh that would lead to premature wear.

Figure 7.5. No-load contact pattern—involute gear; lube nozzle

Figure 7.6. Example of the full load (7,000 lb-in) contact—involute gears

To reduce the average expected test life to less than a year of constant operation (24 hours per day, 7 days per week) test loading was increased from the original 6,300 lb-in to 7,000 lb-in. Additional increases in torque were not applied due to development of micropitting on some of the gear teeth (Figure 7.7). There was a distinct difference in the appearance of micropitting between driver versus driven gearing. The pinions saw more than twice the number of cycles as the gears, therefore the effects were more prominent.

Figure 7.7. Involute gearing showing early micropitting; both parts (circled) drive their respective mates

Only a summary of the surface fatigue failures appears here, a detailed discussion appears in Appendix A. After 86 hours of operation (Test 3) a significant amount of vibration was noted, prompting shutdown of the test fixture. Upon inspection, pinion 1 in gearbox 2 was found to have significant macropitting (see Figure 7.8 through Figure 7.10). The origin of the pitting appears to fall at the pitch line of the tooth flank, which is consistent with the area of highest calculated stresses for involute gearing. This pinion was operated as a reducer—the pinion drives the gear. Note that there is frosting, or micropitting, in the dedendum (near the tooth root) of the tooth flank. In some cases, this phenomenon has been thought to precipitate a macropitting failure, at the pitch line where the macropitting originated no micropitting is present. The macropitting therefore can be considered a classic fatigue failure of the type expected in this test. It is important to note that only 10 hours of operation occurred between inspections. There was no hint of damage prior to the high vibration level (manually monitored at that time) that prompted cessation of the test.

Figure 7.8. Massive macropitting on pinion tooth flank (Test 3, gearbox 2, pinion 1)

Figure 7.9. Additional pitting

Figure 7.10. Further pitting and opposite flank scuffing damage from Test 2

The definition of macropitting failure for this test was set as 1.5% of the operational tooth. In this failure the amount of pitting greatly exceeded the definition. It was expected that macropitting would progress from small pits to larger ones and that, with the increase in pitted area, the measured vibrations would increase. At the time of this failure, however, the test parameters and warnings had not yet been set up.

Test 4 was terminated after 18 hours due to another surface fatigue failure. Figure 7.11 and Figure 7.12 show the failure—again, a pinion operated as a speed reducer, this time in gearbox 1. Here the surface damage was found during a routine inspection. The amount of damage is indicative of the definition of failure. The total test time for this pinion was 104 hours, or 13.7 million cycles.

It is important to note the extensive micropitting on the tooth flanks. For these parts, it occurs almost exclusively in the dedendum of the gear tooth (below the pitch diameter). There has been much discussion in the gear industry concerning whether the type of action (speed increaser versus speed reducer) affects the appearance of micropitting, or whether it is entirely a result of load, lubrication, and surface finish.

During testing, it appears that there is a difference due to the type of action. Both of the failed pinions were loaded as speed reducers (for example, the pinion drives the gear) and there was extensive micropitting in the dedendum of the tooth flanks. The other pinion, which was located in the same gearbox, operated for the same number of load cycles as the failed pinion in Test 4, and was loaded as a speed increaser (Figure 7.13), shows much less wear on the active flank.

Within the gear industry, there is much discussion on the affects of gear geometry, how it is loaded (increaser or reducer), and wear potential. A preliminary analysis of the probability for wear on the involute gear set (first with the pinion driving the gear; and second with the gear driving the pinion) revealed identical specific film thicknesses and similar low probabilities for wear. Outside of the edges (tip/root) of tooth contact, the calculated film thicknesses are the same for both analyses. The film thickness is a function of the relative curvatures between contacting surfaces, therefore the direction of rotation does not appear to have an effect. Additional study into this area is required before definitive conclusions can be drawn.

Figure 7.11. Failed involute pinion from Test 4

Figure 7.12. Pinion loaded as a speed reducer

Figure 7.13. The other pinion in the same gearbox loaded as a speed increaser and showing minimal wear

It is thought that micropitting might precipitate macropitting, therefore the presence of this wear is important. Contact loads on the surface of a gear tooth flank are directly related to the relative curvature between that flank and the mating flank. Standard involute gears have a convex shape, therefore the contact patch between the parts is relatively narrow. Further, the convex-convex contact of the gear teeth acts to “pump out” lubricating oil, reducing the effective film thickness. In contrast, mating Convoloid gear teeth have a convex-concave contact that greatly reduces the relative curvature, creating a wider contact patch under load which maintains a greater amount of oil between the surfaces. This increase in oil-film thickness is expected to reduce or eliminate micropitting in Convoloid gearing. Further, the Convoloid tooth form has no contact in the high-stress area of the pitch diameter. The design method allows Convoloid teeth to have a consistent contact stress across the entire working flank, which is in contrast to the varying stresses found in involute gear teeth.

Another aspect that has generated interest is finding out if the life of a gear is affected by whether the gear component is the driving element (pinion for a speed reducer and gear for a speed increaser) or the driven element (gear for a speed reducer and pinion for a speed increaser). To this point, all failures have been of the driver element (pinions loaded as an increaser), and early small macropitting is visible on a driving gear loaded as an increaser.

A third macropitting failure occurred after a short period of testing. Once again, the failed member was a driving pinion—this time with 82 hours of operation. To date, the only macropitting failures have been found in driving pinions having 86, 104, and 82 hours of operation. The other involute pinions (being driven by the gear) have survived for more than 168 hours and had minimal visible wear. A small macropit is visible on a driver gear; however, this failure has not reached the test definition of failure. All remaining gear flanks show minimal wear.

7.2. Variations During Test

A few situations in which variability might have been introduced to the test must be noted.

7.2.1. Incorrect Torque Application

Early application of load to the test fixture (necessary to confirm operation of the data gathering) caused a greater test load than intended (7,880 lb-in). No ill effects were noted, however, and testing continued.

7.2.2. Loss of Lubrication

In two instances, lubrication of the test gearboxes was lost during operation. One loss affected only gearbox 2, damaging the Convoloid gearing. A second instance was caused by a loss of power to the entire test fixture, damaging all gearing. Although specific data from these gear sets is not available for life comparisons, the operational time (~160 hours) proved to be more significant after experiencing surface fatigue failures in parts having shorter life spans.

7.3. Test Results

A Convoloid test set was destroyed due to a test rig malfunction, therefore time remaining on the contract permitted testing of involute gears only. Although the accumulated life of each involute gear and pinion was logged, only three involute gears could be defined as "test failures."(See Section 6.2.6.2 for a definition.) As noted at the end of Section 7.1.2 (above), the failures occurred in a reasonably tight time span (82 hr to 104 hr), a fact that lends credibility to the consistent accuracy of the gearing, the capabilities of the test rig itself, and many other positive factors. The added fact that all three failures were "driving pinions" is not only another "positive factor," but future testing using the three-hole configuration might shed further light on the relative failure rates of driving pinions versus "driven pinions." Different theories abound regarding the behavior of tooth surfaces with respect to the direction of subsurface shear direction and failure modes and times as a result.

This overall performance provides some assurance that continued testing would produce solid comparisons on the relative performance of involute gearing versus Convoloid (see Appendix A for details). Additionally, these three failures have provided detail on the life cycle–stress relationship through a spreadsheet developed by Genesis Partners. The life factors developed will help guide future testing protocols for the "three-hole continuation," as well as that for general gear testing (see Appendix A for details).

7.4. Continued Future Testing

Appendix A provides detail on the expected times and procedures to complete the test program so that involute and Convoloid performance can be compared with a reasonable degree of statistical surety.

8. Detailed Design of Larger Gearboxes Using Convoloid Gear Sets

8.1. Selection Criteria

The method of selection for the most advantageous existing wind turbine gearbox to study and then to retrofit with gearing designed with the Convoloid tooth form was based on the following criteria.

- Gear system architecture to be common to most gearboxes 1.0 MW and greater.
- The existing substantive data on the market-sell price for the particular model selected.
- The substantial detail on the existing gear designs enabling the proper assessment of existing gearing stress levels.
- General design detail is available—such as housing sizes, enclosing configuration, existing bearings, carrier configuration, and shaft configurations, so that solid modeling can be well established; and
- Genesis Partners has substantive detail on existing bearing types and sizes of the selected model to help assess cost and bearing load/life calculations.

8.2. Extrapolation

The basic steps in application of the data developed in the Micon 108 test (not enough data has been collected from the 3-Hole Test as of the date if this writing) to allow reliable extrapolation to wind turbine gearboxes of larger sizes are as follows.

- As a result of early Convoloid test programs, the stress concentration factor protocols for Convoloid gearing were changed and then used in the Micon 108 test program with great success.
- The calculation of surface durability stress levels in Convoloid gearing appear to be very reasonable. No macropitting was observed in executing the Micon 108 protocol.
- Tip relief calculations as a result of the Micon 108 tests have been refined to compensate for the results of the test.
- Lead modification protocols using an involute-based LVR program worked moderately well for the 108 tests, with major improvements projected for the future.

The gearbox that fit all of the above criteria was a 750-kW gearbox commonly used in the wind turbine industry today; accordingly it was selected for the study.

8.3. Comparison of Original 750-kW Gearbox with Convoloid Enhanced Gearbox

The first step in the redesign process was to study in depth the geometry of the gearing existing in the 750-kW gearboxes. Detailed gear data was available including module, number of teeth, helix angles, and face widths. A solid model of the gearbox was developed (Figure 8.1) that corresponded with the weight of the physical part.

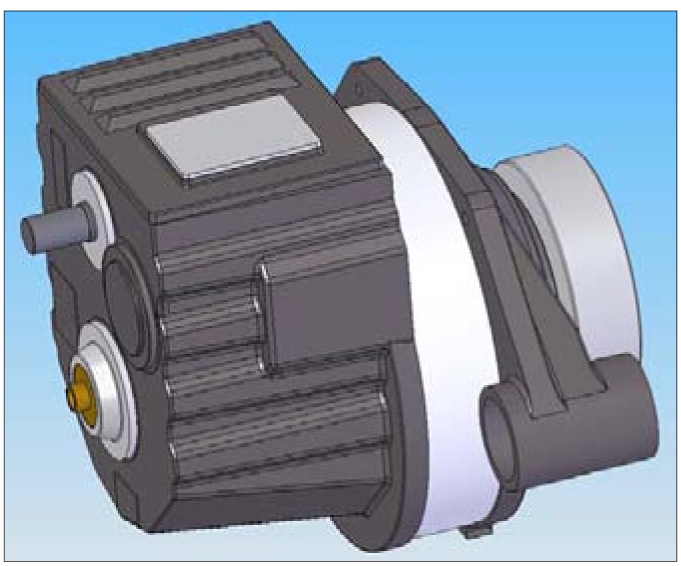

Figure 8.1. Representative 750-kW gearbox

It was assumed that all materials conformed to AGMA 2001 Grade 2 carburizing steel, and that gear accuracies were very high. These two parameters were held constant throughout the analysis.

The following data was reverse-engineered using AGMA/ISO procedures for all gears.

- Bending stresses
- Surface durability stresses
- Radial, tangential, and thrust loads

Input torque and speed to the low speed planetary stage were calculated at full load (750 kW) and standard input speed for a wind turbine gear drive of this power rating (28.6 rpm). From this information bearing loads could be calculated followed by the calculation of bearing L_{10} lives.

It was an additional constraint on the redesign that the planetary ratio and (2) successive helical pair speed increasing stage ratios were distributed in the Convoloid gearbox about equal to those of the involute gearbox. The Convoloid gears then were designed using the same input torques and speeds, but reducing the general size of the Convoloid gears until surface durability stresses were at or below those of the involute gearbox at each gear set. Bending stresses were not the limiting parameter in the design. Load-sharing protocols for the planetary stage followed AGMA practice and were applied consistently for both the Convoloid redesign and the involute gearbox.

The next step was to select and rate the bearing complement of the Convoloid gears. Although the two-speed increasing helical gear pairs were straightforward in the design and rating of their supporting bearing complements, the planetary carrier and carrier bearings were not. In this case, the resulting compact sun, planet, ring configuration of the redesigned Convoloid gears drove the bearing forces considerably higher than could be supported adequately using the involute configuration. The carrier and planet bearing complement had to be redesigned to achieve the desired bearing life. See Figure 8.2 for the solid model carrier rendition and Table 8.1 for a summary of the important parameter that resulted.

Table 8.1. Comparison Between an Existing 750-kW Planetary Stage and the Convoloid Design

Parameter	Existing 750-kW Gearbox	Convoloid Enhanced Gearbox
Planet Speed	34.12 rpm	34.12 rpm
Planet Torque	34.9 kN-M	34.9 kN-m
Bearing Type	Spherical Roller (2)	Cylindrical Roller (4)
Bearing Configuration	Under Planet	Straddle Mount
Bearing Loads	367.2 kN	360.9 kN
Bearing L_{10} life	129,900 hr	124,900 hr
Planet Carrier Weight	775 kg	646 kg (-16.7%)
Planet/Brg Assembly	225 kg	218 kg (-3.1%)
Ring	503 kg	456 kg (-9.3%)

Next, the entire assembly was modeled and enclosed with a housing very similar to the reference 750-kW gearbox, but smaller.

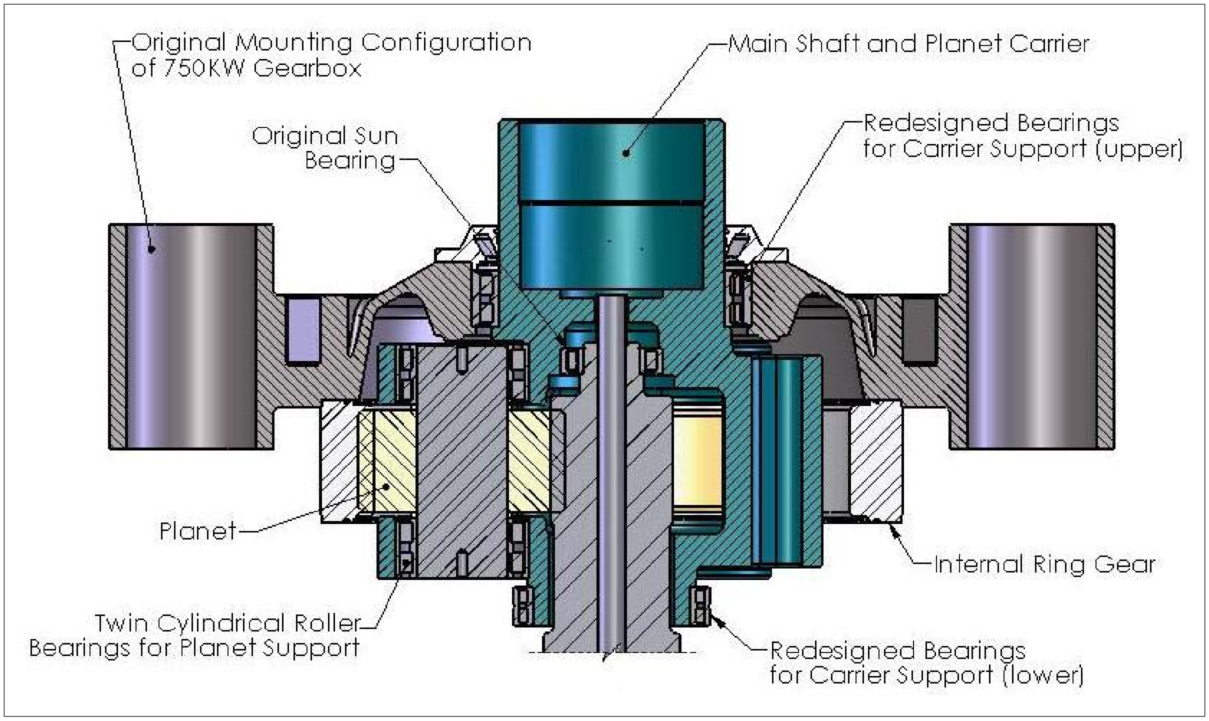

Figure 8.2. Redesigned low speed (planetary) using Convoloid tooth profiles (750 kW gearbox)

8.4. Analytical Model Cost Savings and Extrapolated Results

The fully assembled 3-D Convoloid model was "weighed" electronically and compared to the original 750-kW weight calculation. The resulting weight reduction was 34.9%. Knowing that additional factors must be applied to realistically estimate the actual "in practice" weight savings, a generous safety factor was applied bringing the projected weight savings for calculation of COE values to 21.0%. Weight of a sophisticated mechanical assembly (such as a gearbox) drives the price, therefore it is reasonable to assume that a price reduction of like power capacity, like configuration, and like manufacturing accuracies and materials would be 21%.

9. Cost of Energy Analysis—Summary

9.1. Calculation of Achievable Reductions in Size and Cost of Convoloid Versus Involute Wind Turbine Generator

Section 8.3 described how achievable reductions in size and cost of an upsized Convoloid gearbox could be accomplished. Following the 21% "achievable" decrease in weight of the Convoloid 750-kW gearbox compared to its established in-service involute counterpart, Genesis Partners developed extensive cost models from NREL publications and Genesis' own market research, including actual quotation figures for reasonable production quantities of this specific gearbox. With this information in hand, Genesis Partners made what it thinks are reasonable upscaling assumptions.

- The weight of a gearbox per megawatt is relatively constant up to the 3-MW to 5-MW range.
- Weight is directly proportional to cost.

Further, a conservative cost factor was applied to the already factored weight savings (providing the "achievable" 21% weight reduction) to allow for pricing contingencies. The resulting cost savings with Convoloid designs was calculated out at $35,420/MW (2008 dollars) of capacity, compared to its involute equivalent as a baseline.

9.2. Calculations of Advantages of Retrofitting Convoloid Gearing into Existing Involute Wind Turbine Generators

Section 3.3.3 projects O&M costs as a function of stress level. For there to be economic advantages of retrofitting existing involute gearboxes with Convoloid gearing, it follows that the slightly increased cost of Convoloid gearing—and most probably its bearing complement (due to the effort to balance the lives of the gears and the bearings)—must be less than the resulting O&M projected costs over a reasonably long lifetime of the wind turbine.

- Although of the two stress summaries of the Micon 108 high speed and low speed pass gears show torque increases of 46% and 33% for Convoloid designs (*see* Table 3.1, Table 3.2), to be conservative only a 25% increase in torque is considered for this example.
- For the existing power, the surface durability stresses are reduced by 11.8% with the bending stress reduction being proportional to the torque reduction at 25%.
- This increase in bending strength does not limit the system, therefore the surface durability stresses drive the reliability and, thus, the O&M costs. This conclusion is dependent on the bearing ratings and other component designs keeping pace with the gear-life projections.
- A reduction in stress is directly related to an exponential increase in the anticipated lives of the gearing. This increase in life directly affects O&M costs due to not needing to replace the gearbox as often.
- According to Figure 2-17, page 57 of DOE Publication, "20% Wind Energy by 2030," frequency of repairs to wind turbines of 2.5 kW to 1.5 MW totaled 2% for "drive trains" and 5% for "gearbox."

9.3. Cost of Energy Trade-Offs

In the trade-off calculations that follow, reference was made to the following documents.

- Development of an Operations and Maintenance Cost Model. NREL Subcontract No. YAM-4-33200-07
- "Wind Energy Handbook," Burton & Sharpe (2001)
- "20% Wind Energy by 2030," published by the U.S. Department of Energy Office of Scientific and Technical Information
- "Wind Turbine Design Cost and Scaling Model" NREL/TP-500-40566, December 2006
- "Wind Pact Turbine Rotor Design Study," NREL/SR-500-32495, June 2002
- COE Projection Sheet ($ 2002)—NREL

Three specific trade-offs were examined and analyzed for their respective COE implications. Each was put through the accepted wind energy calculations using the NREL LWST Baseline (2002) Turbine

Calculations—Class 4. Each then was put through the standard calculation for COE using the NREL calculation sheet for the Baseline 1.5-MW Turbine. The results then were summarized and conclusions of the study were articulated. The Baseline COE Projection against which each trade-off was compared is shown in accompanying figures. For a complete analysis of COE calculations for various sizes of wind turbines, see Appendix B.

9.3.1. Wind Energy Generation—Trade-Off 1: Gearbox Cost Reduction / Tower Height Increase (Near Net Cost Neutral)

Trade-off 1 was a study of trading off the lower weight of the Convoloid gearbox compared to its involute counterpart against a 10-m increase of hub height from 65 m to 75 m with an assumed net cost neutral result (*see* Figure 9.3, Figure 9.4). Using "Wind Turbine Design Cost and Scaling Model" and COE Projection Sheet ($ 2002) for reference, the Baseline tower cost is listed as $101,000. An authoritative source (the president of a major wind turbine manufacturer), indicated that the cost of the tower is proportional to the fourth power of any height increase. Thus, the estimated increase in cost to raise the Baseline tower from 65 m to 75 m is as follows.

$$\$101,000 \times \left(\frac{75}{65}\right)^4 = \$133,099 \text{ (2002 dollars)}$$

The increase from $101,000 to $133,099 ($32,099) compares favorably with the decrease in estimated cost of the gearbox at $31,000 (Figure 9.3), and justifies an approximate trade-off because both figures are provided in 2002 dollars.

LWST Baseline (2002) Turbine Calculations - Class 4

10 m windspeed	5.8	m/s		
Weibull K parameter	2.00			
Rated power	1500	kW		
Rotor Dia.	70	meters		
Hub height	65	meters		
Altitude	0	meters		
Air Density	1.225	kg/m^3		
Rotor Cp	0.47			
Cut-in windspeed	3	m/s		
Cut-out windspeed	26	m/s		
Power law shear exponent	0.143			
Hub height windspeed	7.58			
Rated windspeed	11.35	m/s		
Conversion Efficiency	0.95		0.015	
	0.005		0.005	
Soiling Losses	3.5%			
Array Losses	5.0%			
Availability	98.0%			
	Turbine		Weibul Cp	Weibul betz
Energy capture (MWh/year)	4439.53		7668.58	10177.70
Capacity Factor	33.79%			
Energy capture ratio	57.89%			

Figure 9.1. Baseline wind energy with 65-m high tower equals 4,439 MWh per year; wind energy generation trade-off 1, tower height increase (COE benefit of 3.5%)

LWST Baseline (2002) Turbine Calculations - Class 4			
10 m windspeed	5.8	m/s	
Weibull K parameter	2.00		
Rated power	1500	kW	
Rotor Dia.	70	meters	
Hub height	75	meters	
Altitude	0	meters	
Air Density	1.225	kg/m^3	
Rotor Cp	0.47		
Cut-in windspeed	3	m/s	
Cut-out windspeed	26	m/s	
Power law shear exponent	0.143		
Hub height windspeed	7.74		
Rated windspeed	11.35	m/s	
Conversion Efficiency	0.95	0.015	
	0.005	0.005	
Soiling Losses	3.5%		
Array Losses	5.0%		
Availability	98.0%		
	Turbine	Weibul Cp	Weibul betz
Energy capture (MWh/year)	4598.50	8154.10	10822.09
Capacity Factor	35.00%		
Energy capture ratio	56.39%		

Figure 9.2. Convoloid General Electric wind energy with 75-m high tower equals 4,598 MWh per year; wind energy generation trade-off 1, tower height increase (COE benefit of 3.5%)

COE PROJECTION SHEET

Baseline Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - 65 Meter Tower
Improved Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - Convoloid GB/75 Meter Tower
Gearbox Cost Savings Invested in Tower Height Increase.

Rating (kWs)	1500	1500		
Component	Baseline Component Costs $1000	Projected Component Costs $1000	Component Percent Improvement	Major Cost Element % Improvement
Rotor	**248**	**248**	0.0%	
Blades	148	148	0.0%	
Hub	64	64	0.0%	
Pitch mchnsm & bearings	36	36	0.0%	
Drive train,nacelle	**563**	**532**	-5.5%	
Low speed shaft	20	20	0.0%	
Bearings	12	12	0.0%	
Gearbox	151	120	-20.5%	
Mech brake, HS cpling etc	3	3	0.0%	
Generator	98	98	0.0%	
Variable spd electronics	101	101	0.0%	
Yaw drive & bearing	12	12	0.0%	
Main frame	64	64	0.0%	
Electrical connections	60	60	0.0%	
Hydraulic system	7	7	0.0%	
Nacelle cover	36	36	0.0%	
Control, safety system	**10**	**10**	0.0%	
Tower	**101**	**132**	30.7%	
TURBINE CAPITAL COST (TCC)	921	921	0.0%	0.0%

Foundations	49	49	0.0%	
Transportation	51	51	0.0%	
Roads, civil works	79	79	0.0%	
Assembly & installation	51	51	0.0%	
Elect interfc/connect	127	127	0.0%	
Permits, engineering	33	33	0.0%	
BALANCE OF STATION COST (BOS)	388	388	0.0%	0.0%
Project Uncertainty	162	162	0.0%	
Initial capital cost (ICC)	1,472	1,472	0.0%	
Installed Cost per kW for 1.5 MW turbine (cost in $)	981	981	0.0%	
Turbine Capital per kW sans BOS (cost in $)	690	690	0.0%	
LEVELIZED REPLACEMENT COSTS (LRC) ($10.7 kW)	16	16	0.0%	0.0%
O&M $20/kW/Yr (O&M)	30	30	0.0%	0.0%
Land ($/year/turbine)	5	5	0.0%	
NET 5.8 m/s ANNUAL ENERGY PRODUCTION MWh (AEP)	4439	4598	3.6%	3.6%
Net 6.7 m/s ANNUAL ENERGY PRODUCTION Energy MWh (AEP)	5519	5708	3.4%	3.4%
Fixed Charge Rate	11.85%			
COE at 5.8 m/s $/kWh	0.0480	0.0464	-3.5%	
COE at 6.7 m/s $/kWh	0.0386	0.0374	-3.3%	

Figure 9.3. Cost of energy comparison for trade-off 1, tower height increase

9.3.2. Wind Energy Generation—Trade-Off 2: Gearbox Cost Reduction / Blade Diameter Increase (Near Net Cost Neutral)

In this analysis, the hub height was held fixed at 65 meters, but the blade diameter was increased to 75 meters from 70 meters with a cost neutral result (*see* Figure 9.5, Figure 9.6). Using reference numbers (4), (5), and (6) above, per-blade cost increases are calculated as follows.

For the 75-meter diameter blades:

Material:	$.4019R^3—955.24$	=	$20,238.81
Labor:	$2.7445R^{2.5025}$	=	$23,849.34
Subtotal T		=	$44,088.05
Total per blade = T ÷ 0.72		=	$61,233.41

For the 70-meter diameter blades:

Material	=	$16,276.22
Labor	=	$20,067.51
Subtotal T	=	$36,343.73
Total per blade = T ÷ 0.72	=	$50,477.40
Difference per blade	=	$61,233.41 (75m) - $50,477.40 (70m) $10,756.01
Total for (3) blades	=	$32,268.00

This figure compares favorably with the decrease in estimated cost of the gearbox at $31,000 (Figure 9.6), and justifies an approximate trade-off because both figures are given in 2002 dollars.

LWST Baseline (2002) Turbine Calculations - Class 4			
10 m windspeed	5.8	m/s	
Weibull K parameter	2.00		
Rated power	1500	kW	
Rotor Dia.	70	meters	
Hub height	65	meters	
Altitude	0	meters	
Air Density	1.225	kg/m^3	
Rotor Cp	0.47		
Cut-in windspeed	3	m/s	
Cut-out windspeed	26	m/s	
Power law shear exponent	0.143		
Hub height windspeed	7.58		
Rated windspeed	11.35	m/s	
Conversion Efficiency	0.95	0.015	
	0.005	0.005	
Soiling Losses	3.5%		
Array Losses	5.0%		
Availability	98.0%		
	Turbine	Weibul Cp	Weibul betz
Energy capture (MWh/year)	4439.53	7668.58	10177.70
Capacity Factor	33.79%		
Energy capture ratio	57.89%		

Figure 9.4. Baseline wind energy with 70-m diameter blades is 4,439 MWh per year; wind energy trade-off 2, blade diameter increase (COE benefit of 7.5%)

LWST Baseline (2002) Turbine Calculations - Class 4			
10 m windspeed	5.8	m/s	
Weibull K parameter	2.00		
Rated power	1500	kW	
Rotor Dia.	75	meters	
Hub height	65	meters	
Altitude	0	meters	
Air Density	1.225	kg/m^3	
Rotor Cp	0.47		
Cut-in windspeed	3	m/s	
Cut-out windspeed	26	m/s	
Power law shear exponent	0.143		
Hub height windspeed	7.58		
Rated windspeed	10.84	m/s	
Conversion Efficiency	0.95	0.015	
	0.005	0.005	
Soiling Losses	3.5%		
Array Losses	5.0%		
Availability	98.0%		
	Turbine	Weibul Cp	Weibul betz
Energy capture (MWh/year)	4798.56	8803.21	11683.58
Capacity Factor	36.52%		
Energy capture ratio	54.51%		

Figure 9.5. Convoloid GB Wind Energy with 75-m diameter blades is 4,798 MWh per year; wind energy generation trade-off 2. blade diameter increase (COE benefit of 7.5%)

COE PROJECTION SHEET

Baseline Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - 70 Meter Rotor
Improved Turbine: 1.5 MW - 3 Bladed Upwind/Pitch Controlled - Convoloid GB/75 Meter Rotor
Gearbox Cost Savings Invested in Rotor Diameter.

	70	75		
Rating (kWs)	1500	1500		
Component	**Baseline Component Costs $1000**	**Projected Component Costs $1000**	**Component Percent Improvement**	**Major Cost Element % Improvement**
Rotor	248	278	12.2%	
Blades	148	178	20.4%	
Hub	64	64	0.0%	
Pitch mchnsm & bearings	36	36	0.0%	
Drive train,nacelle	563	532	-5.5%	
Low speed shaft	20	20	0.0%	
Bearings	12	12	0.0%	
Gearbox	151	120	-20.5%	
Mech brake, HS cpling etc	3	3	0.0%	
Generator	98	98	0.0%	
Variable spd electronics	101	101	0.0%	
Yaw drive & bearing	12	12	0.0%	
Main frame	64	64	0.0%	
Electrical connections	60	60	0.0%	
Hydraulic system	7	7	0.0%	
Nacelle cover	36	36	0.0%	
Control, safety system	10	10	0.0%	
Tower	101	101	0.0%	
TURBINE CAPITAL COST (TCC)	921	921	-0.1%	-0.1%
Foundations	49	49	0.0%	
Transportation	51	51	0.0%	
Roads, civil works	79	79	0.0%	
Assembly & installation	51	51	0.0%	
Elect interfc/connect	127	127	0.0%	
Permits, engineering	33	33	0.0%	
BALANCE OF STATION COST (BOS)	388	388	0.0%	0.0%
Project Uncertainty	162	162	0.0%	
Initial capital cost (ICC)	1,472	1,471	0.0%	
Installed Cost per kW for 1.5 MW turbine (cost in $)	981	981	0.0%	
Turbine Capital per kW sans BOS (cost in $)	690	690	-0.1%	
LEVELIZED REPLACEMENT COSTS (LRC) ($10.7 kW)	16	16	0.0%	0.0%
O&M $20/kW/Yr (O&M)	30	30	0.0%	0.0%
Land ($/year/turbine)	5	5	0.0%	
NET 5.8 m/s ANNUAL ENERGY PRODUCTION MWh (AEP)	4439	4798	8.1%	8.1%
Net 6.7 m/s ANNUAL ENERGY PRODUCTION Energy MWh (AEP)	5519	5906	7.0%	7.0%
Fixed Charge Rate	11.85%			
COE at 5.8 m/s $/kWh	0.0480	0.0444	-7.5%	
COE at 6.7 m/s $/kWh	0.0386	0.0361	-6.6%	

Figure 9.6. Cost of energy comparison for trade-off 2—blade diameter increase

9.3.3. Wind Energy Generation—Trade-Off 3: Gearbox Cost Reduction / Blade Diameter Increase (Near Net Cost Neutral)—Reduced Wind Speed

In this analysis, the hub height was held fixed at 65 meters but the blade diameter was increased to 75 meters from 70 meters with a cost-neutral result. Further, the nominal wind speed was reduced until the energy capture equaled that of the Baseline 1.5-MW unit at 4,439 MWh per year (Figure 9.7).

LWST Baseline (2002) Turbine Calculations - Class 4

10 m windspeed	5.8	m/s		
Weibull K parameter	2.00			
Rated power	1500	kW		
Rotor Dia.	70	meters		
Hub height	65	meters		
Altitude	0	meters		
Air Density	1.225	kg/m^3		
Rotor Cp	0.47			
Cut-in windspeed	3	m/s		
Cut-out windspeed	26	m/s		
Power law shear exponent	0.143			
Hub height windspeed	7.58			
Rated windspeed	11.35	m/s		
Conversion Efficiency	0.95		0.015	
	0.005		0.005	
Soiling Losses	3.5%			
Array Losses	5.0%			
Availability	98.0%			
	Turbine		Weibul Cp	Weibul betz
Energy capture (MWh/year)	4439.53		7668.58	10177.70
Capacity Factor	33.79%			
Energy capture ratio	57.89%			

Figure 9.7. Baseline with 70-m diameter blades W.E. with Class 4 wind speed (5.8 m/s) equals 4,439 MWh/yr

LWST Baseline (2002) Turbine Calculations - Class 4

10 m windspeed	5.539	m/s		
Weibull K parameter	2.00			
Rated power	1500	kW		
Rotor Dia.	75	meters		
Hub height	65	meters		
Altitude	0	meters		
Air Density	1.225	kg/m^3		
Rotor Cp	0.47			
Cut-in windspeed	3	m/s		
Cut-out windspeed	26	m/s		
Power law shear exponent	0.143			
Hub height windspeed	7.24			
Rated windspeed	10.84	m/s		
Conversion Efficiency	0.95		0.015	
	0.005		0.005	
Soiling Losses	3.5%			
Array Losses	5.0%			
Availability	98.0%			
	Turbine		Weibul Cp	Weibul betz
Energy capture (MWh/year)	4439.74		7667.46	10176.21
Capacity Factor	33.79%			
Energy capture ratio	57.90%			

Figure 9.8. Convoloid GB with 75-m diameter blades W.E. with low wind speed (5.539 m/s) equals 4,439 MWh/yr

9.4. Best Trade-Off Model

The effects of these trade-offs are summarized in Table 9.1.

Table 9.1. Summary of Wind Energy Generation Options

Turbine Configuration	COE	Cost Reduction
Baseline configuration—5.8 m/s wind speed (65-m tower / 70-m diameter rotor)	0.0480	—
Trade-off 1—5.8 m/s wind speed gearbox savings / increase tower height (75-m tower / 70-m diameter rotor)	0.0464	3.5%
Trade-off 2—5.8 m/s wind speed gearbox savings / increase rotor diameter (65-m tower / 75-m diameter rotor)	0.0444	7.5%
Trade-off 2—slower 5.539 m/s wind speed gearbox savings / increase rotor diameter (65-m tower / 75-m diameter rotor)	0.048	—

From this data, two distinct advantages can be highlighted (tabulated in Table 9.1). The first is that, for best performance, trade-off 2 is decidedly the best, offering a cost reduction in COE of 7.5%. The second advantage is that which most explicitly meets the objectives of the NREL LWST program, that is, to produce cost-effective energy capture at lower wind speeds. Using alternative 3, an energy capture of 4,439 MWh per year can be obtained with a conventional Baseline 1.5-MW turbine at 5.8 m/s wind speed (Class 4) (70-m diameter rotor) with a conventional gearbox or 4,439 MWh per year energy capture with a Convoloid gearbox incorporated into the design at 5.39 m/s (Class 3) (75-m diameter rotor).

9.5. Objectives of NREL LWST Development Program

The U.S. Department of Energy has a program goal to develop competitive wind power generation in regions of lower wind speeds—specifically Class 4 winds such as those commonly found in the Midwestern and western United States. Section 1.1.1 provides further detail. The energy capture of the cost-neutral Convoloid gearbox–blade diameter increase typified by alternative 3 using Class 3 winds (5.539 m/s) equals the energy capture of the classic involute Baseline design using Class 4 winds (5.8 m/s). This result exactly matches original NREL LWST program objectives as directed to improve energy capture of the wind resources found in the Midwestern United States.

9.6. Potential Effects of Convoloid Gearing Technology

9.6.1. Performance

Several alternatives exist to improve the performance of wind turbine generators (WTG) through the application of Convoloid gearing technology. At one end of the possible spectrum is the direct replacement of Convoloid gears throughout the gearing system of an existing wind turbine gearbox. In this case, the gear operating stresses both for surface durability and tooth bending are considerably lower than those of the comparable involute gearbox. Under like operating conditions, this should significantly reduce O&M costs due to the greatly increased life of the Convoloid gearbox (see Section 3.3.3 for details).

There are additional extenuating conditions that must be considered to achieve full advantage of such Convoloid gear retrofits.

- The bearing compliment stresses likewise must be examined and possibly redesigned to achieve a life expectancy consistent with that of the Convoloid gearing. If this analysis is not done, then the bearing compliment as originally designed could become the limiting factor in any O&M cost and life projections.
- Comprehensive lubrication analysis.

- Analysis of other components conducted to ensure that no modifications due to the Convoloid design have negatively impacted these components.

At the other end of the spectrum is a completely redesigned WTG, where the size of the Convoloid gearing and the respective operating center distances have been reduced to achieve bending and surface durability stresses at or below those of the involute gearbox it replaces. At the designer's choice, new mounting interface provisions can be used or the existing mounting configuration of the involute gearbox being replaced can be maintained for interchangeability. Here, again, bearing design and ratings are extremely important considerations, as are other factors such as lubrication.

9.6.2. Economic Factors

The economic impact of Convoloid gearing technology applied to wind turbine gearboxes can be directly related to the design and manufacturing alternatives related in the above section. Should a direct replacement of Convoloid gears in place of the involutes in a gearbox be analyzed, the following is projected.

- In accordance with Section 3.6, there is an additional cost of bearings in the bearing complement to achieve an expected life of the bearings coincident or nearly so with the life of the replacement Convoloid gearing.
- The production costs of the Convoloid gears will be slightly higher than the involute set they replace primarily due to single-flank grinding times required for Convoloid gearing at its present stage of development compared with double-flank grinding capabilities of the gear manufacturing state of the art for involutes.

At the other end of the spectrum is the option to completely redesign the gearbox using Convoloid technology. In Section 8.1, it was established that a conservative estimate of the cost savings of this alternative is 21% of the nominal cost of an involute gearbox. Through its market research and information gathering from reliable industry sources, Genesis Partners has established a nominal cost savings of $35,420 per MW (2008 $) of gearbox rated capacity.

Convoloid gearboxes can be designed to be significantly smaller, lighter, and less costly (for equal gear stress levels) than involute gearboxes, therefore compromises in design approaches to optimize lower overall capital costs and O&M costs are possible. Designing for lower Convoloid gearbox stresses improves the O&M costs, provided that bearing ratings, costs, and field performance keep pace with the improvements inherent in the gearing.

9.6.3. Extended Service Life

The significant increases in anticipated gear life have immediate impacts on the O&M costs. Increased gear life causes greater intervals between repair and replacement of the gearboxes. Longer intervals between repairs means that there will be fewer repairs within the expected service life of a turbine—greatly reducing O&M costs. With proper maintenance, achieving the full wind turbine service life is possible with gearboxes containing Convoloid gearing.

Appendix A. Three-Hole Test

This appendix provides supplementary design, rating, and performance data developed up to the cessation of the 3-Hole Test. Details of test instrumentation parameter are listed as are general features of the 3-Hole Test rig. Descriptions of the actual test protocols, gear performance, and failure analysis are contained at the end of this appendix.

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Involute Gear Rating per AGMA 2001

American Gear Manufacturers Association Time: 15:40:34 Date: 2008/11/20
Version 1.07 GEAR RATING PER AGMA 2001-C95 \ Data Set: 1
Genesis 3-Hole Test
7NDP 16/35

****

*** ECHO OF INPUT ***

1		1				Data set number
2	Genesis 3-Hole Test					
3	7NDP		16/35			
4						
5		2		2		In/Out = 1, si = 2, en 3 both units
6		2				= 1 Short = 2 Long Output Form
7		1				= 1 External = 2 Internal Gear
8		1				= 1 Single = 2 Double Helical
9	20.000000				deg	Normal pressure angle
10	7.0000000				in^-1	Normal diametral pitch
11	18.697400				deg	Helix angle
12	4.0000000				in	Working center distance
13		16		35		Pinion / Gear number of teeth
14	2.7930000		5.7570000		in	Pinion / Gear tip diameter
15	1.5000000		1.4500000		in	net face width (gap not included)
16		11				Transmission Accuracy
17	0.0000000		0.0000000		in	Pitch variation
18	0.45865739		0.75120803			addendum modification coefficient
19	0.16000000	E-01	0.16000000	E-01		Tooth thinning for backlash (norm)
20		10000		10000		Number of teeth on tool
21	1.4000000		1.4000000			tool addendum (normalized)
22	0.82146946	E-14	0.82146946	E-14		tool addendum modification coeff
23	0.0000000		0.0000000			tool protuberance (normalized)
24	0.39360256		0.39360256			tool tip radius (normalized)
25	0.0000000		0.0000000			Stock allowance per side (norm)
26	32.000000		32.000000		mu in	surface finish (Ra)
27	0.0000000		0.0000000		in	rim thickness (0 = solid disk)
28	3.0000000				in	Pinion bearing span
29	0.85000000				in	Pinion offset from center bearing span
30	2200.0000				rpm	Pinion speed (rpm)

31	7000.0000				lbs in	Pinion torque
32	300.00000				hours	Design life
33		1		1		number of mesh contacts
34		2		2		Idler use = 1 yes = 2 no
35		4		4		Surface hardness, = 1 HV = 2 HBN. . .
36	58.000000		58.000000			Surface hardness values
37		4		4		Core hardness, = 1 HV = 2 HBN...
38	30.000000		30.000000			Core hardness values
39		1		1		mat, = 1 steel, = 2 cast iron, ...
40		1		1		Material sub classes
41		3		3		Heat treatment = 1 TH/NA = 2 Flam..
42		1		1		= 1 quench & temper = 2 Normalized
43		0		0		pattern type =0 na =1 A =2 B
44		2		2		material grade
45	0.0000000		0.0000000		psi	Allow. contact stress number
46	0.0000000		0.0000000		psi	Allow. bending stress number
47	0.29000000		0.29000000			Poissons ratio
48	30000000.		30000000.		psi	Modulus of elasticity
49		2				Lapped adjusted assy = 1 yes = 2 no
50		1				Lead crowned/corrected = 1 yes = 2 no
51		3				= 1 open = 2 commer = 3 prec = 4 xtra
52		2				Point stress calc = 1 hpstc = 2 tip
53	0.0000000					Pitting resistance service factor
54	0.0000000					Bending strength service factor
55	1.0000000					Pitting resistance safety factor
56	1.0000000					Bending strength safety factor
57	1.0000000		1.0000000			Reliability factor
58	1.0000000					Overload factor (clause 9)
59	0.0000000				lbs	Maximum tangential load
60	0.0000000					Load distribution factor
61	0.0000000					Load Distribution factor at yield
62		2				Application = 1 conser = 2 Indus
63	0.0000000					Size factor (clause 20)
64	0.0000000					Temperature factor (clause 19)
65	0.0000000					Surface condition factor
66	0.0000000					Dynamic factor (clause 8)
67	0.0000000		0.0000000			Life factor pitting
68	0.0000000		0.0000000			Life factor bending
69	0.85000000		0.85000000			Life factor pitting @ 10^10
70	0.90000000		0.90000000			Life factor bending @ 10^10

Rating Routine Error Messages See User's Manual for More Information

1) WARNING: Calculated pitting safety factor for pinion (0.937) is less than input value.
2) WARNING: Calculated pitting safety factor for gear (0.954) is less than input value.

7) WARNING: Contact Load Factor (1959.1) exceeds allowable contact load factor for the pinion.
8) WARNING: Contact Load Factor (2032.6) exceeds allowable contact load factor for the gear.

Effective Case Error Messages See User's Manual for More Information

5) NOTE: Minimum case depth is calculated using contact stress above the maximum value recommended in the standard.

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AMERICAN GEAR MANUFACTURERS ASSOCIATION Time: 15:40:34 Date: 2008/11/20
Version 1.07 GEAR RATING PER AGMA 2001-C95 Data Set: 1 Page 3 of 7
Genesis 3-Hole Test
7NDP 16/35
****
SINGLE HELICAL PINION GEAR

	** GEAR GEOMETRY **	
Number of Teeth (-)	16	35
Normal Diameter Pitch (Module) 1/in (mm)		7.0000 (3.6286)
Normal Pressure Angle deg	20.0000	
Helix Angle deg	18.6974	
Op. Center Distance in	4.0000	
Outside Diameter in	2.7930	5.7570
Face Width in	1.5000	1.4500
Effective Face Width in	1.4500	
Gear Ratio (-)	2.1875	

** GEAR GEOMETRY—NORMALIZED **		
Addendum Mod. Coeff (-)	0.4587	0.7512
Tooth Thinned for B/L (-)	0.0160	0.0160
Stock Allow./Tooth Flank (-)	0.0000	0.0000

** TOOL GEOMETRY—NORMALIZED **		
Add. Mod. Coeff of Tool (-)	0.0000	0.0000
Protuberance of Tool (-)	0.0000	0.0000
Addendum of Tool (-)	1.4000	1.4000
Tool Tip Radius (-)	0.3936	0.3936
Number of Teeth on Tool (-)	10000	10000

** LOADING DATA **		
Design Life hours	300	
Pinion Torque (input) lb in	7000.0000	
Speed rpm	2200.00	1005.71
Pitch Line Velocity ft/min	1445.5433	
Max Tang. Load (input) lb	0.0000	
Type of Practice	INDUSTRIAL	
Type of Service	PRECISION ENCLOSED GEARING	
Reliability Factor(input) (-)	1.0000	1.0000

** ADDITIONAL INPUTS **		
Type of Gear Set	EXTERNAL	
Bearing Span in	3.0000	
Distance "s1" in	0.8500	
Rim Thickness in	0.0000	0.0000
Dynamic factor (-)	N/A	
Transmission Accuracy No (-)	11	
Abs. Pitch Variation (-)	N/A	
Lead Correction or Crown	YES	
Lapped or Adjusted	NO	
Number of Contacts (-)	1	1
Idler	NO	NO
Spur Loading	N/A	

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Genesis 3-Hole Test
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****
SINGLE HELICAL PINION GEAR

** INPUT FACTORS **		
Load Distribution Factor (-)	0.0000	
Dynamic Factor (-)	0.0000	
Surface Condition Factor (-)	0.0000	
Overload Factor-Default (-)	1.0000	
Size Factor (-)	0.0000	
Temperature Factor (-)	0.0000	
Load Dist. Ftr Overload (-)	0.0000	
Pitting Stress Cycle Fac (-)	0.0000	0.0000
Bending Stress Cycle Fac (-)	0.0000	0.0000
Pitt Stress Cycle @10^10 (-)	0.8500	0.8500
Bend Stress Cycle @10^10 (-)	0.9000	0.9000

** MATERIAL DATA **		
Material	STEEL	STEEL
Material Type	PLAIN STEEL	PLAIN STEEL
Material Grade or Class	GRADE 2	GRADE 2
Heat Treatment	CARB & HARDENED	CARB & HARDENED
Induction Hard. Pattern	N/A	N/A
Quench	QUENCH & TEMPER	QUENCH & TEMPER
Surface Finish mu in	32.0000	32.0000
Modulus of Elasticity psi	30000000.	30000000.
Poisson’s Ratio (-)	0.2900	0.2900
Allow. Cont. Stress (input) psi	0.	0.
Allow. Bend. Stress (input) psi	0.	0.
Core Hardness Number (-)	30.00	30.00
Core Hardness Scale	ROCKWELL C	ROCKWELL C
Surface Hardness Number (-)	58.00	58.00

** MATERIAL DATA **		
Surface Hardness Scale	ROCKWELL C	ROCKWELL C
Brinell Core Hardness (-)	287.16	287.16
Brinell Surface Hardness (-)	620.29	620.29
Allow Contact Str. No psi	225000.	225000.
Allow Bending Str. No psi	65000.	65000.
Elastic Coeff	(lb/in^2)^0.5	2283.2142

** EFFECTIVE CASE DATA **		
Process Fact / Core Coef psi	6400000.	6400000.
Maximum Effective Case in	0.0511	0.0508
Min. Eff./ Total Case in	0.0290	0.0290
Heavy Minimum in	0.0297	0.0297
Normal Minimum in	0.0225	0.0225

SINGLE HELICAL PINION GEAR

** PRESSURE ANGLES **		
Inv. of Norm.Press Angle (-)	0.014904	
Std. Trans. Pressure Angle deg	21.0192	
Normal Op. Press Angle deg	24.8704	
Trans. Op. Press Angle deg	26.1712	
Inv. of Trans. Press Angle (-)	0.017394	
Trans. Tip Press Angle deg	36.2466	31.1421
Inv. of T. Tip Press Angle (-)	0.100516	0.060709

** PITCHES **	
Transverse Dia. Pitch 1/in	6.6306
Normal Base Pitch in	0.4217
Transverse Base Pitch in	0.4423
Axial Pitch in	1.4000

** HELIX ANGLES **	
Op. Helix Angle deg	19.3920
Base Helix Angle deg	17.5319

** DIAMETERS **		
Ref. Pitch Diameter in	2.4131	5.2786
Operating Pitch Diameter in	2.5098	5.4902
Root Diameter in	2.1378	5.0868
Root Diameter (ref. 908) in	2.1378	5.0868
Base Diameter in	2.2525	4.9273

** TOOTH GEOMETRY **		
Tooth Whole Depth in	0.3276	0.3351
Norm Tooth Thk (ref dia) in	0.2698	0.3002
Norm Top Land Thickness in	0.0913	0.0908

** CONTACT RATIOS **	
Trans Contact Ratio (-)	1.2439
Axial Contact Ratio (-)	1.0357
Total Contact Ratio (-)	2.2796

** MISCELLANEOUS FACTORS **		
Gear Ratio Factor (-)	0.6863	
Effective Protuberance (-)	0.0000	0.0000
Minimum Contact Length in	1.8787	
Fractional Part of m f (-)	0.0357	
Adjusted No of Teeth (-)	16.8914	36.9500
Fractional Part of m_p (-)	0.2439	

** LINE OF ACTION DATA **						
Points Along LOA	A	B	C	D	E	F
Dist (c_1-c_6) in	0.2756	0.3834	0.5535	0.7178	0.8257	1.7642
(gamma_a-e)	-0.5021	-0.3073	0.0000	0.2970	0.4918	
Rol. Ang (eqs)deg	14.0189	19.5057	28.1573	36.5189	42.0057	

SINGLE HELICAL PINION GEAR

** MESH FACTORS **	
Mesh Align Correction Factor (-)	1.0000
Surface Condition Factor (-)	1.0000
Mesh Alignment Factor (-)	0.0859
Lead Correction Factor (-)	0.8000
Pinion Proportion Factor (-)	0.0384
Pinion Proportion Modif (-)	1.1000
Load Distribution Factor (-)	1.1025

** AGMA 908 DATA (normalized) **		
I-Factor (-)	0.1776	
J-Factor (-)	0.5112	0.5456
Working Center Distance (-)	28.0000	
Number of Teeth (-)	16.0000	35.0000
Effective Face Width (-)	10.1500	
X-Factor (-)	0.4587	0.7512
Number of Teeth (tool) (-)	10000.0000	10000.0000

** AGMA 908 DATA (normalized) **		
X-Factor (tool) (-)	0.0000	0.0000
Tool Addendum (-)	1.4000	1.4000
Tool Tip Radius (-)	0.3936	0.3936
Effective Protuberance (-)	0.0000	0.0000
Tooth Thinning for B/L (-)	0.0160	0.0160
Strength Conc. Factor (-)	1.5173	1.5956

** STRESS FACTORS SUMMARY **		
Load Dist Factor-Ovrload (-)	1.0909	
Number of Stress Cycles (-)	0.39600E+08	0.18103E+08
Reliability Factor (-)	1.0000	1.0000
Overload Factor (-)	1.0000	
Hardness Ratio Factor (-)	1.0000	1.0000
Size Factor (-)	1.0000	
Temperature Factor (-)	1.0000	
Yield Strength Factor (-)	0.7500	0.7500
Stress due to Wmax psi	47827.85	42617.50
Allow. Yield Strength psi	105612.70	105612.70
Max. Tangential Load lb	5578.1250	
Pitting Stress Cycle Fac (-)	0.9681	0.9861
Bending Stress Cycle Fac (-)	0.9931	1.0070
Dynamic Factor (-)	1.0903	
Transmission Accuracy No (-)	11	
Abs. Value of Pitch Var. in	N/A	
Calculated Driver Power hp	244.4444	
Member Torque lb in	7000.00	15312.50
Max. Pitch Line Vel ft/min	10000.0000	
Tangential Load lb	5578.1250	

****

AMERICAN GEAR MANUFACTURERS ASSOCIATION Time: 15:40:34 Date: 2008/11/20
Version 1.07 GEAR RATING PER AGMA 2001-C95 Data Set: 1   Page 7 of 7
Genesis 3-Hole Test
7NDP 16/35

****

SINGLE HELICAL PINION GEAR

****

*** MAIN RATING VALUES ***

****

** PITTING **		
Allowable Transmitted Power at Unity Service Factor hp	214.4104	222.4548
Allowable Power at input Service Factor hp	N/A	N/A
Service Factor (input) (-)	0.0000	
Service Factor (calc) (-)	N/A	N/A
Contact Load Factor psi	2233.4826	
Allow Contact Load Fact psi	1959.0620	2032.5634
Contact Stress Number psi	232588.2285	

** PITTING **		
Allowable Power at input Safety Factor hp	214.4104	222.4548
Safety Factor (input) (-)	1.0000	
Safety Factor (calc) (-)	0.94	0.95
** BENDING **		
Allowable Transmitted Power at Unity Service Factor hp	263.0853	284.7086
Allowable Power at input Service Factor hp	N/A	N/A
Service Factor (input) (-)	0.0000	
Service Factor (calc) (-)	N/A	N/A
Unit Load psi	26928.8793	
Allowable Unit Load psi	28982.4234	31364.5225
Bending Stress Number psi	59975.8751	56197.9283
Allowable Power at input Safety Factor hp	263.0853	284.7086
Safety Factor (input) (-)	1.0000	
Safety Factor (calc) (-)	1.08	1.16
** Power Summary **		
Input Power hp	244.4444	
Allowable Transmitted Power for input Service Factor hp	N/A	
Allowable Transmitted Power for input Safety Factor hp	214.4104	

3-Hole Test Instrumentation Data Points

3-Hole Test Instrumentation Parameters

Notes
1. Involute gearbox is gearbox 1 (nearest the motor).
2. Convoloid gearbox is gearbox 2.
3. Shaft notation with respect to torque actuator or Lebow Transducer.

	Recorded in Data File	Alarms	Signal Type
Convoloid Oil Flow	Y		Pulses
Convoloid Oil Temp Out	Y		
Convoloid Oil Temp In	Y		
Convoloid Particles	Y	Excessive Particles	
Convoloid Oil Pressure at Manifold	Y	Loss of Oil Pressure	DC Volts
Convoloid, Outside, Lebow Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid, Outside, Intermediate Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid, Outside, Torque Actuator Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid, Inside, Lebow Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid, Inside, Intermediate Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid, Inside, Torque Actuator Bearing Temperature	Y	Excessive Temperature	TC Isolated
Convoloid Reservoir Temperature			TC
Convoloid Magnetic Chip Detector	Y	Particles Detected	DC Volts
Convoloid Filter Pressure Drop	Y	Excessive Filter Pressure Drop	
Convoloid X Vibration	Y	Excessive Vibration	AC mV

	Recorded in Data File	Alarms	Signal Type
Convoloid Y Vibration	Y	Excessive Vibration	AC mV
Convoloid Z Vibration	Y	Excessive Vibration	AC mV
Convoloid Total Vibration (calculated)	Y	Excessive Vibration	calc
Convoloid Heat Exchanger Oil Temp—Out			TC
Convoloid Particle Count 4u			DC Volts
Convoloid Particle Count 6u		High Count	DC Volts
Convoloid Particle Count 14u		High Count	DC Volts
High Speed Shaft RPM (Motor RPM)	Y		Pulses
Circulating Torque	Y		SG
Windup Force HP	Y		
Input Wattage	Y		
Ambient Temperature (Computer Side)	Y		TC
Ambient Temperature (Test Fixture Side)	Y		TC
Smoke Detector	Y	Smoke Detected	
Setpoint	Y		
Bypass Valve	Y		
Hydraulic Actuator Pressure		Low Pressure	DC Volts
Involute Oil Flow	Y		
Involute Oil Temp Out	Y		
Involute Oil Temp In	Y		
Involute Particles	Y	Excessive Particles	
Involute Oil Pressure	Y	Loss of Oil Pressure	DC Volts
Involute, Outside, Lebow Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute, Outside, Intermediate Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute, Outside, Torque Actuator Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute, Inside, Lebow Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute, Inside, Intermediate Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute, Inside, Torque Actuator Bearing Temperature	Y	Excessive Temperature	TC Isolated
Involute Reservoir Temperature	Y		TC
Involute Chip Detector	Y	Particles Detected	DC Volts
Involute Filter Pressure Drop	Y	Excessive Filter Pressure Drop	
Involute X Vibration	Y	Excessive Vibration	AC mV
Involute Y Vibration	Y	Excessive Vibration	AC mV
Involute Z Vibration	Y	Excessive Vibration	AC mV
Involute Total Vibration	Y	Excessive Vibration	calc
Involute Heat Exchanger Oil Temp Out			TC
Involute Particle Count 4u			DC Volts
Involute Particle Count 6u		High Count	DC Volts
Involute Particle Count 14u		High Count	DC Volts

Overview of 3-Hole Fixture Design and Shakedown

To free up manufacturing space, it was necessary to move testing outside of the original facility. Instead of locating the test in a fixed place, all equipment was fit into a 20-ft long shipping container (Figure A.1). Recently refurbished and customized with a man-door, window, heat exchanger openings (center bottom), and air-intake vents, this enclosure allowed the entire testing fixture to be set into place through the large end doors. All power and controls are enclosed and all that is necessary for testing is to connect power (460 v, 3-phase), air, and telephone/DSL.

Figure A.1. Test fixture enclosure

Manufacture of the test fixture was straightforward. The test fixture (Figure A.2) was covered with epoxy-based paint inside. If testing must occur with synthetic oil, multiple fittings designed into the fixture allow ready access for instrumentation and oil flow. Not shown in the figure are the internal baffles or the lubricant deflection funnels used to direct oil falling out of the test gearboxes past the chip detectors.

During assembly of the gearboxes to the test fixture, it was found that the mounting surface of the fixture had deformed when the top was welded to the tank. The resulting gaps were sufficient that oil would leak from between the gearbox and the surface. Additionally, potential harmful stresses could be applied to the gearbox housings (and subsequently to the bearings) if the mounting bolts are tightened beyond recommended torque values. Therefore, components were removed from the fixture and the mounting surface was milled. Subsequent assembly showed a much improved fit. All electrical relays and primary controls were contained in one enclosure (Figure A.2; Figure A.3).

Figure A.2. The 3-Hole Test fixture after paint and initial top milling

Figure A.3. The 3-Hole Test electrical circuit assembly

Figure A.4. Electrical enclosure

The computer control-system equipment was the same as used with the Micon 108 test, with a few additional control points—specifically, torque control and fire alarms sensors (*see* Figure A.5). After initial operation, temperatures were excessive around the computer equipment, requiring the installation of an air conditioner unit and separation of the test rig from the computers. The insulated barrier includes a polycarbonate window for visual monitoring of the test.

Due to the long delay of achieving Convoloid gearing with good contact, both gearboxes were assembled with involute gearing and mounted to the fixture for shakedown and preliminary setup purposes. When the gearboxes were in place, the motor and other components could be installed. Extreme care was taken to align the various shafts, thereby reducing the risk of vibrations that could possibly affect data readings. After the shafts were aligned, the gearboxes and motor were pinned to the fixture with removable taper pins. When components needed to be removed during testing, they could be realigned easily. Removal of the gearboxes for part change outs (versus changing parts in place) is preferred, to keep out foreign contaminants and to be assured of proper bearing settings.

After all the components were in place (Figure A.6), preliminary operation at modest speeds resulted in excessive vibrations that could be felt in the floor of the test container. The cause originally pointed to a bent shaft in the torque actuator. After repairs, however, significant vibrations still were evident. A number of couplings then were found to have severe run-out. Subsequent repairs—including adding machined alignment sleeves over the couplings—did not correct the vibration.

Figure A.5. View inside the test container showing the computer monitoring system and polycarbonate view window

Figure A.6. Initial assembly of 3-Hole Test fixture

Further investigation revealed that the chosen couplings were not able to self-center due to the weight being carried between the two gearboxes (Lebow torque transducer on one shaft pair and the torque actuator between the other). Additionally, because both couplings on a given shaft (e.g., on either side of the Lebow) were flexible, a dynamically indeterminate load arrangement occurred causing the components in the middle to vibrate. (See Figure A.7.)

Figure A.7. The 3-Hole Test fixture showing couplings with alignment covers

The vibration issue finally was solved by using different couplings with a rigid-flexible arrangement. This type of coupling reduces a degree of freedom, thereby better locating the coupled equipment (Figure A8, Figure A.9). Installation involves bolting the two halves together. With the bolt clearances, however, out–of-balance vibrations again were felt. Careful installation using a dial indicator to center the bolted connections resulted in very smooth operation.

Figure A.8. Fixed half of new coupling

Figure A.9. Flexible half of new coupling

Installation of rigid-flexible type couplings in place of the previous coupling design resulted in greatly improved operation. No vibration from the actuator side was evident and light vibration was felt in the torque transducer side. Efforts to balance the system, however, were not fruitful. Investigation revealed that the internal bore was not perpendicular and was not centered in one of the coupling hubs, causing the transducer assembly to oscillate. To allow vibration monitoring (necessary for unattended operation) this coupling part required correction. Although the vibration precluded unattended operation, shakedown of the fixture continued. A certain amount of torque was desired to test the system instrumentation, therefore it was thought that a reasonable amount hydraulic pressure could be applied to the torque actuator (approximated to be 25% of the operational torque). After a period, some electronic verification was possible and it was determined that the actual load was nearer 45%.

As the test fixture warmed to operational temperatures (155°F to 160°F), pressures in the lubrication manifold of the Convoloid dropped well below (~20 psi) the specified pressure of 25 psi. (At the same time and temperatures, the involute pressures were only a slightly less than required.) Concerns regarding supplying lubricant to the gear meshes necessitated investigation of the lubricant supply pumps. These concerns are warranted because oil must be supplied to the gearing as they come into mesh. Depending on the load direction (*see* Figure A.10), the oil supplied to the gearing could come from the bottom. Without enough pressure there is the possibility of insufficient lubrication.

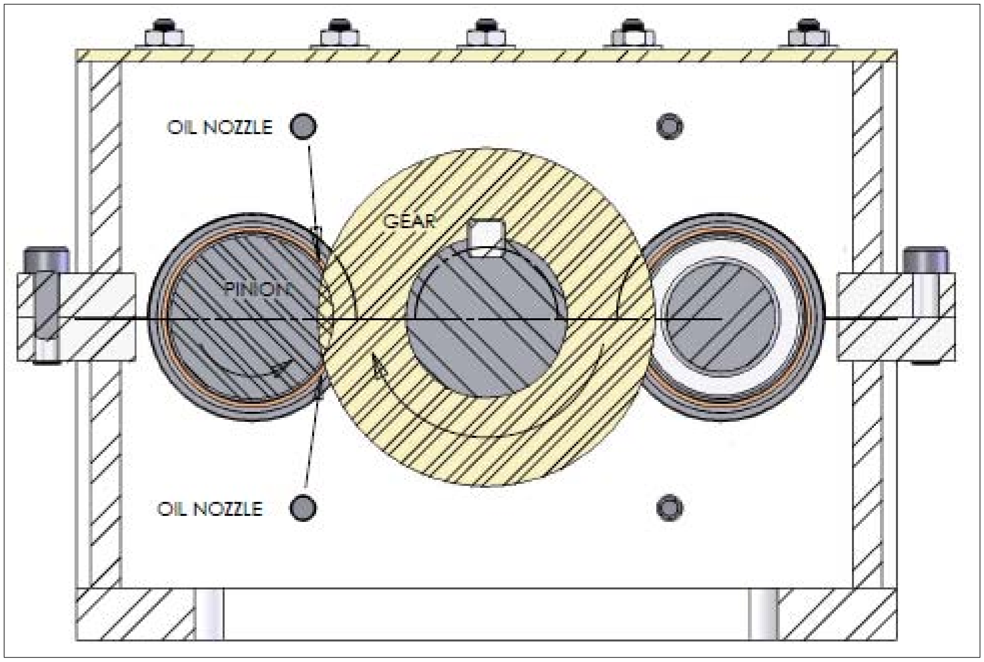


Figure A.10. Cutaway of the 3-hole gearbox showing the gear mesh lubricating method

Positive displacement pumps require a bypass valve to prevent over-pressure during start-up with cold oil. At high pressures, lubricant is diverted from the high-pressure side of the pump back to the low-pressure side, causing a portion of the oil to circulate within the pump. As the pressure decreases the valve closes. The valve is controlled by spring pressure created by a set-screw. This screw is covered with a cap to prevent air leakage and contamination.

When the cap was being removed, the adjusting set-screw stuck to the cap and, inadvertently, it also was being removed. At some point, the pressure fell enough that no oil was being fed to the gear meshes. The fact that the set screw was being removed was not realized until it fell out of the pump housing—which caused 160°F oil to flow out of the machine. The machinery was shut down immediately and the pump was repaired. Inspection of the gear meshes, however, revealed extensive scuffing (welding and tearing of the loaded tooth flank) (Figure A.11; Figure A.12).

Figure A.11. Severe scuffing on the Convoloid pinions due to loss of lubricant

Figure A.12. Severe scuffing on the Convoloid pinions due to loss of lubricant

These flanks no longer could be used for testing, therefore it was thought that continued operation for shakedown purposes would not further damage the parts. As expected, when under no load the Convoloid gearbox was louder than the involute gearbox. After applying the same load as used previously, however, the Convoloid gearbox became quieter than the involute gearbox.

As instrumentation was calibrated, the torque-monitoring equipment revealed that the system was operating considerably above the expected test torque. The applied torque was reduced immediately, but the system had operated for approximately 10 hours at the greater load. The system operated for 5 more hours before an unexpected noise in the Convoloid gearbox caused the test to be shut down. Initial inspection of the inside of the gearbox revealed an interference pattern on the unloaded flanks. The gearbox was disassembled, and it was found that the intermediate shaft had broken. It is interesting to note that the flanks with severe scuffing appeared to have smoothed during the operation at lower torque levels.

An immediate investigation into how to prevent a similar incident led to (1) An increase in pump volume to maintain the requisite 25 psi at the distribution manifold; and (2) A redesign of the intermediate shaft to strengthen the bearing shoulder area. To address the low oil pressure, pumps with increased capacities were installed. Subsequent operation confirmed proper flow (Figure A.13).

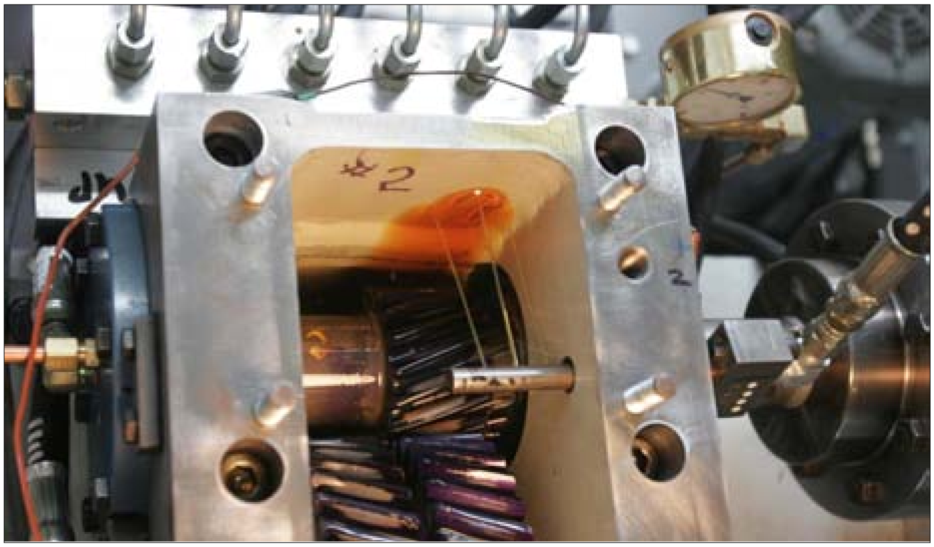


Figure A.13. The 3-Hole Test gear mesh oil nozzles with proper oil pressure (rotated to show flow)

Redesign of the intermediate shaft assembly commenced immediately following the failure. Safety factors of the new shaft design are nearly two times greater than with the original design. The new design required a trade-off with bearing size; however, with the controlled lubricant system and replacement of bearings planned with every gear change, the reduced bearing life was deemed to be sufficient for the test period. Manufacture of two shafts began immediately and four additional shafts were made to facilitate easier gearing changeover and longer shaft life.

Inspection of the coupling hubs on either side of the Lebow torque transducer revealed that the bores in the solid hubs were not round, nor where they perpendicular to the mounting face. This caused the vibration that was experienced during operation and explains why the hubs were so difficult to install. Subsequent balancing of the hubs showed that minimal correction was needed.

To allow the instrumentation to be qualified and to enable unattended operation, both gearboxes were assembled with involute gear sets. The number of cycles and load amount were logged for each loaded flank, so all test cycles could be tracked. Any gearing failures were documented to help generate a baseline for future Convoloid comparisons. A photo of the final test fixture configuration is shown in Figure A.14.

Testing progressed with both gearboxes using involute gearing. Completing the programming of the control system was of prime importance. Additionally, until the safeguards were in place—including emergency shutdown and notifications—unattended operation is impossible. These safeguards are necessary to prevent the failure of instrumentation or test equipment from catastrophically affecting the test fixture.

Figure A.14. Operational configuration of the 3-Hole Test fixture

The control system was designed to monitor critical aspects of the test fixture. Accelerometers in three planes are used to gather vibration data to determine any pending failures. (A view of the data screen for the vibration collection is shown in Figure A.15.) In previous testing, changes in the vibration level of the minimum amplitude planes correlated closely with bearing failures, and they also should give notice of pending gear wear. If wear begins causing loss of gear material, then chip detectors that are located in the oil flow from the gearboxes trigger an immediate shutdown of the test. Upon inspection of the gear teeth, additional testing can occur until the amount of tooth wear reaches 1.5% of the active tooth flank.

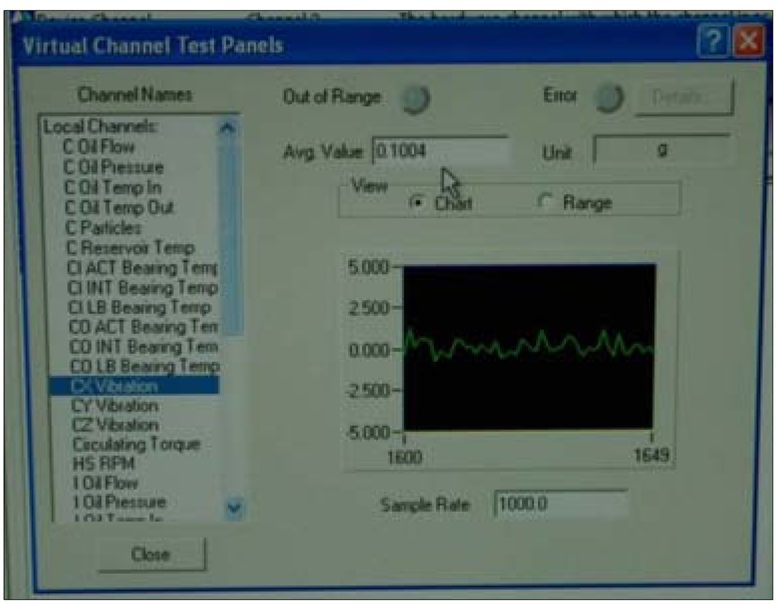


Figure A.15. Sample of the monitoring software page (vibration shown)

To catch system failures before they lead to damage of the test gearing, numerous sensors with alarm settings have been included. Among these are bearing temperature, lubricant flow and pressure, lubricant temperature, and applied torque. Additionally, external sensors recently were installed to shut down the system if smoke or fire occurs within the container. Lastly, a fail-safe system has been added—the drive, torque, and lube systems will shut down if the computer controller becomes damaged or ceases to operate. When any failure mode occurs, notification is sent to the test supervisors and Genesis personnel so they can inspect and evaluate the alarm condition.

One area of concern was found during the test operation: The ambient temperature within the test container exceeded 115°F after only 1 hour of having the test container closed. This occurred on a 65-degree spring day. Having such high temperatures is excessive for the instrumentation electronics, thus necessitating the installation of an air conditioner for cooling and a barrier between the test fixture and the test computers. It was important to protect the monitoring equipment because its failure could cause the test fixture to "run away." This led to the addition of the fail-safe circuit that is held closed as long as the computer is operational. If the electronic equipment fails the circuit opens, thus shutting down the system. Continued discussion of the test fixture operation is included below.

Micropitting Test Protocol
3-Hole Gearbox Test

Low Wind Speed Turbine (LWST) Project Phase Ii
Component Development
Revision 1
Modified 6-13-05

NREL Contract #ZAM-5-33200-12

April 9, 2007

Initial testing in the 3-hole gearboxes is intended to qualify profile corrections in the Convoloid tooth form. This testing is in preparation for the main extended comparison test. Previous test gears developed micropitting early in the test cycle. This failure mode is not in itself catastrophic but it can lead to macropitting and, in extended operation, to complete failure of the gear. Before extended testing can occur the micropitting phenomena must be solved. This procedure applies only to the initial micropitting test. Subsequent testing is addressed in later procedures. This procedure assumes that all equipment and instrumentation has been assembled and tested in the static state (including a pressure test of the fixture). The gearboxes and oil distribution manifolds should not yet be installed. Gear teeth should be uniquely labeled prior to assembly into the gearboxes.

First Start Procedure

1. Familiarize test personnel with instrumentation and test protocol. Set up and start manual test log.
2. Obtain and label one 3-oz sample of new oil from drum for later testing.
3. Fill each tank to the appropriate level with test oil. Oil should be pumped out of the barrel through a 3μ filter.
4. Activate the circulation pump to suspend any particles, and heat oil to 150°F.
5. Channel the outlets from the filters back into the sump, and activate primary lubrication pumps. Allow to run for at least 4 hours, or until the oil cleanliness reaches -/15/12.

6. Stop the lubrication pump (circulation motor should remain on) and attach lubrication manifolds and oil lines. Operate the pump to flush any particles then stop the pump.
7. Install the gearboxes, torque actuator, coupling, and motor.
8. Assemble the lubricant manifold and complete piping.
9. Ensuring no rotation of the gearboxes, start lubrication pump and verify lubricant flow through the housing. Check gearing spray-nozzle operation. Slowly hand-turn the shafts while lubricant flows, to assure no particles are present in the bearings.
10. Assure that all covers are in place and sealed.

Micropitting Test Procedure

1. Apply marking compound (bluing) to teeth flanks, and verify that each tooth is uniquely labeled.
2. Assure that the circulating and lubricant pumps are operating.
3. Take one 3-oz oil sample from each gearbox and perform a preliminary patch test.
4. Energize drive motor and slowly bring it up to operating speed (2,250 rpm).
5. Verify that all components are running smooth with minimal vibration.
6. Bring the load up to 10% of test torque (630 lb-in) and operate for 1 hour. Confirm that instrumentation operates properly. Continuously monitor particulates; if they exceed -/18/13, shut the system down and determine the reason. Increase load to 25% of test torque (1,575 lb-in) and operate for 1 hour.
7. Stop operation and inspect wear pattern. Photograph the results. If any wear is visible through the transition zone, evaluate the extent of the micropitting. The test can be stopped at this point for further analysis and modifications to the profile correction. Some micropitting due to lead variations is acceptable (up to 10% of the active face).
8. Restart and slowly increase the load to 50% of test torque (3,150 lb-in) and operate for 1 hour.
9. Stop operation and inspect and photograph wear pattern.
10. Repeat steps (7) and (8) for 75% (4,725 lb-in) and 100% (6,300 lb-in) load with inspections and photographs after each test. Note any abnormalities or early wear. If any micropitting is visible, then continuation operation of the test must be evaluated. Further analysis and profile correction could be required.
11. Once operation is proven at 100% of load, restart and operate for 24 hours and then re-inspect for micropitting.
12. If no micropitting is observed after the 24-hour test, then testing can progress to the extended comparison test.

Note that the bulk oil temperature should not exceed 155° F, measured at inlet to gearbox, at any time. Nor should any bearing temperature exceed 190° F (an alarm will sound). If either of these conditions exists, stop the test run and contact the instrumentation engineer.

Inspection of gear teeth will require removal of the cover plate. Inspection lights should be used at various angles to adequately view the tooth flanks. Photos should be in focus and of high enough resolution that subsequent analysis is possible. If wear is visible, then a decimal inch scale should be included in the photo (for comparison). Scale markings should be readily visible as should the tooth wear. Be aware of flash glare on the tooth surface.

If there is a shutdown due to mechanical or lubrication problems, the First Start Protocol is repeated, starting with step 5. The micropitting test procedure, at the discretion of the test supervisor, then commences either from the beginning or from the last successful load step. If there is a shutdown due to instrumentation problems, the protocol is repeated from the last successful load step. (Repeat one load step.)

Extended Comparison Test Protocol
3-Hole Gearbox Test

Low Wind Speed Turbine (LWST) Project Phase II
Component Development

Revision 1
Modified 6-13-05

NREL Contract #ZAM-5-33200-12

April 9, 2007

The initial (shakedown) testing should be completed as specified in the Micropitting Test Protocol. Once complete, testing progresses to a long-term phase where a macropitting failure on one of the gear tooth flanks is anticipated. The intent of this procedure is to specify a method for extended testing with minimal physical supervision (unmanned), that still provides for proper control and inspection. The definition for a macropitting failure for use in this protocol follows guidelines specified in AGMA 925-A03 and is defined as damage (macropitting) occurring on a loaded tooth flank that is approximately 1.0-mm in diameter.

Full load test torque is 6,300 lb-in, however this torque can vary as needed to achieve a pitting failure in a reasonable test time. Test motor speed is 2,250 rpm. Variations in torque and speed are allowed as required for proper operation; however, all changes must be approved by Genesis Partners personnel.

The point of reference for determining the direction of rotation of the test is from the fan end of the motor. This could be considered the “back” because it is not the end with the motor shaft, but this terminology has a different definition in the electric motor world. Figure A.16 shows the point of reference for determining direction of rotation.

Operation of this 3-Hole Test allows two different mesh conditions (a reduction gear set and an increaser gear set) in each gearbox (involute and Convoloid). Therefore, four gear meshes were tested at a time. Additionally, subsequent testing can be conducted by reversing the direction of rotation of the motor. Thus, sufficiently descriptive and unique annotations are required for each component as well as *each tooth flank*. Referencing Figure A.16, gearing component designations could follow the descriptions provided below.

Involute Gearbox		Convoloid Gearbox	
1	Involute Pinion 1	5	Convoloid Pinion 1
2	Involute Gear 1	6	Convoloid Gear 1
3	Involute Pinion 2	7	Convoloid Pinion 2
4	Involute Gear 2	8	Convoloid Gear 2

Gear sets 5 to 6 and 3 to 4 act as increasers and gear sets 1 to 2 and 7 to 8 act as reducers. (For shakedown, see "Micropitting Test Protocol.")

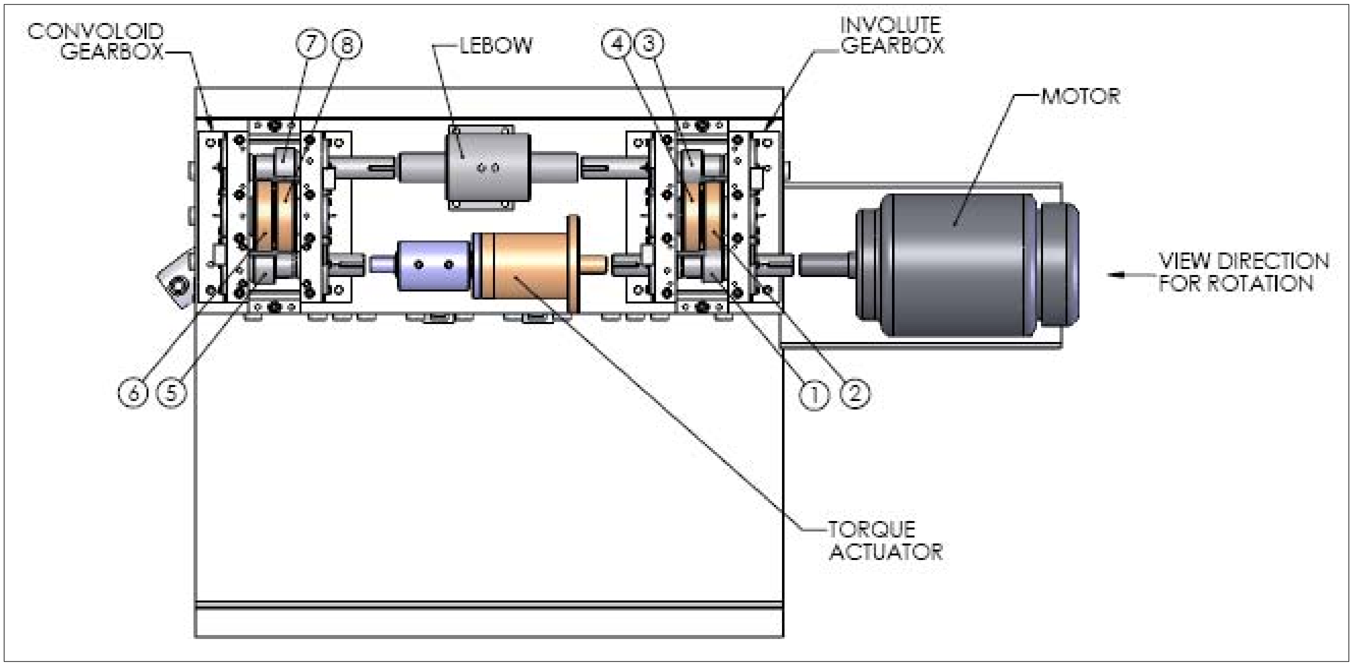


Figure A.16. 3-Hole Test fixture (top view)

Extended Comparison Test Procedure

1. Gear teeth (including each flank) should be uniquely labeled prior to assembly of the gearboxes.
2. Ensure that the circulating and lubricant pumps are operating.
3. Take one 3-oz oil sample from each oil reservoir and perform a preliminary patch test. Retain the samples for comparison purposes.
4. Energize drive motor and slowly bring it up to operating speed (2,250 rpm).
5. Verify that all components are running smoothly and with minimal vibration.
6. Bring the load up to 100% of test torque in 25% increments (1,575 lb-in, 3,150 lb-in, 4,725 lb-in, & 6,300 lb-in). Confirm that instrumentation operates properly. Continuously monitor particulates; if they exceed -/18/13, then shut the system down and determine the reason. If particles are detected in the chip detector, then the test should be stopped. All components should be inspected and the cause for stoppage investigated. If no gear or bearing damage is found that would compromise the test, ensure that the circulated oil cleanliness is -/15/12 and restart the test.
7. An inspection should occur after approximately 165 hours (1 week) to verify tooth contact and wear (or lack thereof). The inspection should involve removing the coverplate and photographing the contact flanks of each gear/pinion. If any wear is evident, a metric (millimeter increments) scale

should be included with the pictures. A 3-oz oil sample should be taken from each oil reservoir and retained for 2 months. Conduct a patch test and archive the results.

8. Subsequent inspections should occur approximately every 335 hours (2 weeks) following the above procedure. When wear appears on a tooth flank, however, the inspection interval should be reduced.
9. Once a member of one of the gear sets exhibits a macropitting failure, the test should be halted. An oil sample (with patch) should be taken and stored with the gear set. Notes for that specific gear set (loads, increaser/reducer, number of hours to failure) should be recorded. After inspection of all gearing, a decision on how to proceed must be made. In most instances two options exist (assuming a non-catastrophic tooth failure).
   A. The failed gear set should be removed from the test gearbox and be replaced by a new or undamaged gear set. Label and archive the gear set removed. To prevent variations due to possible bearing damage, new bearings should be used with the new gear set. Other wear items should be replaced as necessary. The remaining components in the test are assumed to still be in test, therefore great care should be exercised to prevent damage to these components. Additional testing on the removed component is possible (changing the motor direction of rotation, thus loading the opposite tooth flank), therefore removed components must be lubricated and stored carefully.
   B. Reverse the motor rotation to conduct testing on the opposite tooth flanks. Prior to restarting the test fixture, documentation should refer to the load and number of hours sustained by the previously loaded flank. If no wear is visible after subsequent testing, damaged components can be removed and be replaced with those with no visible wear. In this manner, additional cycles can be applied to various test components until tooth failure occurs.
10. After the gearbox has been reassembled, circulate lubricant while the gearbox is at rest to remove any contaminants introduced during assembly. When the oil cleanliness reaches -/15/12, restart the procedure at step 3.

Note that the bulk oil temperature should not exceed 155° F—measured at inlet to gearbox—at any time. Nor should any bearing temperature exceed 190° F (an alarm will sound). If either of these conditions exists, stop the test run and contact the instrumentation engineer.

Any change in sound, vibration, oil cleanliness, or any other abnormal operation is acceptable cause to shut down the operation of the load cell and motor (lubricant pumps should remain on until all motion is stopped). A thorough inspection of all components should be conducted before continuing the test. All variations should be noted in the test notebook. If there is a shutdown due to mechanical instrumentation or lubrication problems, then follow step 9. A preliminary list of computer-monitored conditions follows, however there can be additions or subtractions as needed. Each of these conditions has warning levels and shutdown levels enabling the test to run unattended.

- Loss of oil pressure (pumps failed)
- Oil differential pressure is too high across filters (filters clogged)
- Chip detector goes low (ferrous material is in the oil)
- Bearing temperatures are greater than 190 deg F
- Oil flow is too low

- Oil pressure is too high at the manifold (manifold orifices clogged)
- Vibration is over limits in any of three planes
- Particle counters indicate high particulate count
- Torque loss occurs
- Power loss to test container
- Bulk oil temperature is too great
- RPM of drive system is out of test limits
- Visual information from the webcam that indicates shutdown is needed

Start-Up Procedure

This discussion might duplicate some steps in the procedure described above, but it is followed when restarting the test equipment.

1. Ensure that the oil temperature is 150°F.
2. Activate the oil circulation pumps. Circulate the lubricant until the oil cleanliness reaches -/15/12. With the gearbox coverplate removed, oil should emerge from every bearing and the gear mesh nozzles. A lack of oil flow indicates a clogged orifice (lubricant pressure also can be elevated).
3. Energize the motor (motor operation is not possible if the oil flow rate is too low.)
4. Once the motor is operating at the desired speed, gradually apply the load to the system.
5. After loading has reached the test parameter, confirm that operation of all components is smooth and consistent. Log the test time.

Shutdown Procedure

Smoothly remove the load from the system. In the case of a catastrophic failure, immediate shutdown might be required.

1. Once the load is removed, log the test time.
2. Smoothly reduce motor speed until the motor stops.
3. Stop main lubrication pump (which feeds the distribution manifold). During a normal inspection, the circulating pumps can remain in operation at the user's discretion.
4. For extended periods of inactivity, shut down the lubricant heaters and circulating pumps.

3-Hole Test—Detail of Test

The discussion of the Micon-108 test noted that a four-square type test required the test gearboxes (or components) to have the same ratio. This condition created limitations for either the Convoloid or the involute gear sets, therefore the "best" involute could not be tested against the "best" Convoloid. Further, as center distance increases for gearing, the torque requirements increase dramatically. The chosen center distance size for the 3-Hole Test was 4.0 in to allow reasonable testing size without requiring too much torque. Related components (couplings, torque actuators, torque transducers), however, are difficult to fit into a small space. With this in mind, the 3-Hole Test gearbox and test were designed (Figure A.17).

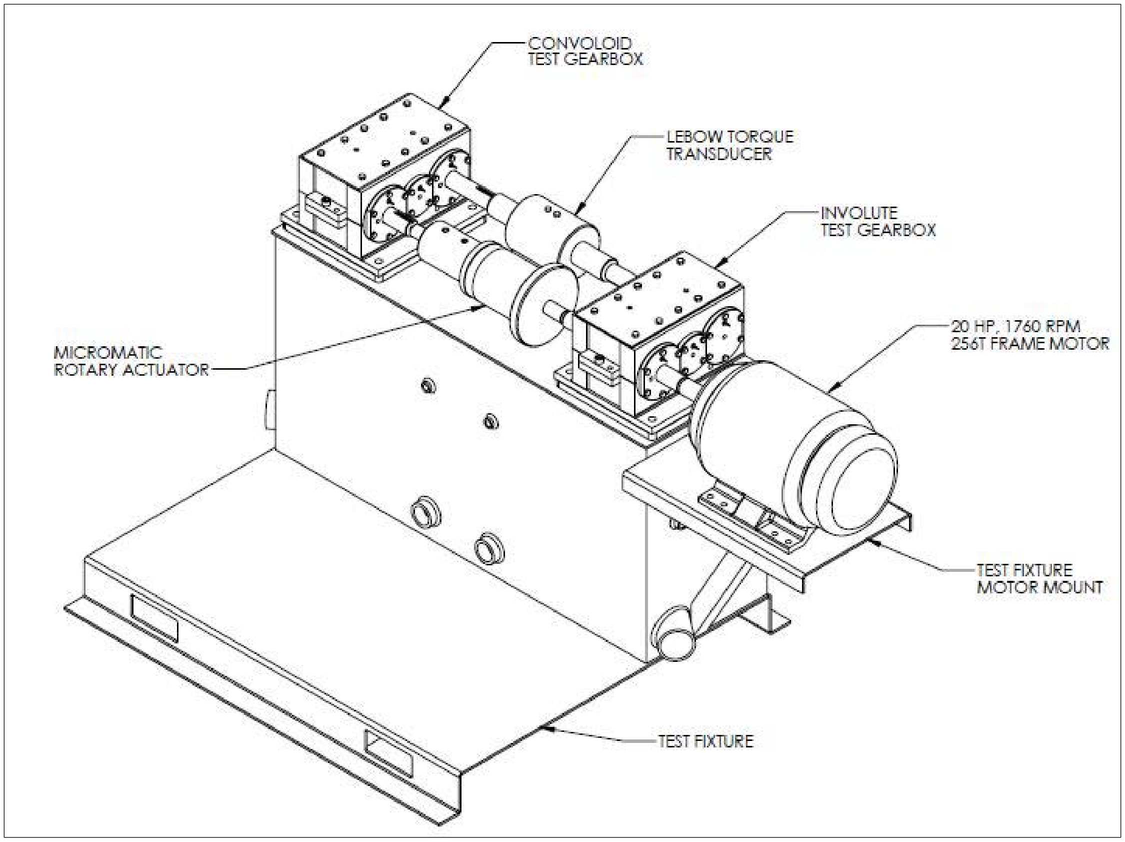


Figure A.17. Basic layout of the 3-Hole Test

Having two gear sets within the test gearbox means that the distance between the shaft extensions is twice the operational center distance, thus allowing much more room for other components. Also, the final ratio between the shaft extensions for both Convoloid and involute gearboxes is 1:1. As long as the gearing is designed to fit into the gear housing, it can be optimized for the test loads. For this test the individual gear ratios were approximately 2.2:1. The configuration of the test fixture is discussed in detail above. Some testing of the gearing, however, occurred during this time.

After assembling the Convoloid gearing into gearbox 2, a minimal torque was applied as the motor was energized to a low speed. Subsequent inspection of the gear teeth revealed acceptable light-load wear patterns on the tooth flanks (the flanks had been painted with machinist dye). Figure A.18 and Figure A.19 show representative no-load contact on the involute and Convoloid gearing. After the lubrication and shaft failure in the Convoloid gearbox, involute gearing was assembled into both gearboxes. All subsequent testing occurred using this configuration.

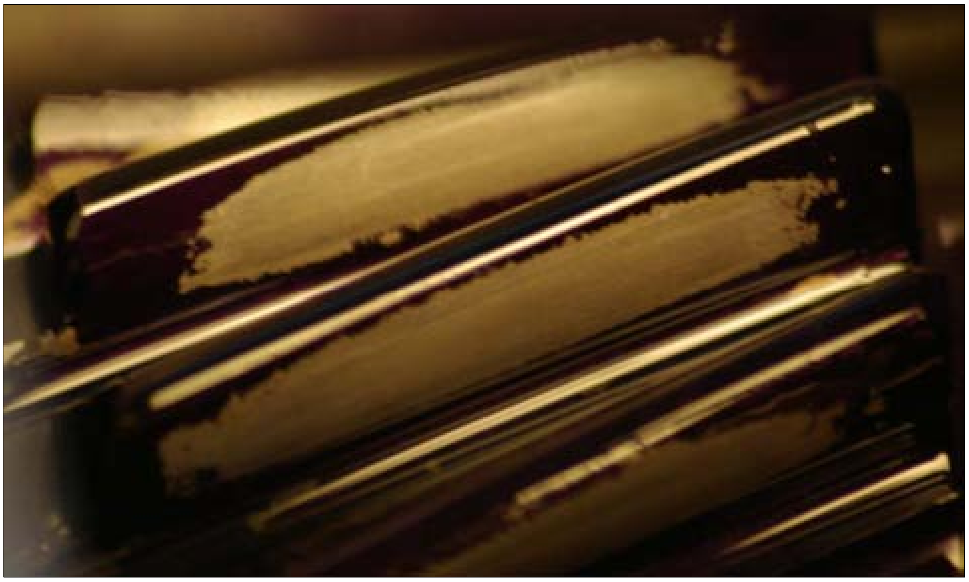


Figure A.18. Representative involute no-load contact

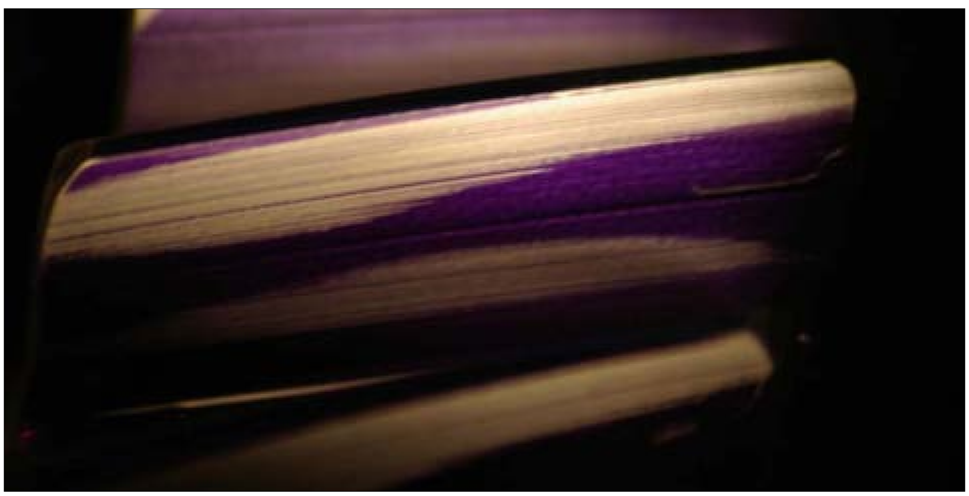


Figure A.19. Convoloid no-load contact

The original test load of 6,300 lb-in yielded a calculated life (L_1) for the involute gear set of 42 hours. Although this time frame (less than 2 days of 24-hour operation) sounds sufficient, it only represents the fact that out of 100 gear sets only 1 possibly would fail in that time. When looking at the average life (L_{50}), the expected life is in excess of 24,000 hours (nearly 3 years of test time). Assuming that the gearing and materials are of good quality, the average test cycle is prohibitive. Also, at this load the projected average life for the Convoloid gearing is more than 700,000 hours.

Analysis of potential test loads that would shorten the anticipated time-to-failure for surface fatigue yet keep the chance of a bending failure remote initially resulted in a test load of 7,000 lb-in. At this load, the L_{50} life for the involute gear set was 9,400 hours—more than 1 year. The L_1 life was 16.4 hours. Further refining of the analysis resulted in a test load of 7,300 lb-in which gave an L_{50} life of 6,570 hours (L_1 = 11 hours). The resulting involute bending life is 45,000 hours. For comparison, at the same load conditions, the Convoloid gearing has an L_{50} contact fatigue life of 190,000 hours and a bending life of more than 300 E6 hours.

It is important to remember that the bending life must be much greater than the surface durability life due to the operation of the test fixture. Once a surface failure is observed and documented, the loading is reversed (with the same gear set) thus causing contact on the previously unloaded flank. The test is operated until a surface failure occurs. The test load was chosen assuming the bending L_{10} life had to be greater than twice the surface durability life L_{50} life. Also, bearing lives and shaft strength considerations must be kept in mind. At the 7,300 lb-in load, the involute gear set bending life probability (at twice the surface durability life) is ~ 6%. The limiting bearing is the intermediate bearing (recently reduced in size to accommodate a larger shaft diameter); however, the shortened catalog life should be offset by the ready supply of cool-clean lubricant. If necessary the bearings can be replaced, allowing the test to continue.

Upon installation, the no-load contact pattern for the involute gearing proved to be ideal (Figure A.20). Also shown in Figure A.20 is the lubricant supply nozzle for one side of the gear mesh. A similar nozzle is positioned below the mesh, thereby assuring oil application at the inlet and outlet of all gear meshes. Proper oil pressure (25 psi) is necessary for these nozzles. By the end of the month, the 3-Hole Test fixture was nearly operational with good full-load (7,000 lb-in) contact (Figure A.20). Operational speed is 2,200 rpm.

Figure A.20. No-load contact pattern—involute gear; lube nozzle

Figure A.21. Example of the full load (7,000 lb-in) contact—involute gears

Initial testing for shakedown purposes used a load of 7,000 in-lb which is less than the final approved amount of 7,300 in-lb. Early in the test time, however, micropitting was observed in all the loaded flanks with more severe wear appearing on the reducer gear sets as compared with the increaser gear sets (*see* Figure A.23, Figure A.24). This wear was observed under the existing load, therefore it was decided that the test load would remain at the lower level. The operating oil temperature was 155°F—160°F.

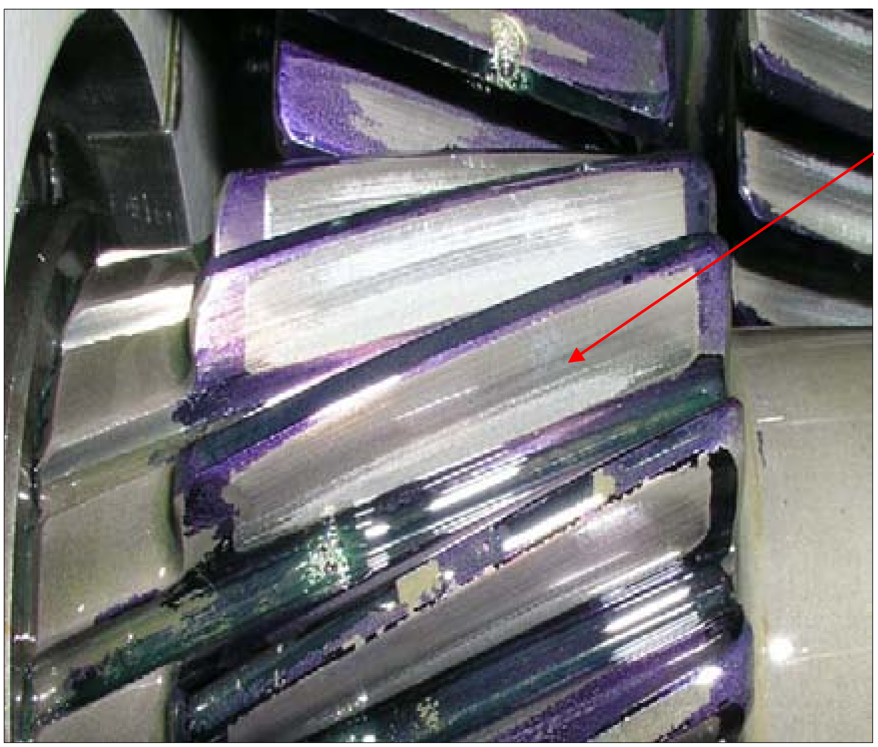


Figure A.22. Involute pinion 1—reducer operation (pinion driving gear)—showing extensive micropitting on both the addendum and dedendum

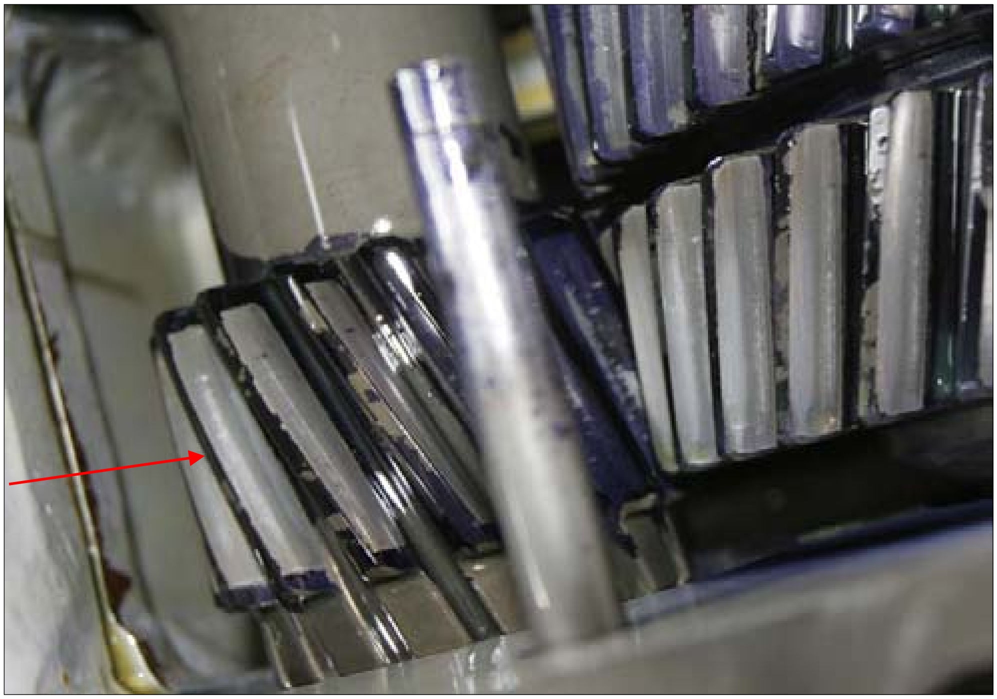


Figure A.23. Involute pinion 2—increaser operation (gear drives pinion)—shows a smaller amount of micropitting on the pinion (arrow) as well as on the mating gear

Appearance of the micropitting in the involute gear sets caused some concern. Analysis of the load conditions, lubricant, and lubricant temperature determined that the test was operating in lubrication regime between I and II. At this condition there is expected to be some metal-to-metal contact because the lubricant film thickness is not sufficient to fully separate the gear flank surfaces. The temperature for the test was specified to be 160°F. Although the film thickness was judged to be much too thin for proper operation, it was thought that some knowledge could be gained from this test. Subsequent tests will reduce the temperature to 140°F, raising the lubrication regime to the region between II and III—thereby nearly having a film thickness great enough to fully separate the metal surfaces.

Early in testing, a failure was reported in *all* of the test gear sets. The test time was 162 hours at 7,000 lb-in torque, 2,200 rpm, and with an oil temperature of 155–160°F. This comprises approximately 2.1 times 10^7 cycles. Although this time is greater than the calculated L_1 life of 16 hours, it is nowhere near the average life of 9,440 hours. The sudden appearance of surface failures in all gearing in both gearboxes was questioned immediately. Initially, the cause was thought to be an abrupt fatigue failure, but the appearance of the damaged tooth flanks is not consistent with a classic fatigue case (*see* Figure A.24).

Figure A.24. Involute (1) reducer pinion surface wear (Test 2)

Instead of classic macropits where material is removed due to subsurface microcracks caused by fatigue, there is distinct surface distress that appears to be scuffing (where the contacting tooth flanks weld together, then tear away). Investigation into the circumstances surrounding the failure revealed that during operation, vibrations during installation of the air conditioner support brackets caused one of the fuses in the main circuit breaker to come loose. This initiated an abrupt shutdown of the entire system in which the motor stopped while the system was under full load. All other equipment also lost power (e.g., computers, lube pumps).

After the fuse was replaced (and blown fuses in the motor breaker were replaced), the test was restarted. After approximately 10 hours of operation, the system was shut down due to high vibration and excessive noise. Subsequent inspection of the gearing revealed damage to all gear flanks. It is thought that some damage to the gear teeth occurred during the abrupt shutdown and this caused premature wear. Once the surfaces were distressed (and/or distorted), additional wear occurred rapidly.

The wear occurred very rapidly as inspections less than 10 hours prior to the loss of power revealed no metal particles on the chip detectors and a reduction in the advance of micropitting on the tooth surfaces. A normal fatigue situation reveals gradual increases in macropits, beginning on the pinion surfaces because these elements see more cycles than the mating gears. This information added to the hypothesis that some tooth wear occurred during the shutdown. After the unexpected shutdown and before the final hours of operation there were no additional inspections of the gear teeth.

Examination of test data was not possible because the test computer was not yet configured to gather data or to set off alarms for out-of-range conditions. Manual checks of the chip and vibration sensors, however, revealed that a shutdown condition existed—which would have been noticed if the computer monitoring had been operational (*see* Figure A.25). Evidently, modifications to the program to allow for emergency shutdown and notifications removed the previous settings, causing test personnel to operate in a near-manual mode. All data was manually gathered and recorded.

Figure A.25. Gearbox 1 chip detector after gearing failure; note material crossing the gap

In addition to updating the computer control for failsafe operation and notifications, controls were added for torque adjustment. It originally was thought that the torque setting would be fixed for the duration of the test. Temperature changes and other unknown variables, however, have required the hydraulic pressure for the torque actuator to be adjusted periodically. Without changes the torque values appear to vary by nearly 10%. Before the automated system was operational, the personnel who monitor the test performed this touchy function—which requires a "dwell" time after each modification.

Operation of the test fixture is necessary for the computer configuration, therefore it was decided that testing would continue with load on the opposite gear tooth flanks. With the previous test, the depth of the damage on the gear tooth flanks was deemed to not be detrimental to the bending strength of the gear teeth. The calculated L_1 bending life at 7,000 lb-in is greater than 150,000 hours. Prior to operation of the fixture, the oil was completely drained and filtered twice through a 3-micron filter and the reservoir was wiped clean prior to being refilled. The chip detectors also were cleaned to prevent a false positive. Finally, the fixture lube system was cycled at operating temperature for more than 1 hour.

As a modification of the test procedure, any time a "non-normal" situation occurs with the test equipment, all gearing as well as the chip detectors are to be inspected for wear and particles. A subsequent inspection also should be conducted after 5 to 10 hours, to ensure that no unexpected wear begins on the gear teeth. If there is damage then the test on those tooth flanks could be considered void. Additionally, because this test is to determine the surface fatigue life of the involute and Convoloid gearing, an appropriate lubricant film thickness is necessary. The original oil temperature of 160°F was chosen to approximately duplicate the operating conditions inside a wind turbine gearbox. This temperature, however, does not allow the lubricant (ISO 320 viscosity) to develop enough of a film thickness to separate the metal surfaces of the gear teeth (an oil regime between I and II). Therefore, subsequent testing occurred with an oil temperature of 140°F that should allow operation near oil Regime III. Operation of the test continued with 100% supervision and manual data gathering with a test load of 7,000 lb-in torque and 2,200 rpm motor speed.

After 86 hours of operation (Test 3) a significant amount of vibration was noted, prompting shutdown of the test fixture. Upon inspection, pinion 1 in gearbox 2 was found to have significant macropitting (see Figure A.26 through Figure A.28). The origin of the pitting appears to fall at the pitch line of the tooth flank, which is consistent with the area of highest calculated stresses for involute gearing. This pinion was operated as a reducer where the pinion drives the gear. Note that there is frosting (micropitting) in the dedendum (near the tooth root) of the tooth flank. Although, in some cases this phenomenon has been thought to precipitate a macropitting failure, at the pitch line where the macropitting originated no micropitting is present. Therefore, the macropitting can be considered a classic fatigue failure of the type expected in this test. It is important to note that only 10 hours of operation occurred between inspections. There was no hint of damage prior to the high vibration level (manually monitored at that time) that prompted cessation of the test.

Figure A.26. Massive macropitting on pinion tooth

Figure A.27. Additional pitting flank (Test 3, gearbox 2, pinion 1)

Figure A.28. Further pitting and opposite flank scuffing damage from Test 2

The definition of macropitting failure for this test has been set as 1.5% of the operational tooth flank. For the involute gearing this translates into a circular pit with a diameter of approximately 0.10 in. The Convoloid gearing has a larger contact area, thus the approximate circular pit diameter is ~0.12 in. In this failure, the amount of pitting greatly exceeded the definition. It is expected that macropitting will progress from small pits to larger ones, and that with the increase in pitted area the measured vibrations will increase. This increase in vibration levels will be used for automatic shutdown of the test rig, however, at the time of Test 2, the test parameters and warnings had not yet been set up.

Upon close inspection of the mating gear, no detrimental wear was observed. Therefore, once the failed pinion was replaced, testing continued (Test 4) with loading on the same flanks. As noted, the number of load cycles for all parts is logged so that, upon failure of a given part, the total number of load cycles can be charted. Comparisons between the data scatter for the involute gears versus the Convoloid gears should show a distinct increase in component life for the Convoloid gearing.

Control of the test environment is necessary to reduce as many outside variables as possible, and this was finally accomplished with the implementation of the computer-controlled monitoring and control system. Up to this point, data values had to be inspected individually using the computer, with the values being manually written and no warning or shutdown system in operation. Now the software has been completed to allow full monitoring and data collection as well as torque control and alarms (lights and e-mail notifications).

In the past, torque control has been a manual operation with the operator watching variation in the torque readings and adjusting the torque potentiometer accordingly. The computer now controls the torque and holds values much closer to desired values (less than ±.01%) than was possible before. The torque being applied to the test remains at 7,000 lb-in. The test speed is 2,200 rpm.

Test 4 was terminated after 18 hours, due to another surface fatigue failure. Figure A.29 and Figure A.30 show the failure—again, a pinion operated as a speed reducer, this time in gearbox 1. Here the surface damage was found during a routine inspection. The amount of damage is indicative of the definition of failure. The total test time for this pinion was 104 hours, or 13.7 million cycles.

It is important to note the extensive micropitting on the tooth flanks. For these parts, it occurs almost exclusively in the dedendum of the gear tooth (below the pitch diameter). There has been much discussion in the gear industry concerning whether the type of action (speed increaser versus speed reducer) affects the appearance of micropitting, or whether it is entirely a result of load, lubrication, and surface finish. During testing, it appears that there is a difference due to the type of action. Both of the failed pinions are loaded as speed reducers (the pinion drives the gear), and there was extensive micropitting in the dedendum of the tooth flanks. The other pinion in the same gearbox with the same number of load cycles as the failed pinion in Test 4, yet loaded as a speed increaser (Figure A.31) shows much less wear on the active flank.

Within the gear industry there is much discussion on the effects of gear geometry, how it is loaded (increaser versus reducer), and wear potential. A preliminary analysis of the probability for wear on the involute gear set (first with the pinion driving the gear, and second with the gear driving the pinion) revealed identical specific film thicknesses and similar low probabilities for wear. Outside of the edges (tip/root) of tooth contact, the calculated film thicknesses are the same for both analyses. The film thickness is a function of the relative curvatures between contacting surfaces, therefore the direction of rotation does not appear to have an affect. Additional study into this area is required before definitive conclusions can be drawn.

Figure A.29. Failed involute pinion from Test 4

Figure A.30. Pinion loaded as a speed reducer

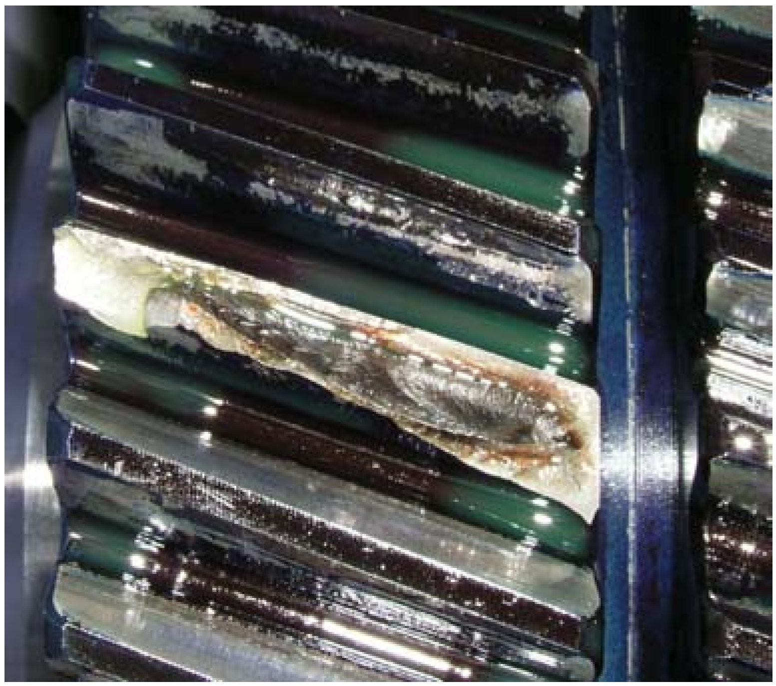
Figure A.31. The other pinion in the same gearbox loaded as a speed increaser showing minimal wear

It is thought that micropitting can precipitate macropitting, therefore the presence of this wear is important. Contact loads on the surface of a gear tooth flank are directly related to the relative curvature between that flank and the mating flank. Standard involute gears have a convex shape; therefore the contact patch between the parts is relatively narrow. Further, the convex-convex contact of the gear teeth acts to “pump out” lubricating oil, reducing the effective film thickness. In contrast, mating Convoloid gear teeth have a convex-concave contact that greatly reduces the relative curvature, creating a wider contact patch under load which maintains a larger amount of oil between the surfaces. This increase in oil-film thickness is expected to reduce or eliminate micropitting in Convoloid gearing. Further, the Convoloid tooth form has no contact in the high-stress area of the pitch diameter. The design method allows Convoloid teeth to have a consistent contact stress across the entire working flank, which is in contrast to the varying stresses in involute gear teeth.

Another aspect that has generated interest is determining if the life of a gear is affected by whether the gear component is the driving element (pinion for a speed reducer and gear for a speed increaser) or the driven element (gear for a speed reducer and pinion for a speed increaser). To this point, all failures have been of the driver element (pinions loaded as an increaser), and early, small macropitting is visible on a driving gear loaded as an increaser.

An unexpected failure was encountered (Test 5) involving a tooth breakage in one of the gear teeth (Figure A.32). This component originally had been loaded on the opposite flank prior to the scuffing failure of Test 2, therefore it was thought that the tooth had failed in bending. Further inspection revealed that the hardened outer case had separated from the softer core, causing the tooth to break. This failure—although not unknown—is very rare and is not included in gearing-life calculation procedures. The gear had operated for 163 hours in the current load direction and 174 hours in the opposite direction (a total of 20.3 million cycles). This time frame represents only a 2.2% probability of failure in bending. A detailed inspection of the hardness spectrum and metallurgy was conducted by Northwest Laboratories, Inc. (Seattle, WA) to determine if there was material or heat treatment weaknesses in the gear. One theory was that the case depth for this part could be less than desirable for the load being applied.

According to ANSI/AGMA2001-D04, there are three methods that can be used to determine the amount of effective case depth for carburized gearing. Two are specified by means equations, in Figure 13 in Section 16 of ANSI/AGMA2001-D04, that are related to the tooth size (diametral pitch) only and equate to values for the effective case depth, $h_{e\ min}$, of 0.0225 in and 0.0297in for normal and heavy case values respectively. It is recommended, however, that the operating effective case depth should be determined by using the actual contact stresses expected (equation 43 of the standard), resulting in a value of 0.0262 in. Inspection of the broken part showed that the actual case depth was 0.0339 in—well within effective case depth specifications. Also, no metallurgical abnormalities were observed during a microscopic inspection of sections of the gear tooth. Therefore this broken gear remains a "random event"–type failure and is not included in the failure summaries.



Figures A.32. Broken involute gear tooth from Test 5; failure was due to case/core separation

Testing continued after replacement of the damaged gear with the last part in inventory. Some adjustments of the computer-control system were necessary to assure a smooth application of torque and to help isolate the three axes of vibration. Upon startup, the computer control smoothly ramps up to full test load over a period of 30 minutes. This assures full lubrication and no torque spikes that inadvertently could damage the gearing. Also, specific set-points were programmed for the accelerometers in each axis of the vibration. During previous testing (the Micon 108 test), it was found that vibrations in the minor planes showed early wear much sooner and more accurately than the high-vibration planes. The vibration in these planes varies after each part change (due to the small magnitude), therefore a different alarm point must be set for each test. After the nominal operating vibrations are determined, the alarm point for each vibration axis is set (expected to be 50% higher than the nominal). In this manner, because there is no all-encompassing set value, the test can be more sensitive to early gear wear. It is hoped that a correlation between the system vibrations and early gear flank wear amounts can be developed.

After a short time of testing, the third macropitting failure in the involute gearing occurred. Once again, the failed member was a driving pinion—this time with 82 hours of operation. This part had replaced an earlier failed pinion. To date, the only macropitting failures have been with *driving* pinions with 86, 104, and 82 operation hours. The other involute pinions (being *driven* by the gear) have survived for more than

168 hours with minimal wear visible. A small macropit is visible on a driver gear, however this failure has not reached the test definition of failure. All remaining gear flanks show minimal wear.

A summary of the relevant tests is shown in Figure A.33. The graph excludes the broken gear (Test 5) and early setup cycles (Test 1) where there was no failure in the involute gearing. A previous failure (Test 2, which was due to a loss of lubricant), is included to show that these parts had exceeded the current failure lives. Therefore, it is thought that subsequent tests should indicate longer lives than what these three tests reveal.

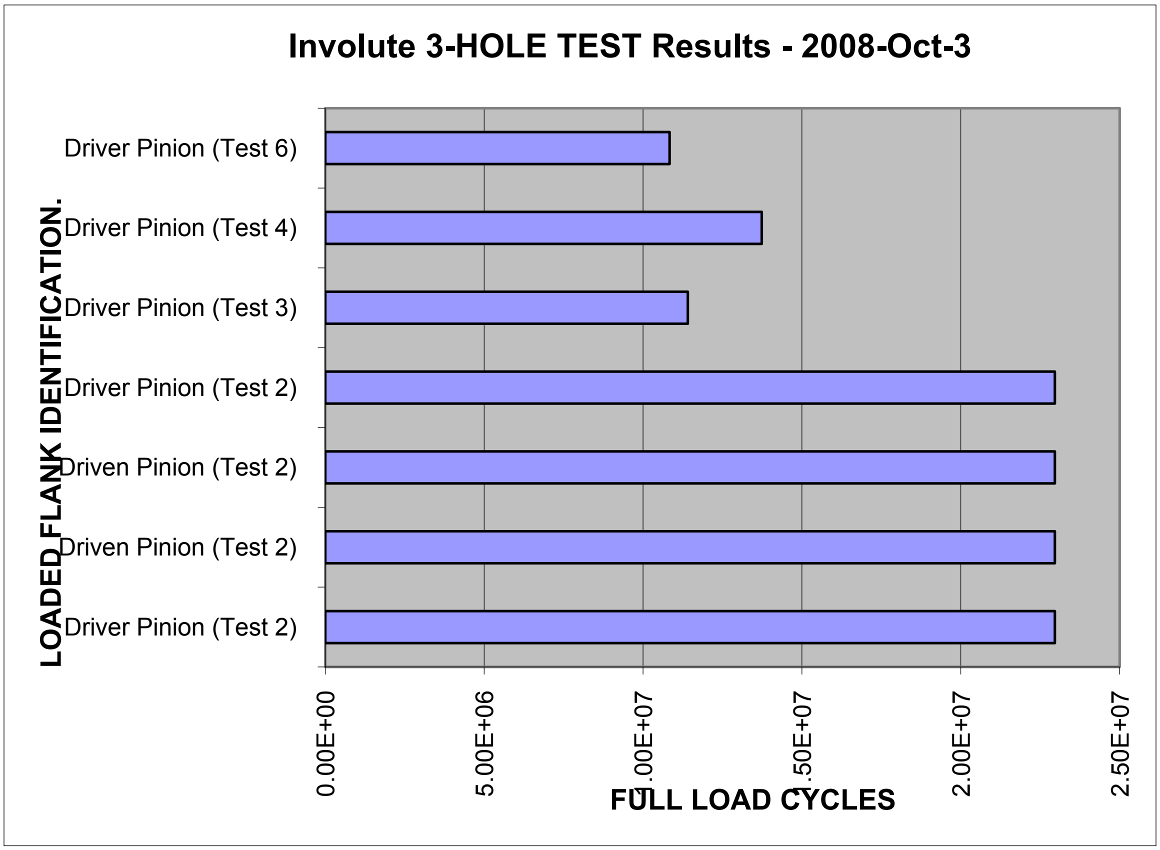


Figure A.33. Summary of involute gear failures in the 3-Hole Test

Testing currently has been suspended due to lack of funding. There are eight sets of involute gearing and one set of Convoloid gearing available for testing. An additional 6 sets of Convoloid gearing await confirmation of the micropitting protocol before having a final tooth grind. The anticipated time frame for testing, based upon gearing components currently in process, is expected to depend primarily on the number involute pinions available because they have the shortest expected lifespan. Assuming both tooth flanks of each component are able to be loaded in turn and a component lifetime of approximately twice the current failure rate, the existing parts being manufactured allow for approximately nine months of operation. During this time, approximately 24 data points should be gained for the involute gearing and approximately 10 data points should be gained for the Convoloid gearing. If testing follows fatigue theory, then the time frame will be greatly lengthened.

Unless additional involute pinions and gears are made later in testing, it is expected that some of the Convoloid parts will not be used by the time the last involute part has failed. To this point, all testing has coincided with configuration of the test fixture and only has involved involute gearing after the original Convoloid gearing was damaged due to a lubricant pump failure. Data gathered from the involute gear failures is expected to act as the control for the Convoloid testing. These data points are expected to confirm current involute gear theory.

Implementation of the full test—involute versus Convoloid—is expected after the Convoloid gearing all have been re-ground to achieve an appropriate no-load contact. Only one set of Convoloid gearing previously has been ground to act as a qualification of the grinding method and to ensure that premature micropitting wear does not appear on the Convoloid profile (as was experienced in the Micon 108 test).

Estimated Test Time—Continuation of 3-Hole Test

3-Hole Test
Estimated Time of Test with Existing Components
as of 9/8/2008

As preparation for full testing of the 3-Hole Test project, it is necessary to estimate how long testing can continue with the parts currently in inventory or in the manufacturing process. At the time of this writing, the parts available for this test (both finished and in process) are as follows.

- Convoloid Pinion 1—1 complete, 1 possible scrap at Bradford Inst., 1 in process at Carnes-Miller, 6 in process at The Gear Works (TGW)
- Convoloid Pinion 2—1 complete, 1 possible scrap at Bradford Inst., 1 in process at Carnes-Miller, 6 in process at TGW
- Convoloid Gear—2 complete, 4 in process at Carnes-Miller, 7 in process at TGW
- Involute Pinion 1—8 in process at TGW
- Involute Pinion 2—8 in process at TGW
- Involute Gear—17 in process at TGW

The anticipated test time is dependent upon the length of time necessary for components to fail. It is expected that components will fail with a surface fatigue (macropitting) mode, however experience has shown that unanticipated failures are possible. Further, a minimum amount of time is required for changing-out failed components and reassembling the fixture with new parts. The estimation of life for test components is based upon both current testing experience as well as fatigue theory. There have been failures earlier than theory predicts, therefore the estimated life must reflect that situation. Due to theory, however, it is assumed that the average life of test components will be longer than in previous tests. If components last much longer—to their calculated average life, for example—then testing will require additional time.

Due to the reduced number of cycles, the gears have a longer life (in hours) than the pinions. Therefore the pinion life becomes the limiting factor. Also, because there is a much greater bending life, both flanks of the pinions are able to be loaded with minimal risk of having broken teeth. Lastly, as one component suffers a surface failure, early testing has revealed that no adverse damage occurs to the mating flank. This is not assumed, however, and a thorough inspection is conducted after every failure. If no adverse damage occurs to the mating flank, then only the failed component must be replaced—allowing for additional test points.

As components fail they must be replaced. Provisions within the test fixture allow the torque to be reversed without modifying the gearboxes or the fixture. Therefore, in many cases, two tests can be conducted before it is necessary to shut down the fixture and replace parts. From previous testing, it is estimated that the changeover requires 37.5 man-hours. Replacing damaged components requires a number of steps.

1. Removal of the gearbox from the test fixture
2. Disassembly of the affected portion of the gearbox
3. Cleaning of components
4. Reassembly of the gearbox with replacement of the damaged component(s)
5. Installation of the gearbox onto the test fixture
6. Balancing couplings to reduce vibrations
7. Heat test of fixture oil and thoroughly circulate it to ensure clean lubricant
8. Start the new test

Following is a summary of the anticipated length of the test with current components, based upon an anticipated failure rate. Variations in these failure rates are expected.

Involute

The average life of all failed components (including those that lasted until the lubrication incident) was 148 hours, with the shortest being 86 hours. It is thought that the times of these failures are on the low end of the spectrum because all the parts of Test 2 lasted until a power outage caused loss of lubricant and resulted in a scuffing failure (164 hours) before loading the opposite flank. Therefore, a value of 220 hr is used as the anticipated life. According to the life calculations, this represents an $L_{10.5}$ life (10.5% probability of failure). With the average surface contact life (L_{50}) at 6,460 hours, it would be expected that testing should progress well beyond the $L_{10.5}$ level. That has not been the case in early testing, however. The corresponding bending life (with double the L_{50} contact cycles—because both flanks will be loaded) results in an $L_{4.6}$ life (4.6% probability of failure). Based upon the life of the pinions, the approximate test time becomes the following.

Test Time = 220 hours * 8 pinions * 2 flanks = 3,520 hours

The amount of time required for changing-out failed components is reduced somewhat due to the ability to reverse loading in the test. Unexpected situations could arise, however, causing some part changes. Based on the available components, the approximate test time required for replacing failed components (assuming an active test time of 3,520 hours) becomes the following.

16 total pinions, 1 pinion failure at the end of the test + 8 gear changes + 5 unexpected changes = 30 gearbox changeovers

28 changes * 37.5 hours/changeover = 1,050 hours

Convoloid

Using the same failure definition as for the involute gearing, the life of the Convoloid pinions is 320 hours. The average surface contact life (L_{50}) is calculated to be 190,000 hours. The corresponding bending life (with twice the L_{50} contact cycles) results in an $L_{7.8}$ (7.8% probability of failure). Initially, this seems a higher probability than for the involute, however the Convoloid gears see a number of cycles that is 29 times greater.

With the life of the Convoloid pinions expected to exceed that of the involute pinions and with the usable number of parts being equal, the only addition to the total test time will be the time it takes to change over the Convoloid gearbox. Based upon the life of the pinions, the approximate number of pinions to be used during the involute test time becomes following.

Number of Convoloid pinions = 3,520 hours / 320 hours / 2 flanks = 5.5 ~ 6 pinions

Therefore, potentially 6 of each pinion will be used during the time it takes to fail all 8 of the involute pinions. In that time it is expected 3 gears for each pinion will fail (due to the reduced number of cycles on the gears). This leads to the number of changeovers and thus the amount of time necessary for Convoloid changeovers.

(12 total pinions + 6 gear changes + 4 unexpected changes) = 22 gearbox changeovers
22 changes * 37.5 hours/changeover = 825 hours

Therefore, the total anticipated test time is 3,520 + 1,050 + 825 hours = 5,395 hours. Allowing 20% for contingencies, this leads to a total consumed time of 6,474 hours (approximately 9 months).

Early Test Results

3-Hole Early Test Results
2008 Aug 26

- Test Components: 16 x 35 involute gearing, 7 NDP, 20° NPA, 18.697° HA, 1.450 in FW, 4.00 in CD
- Test Conditions:
- Test Load: 7,000 lb-in pinion torque
- Contact Stresses: 218,715 psi calculated per ISO6336 (AGMA Contact Stress Number 232,994 psi)
- Calculated L_1 life: 4.29E7 cycles (325 hours) per ISO6336; 5.28E6 cycles (40 hours) per AGMA: 2001

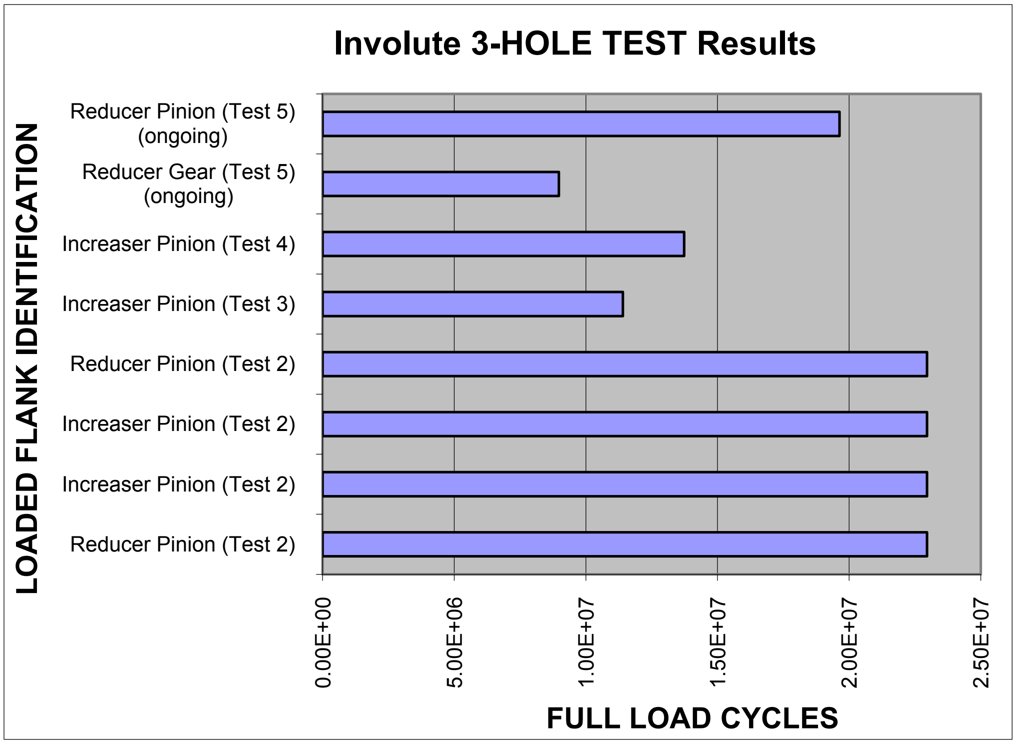


Figure A.34. Involute 3-hole test results

Failure Modes

- Test 2: Severe scuffing due to loss of lubricant
- Test 3: Extensive macropitting on numerous increaser pinion flanks
- Test 4: Macropitting on two increaser pinion flanks
- Test 5: Test ongoing—no damage on pinions, slight pitting on one reducer gear
- All components showing macropitting are the driving member

3-Hole Test Life Analysis

3-Hole Test: Involute stresses at 7,000 lb-in

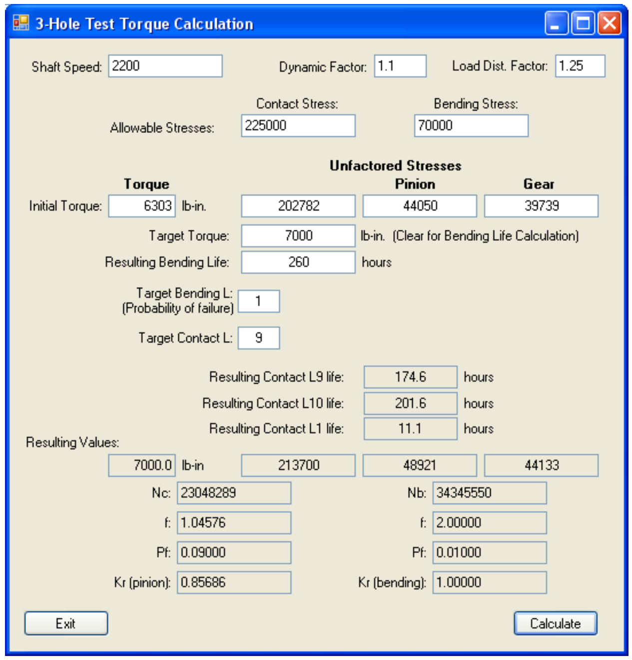


Figure A.35. Contact probability (L9) that matches current failure lives; bending L_1 life is much higher (for 2X contact cycles)

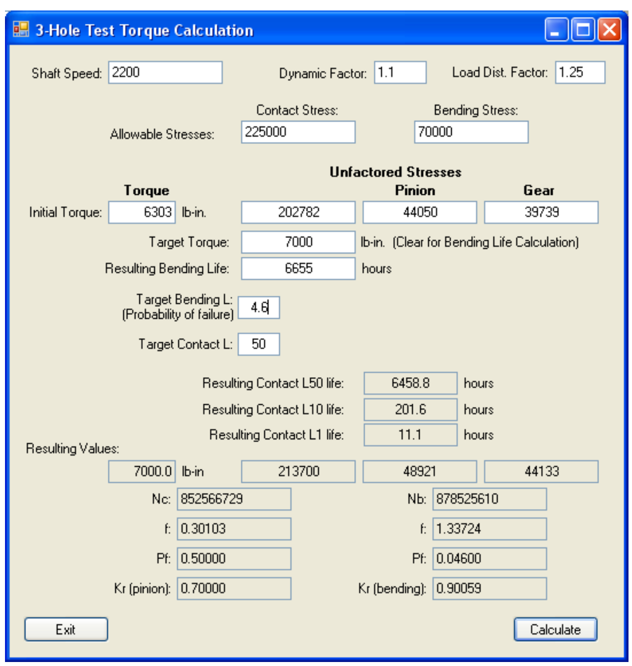


Figure A.36. Calculated L_{50} contact life and the corresponding bending life ($L_{4.6}$)

Low Wind Speed Turbine Baseline Turbine—
Baseline Operating and COE Parameters
Rev 1. 11/09/2004

The following describes the operating conditions and parameters used to establish the 2002 LWST Baseline Turbine cost numbers. These operating conditions and numbers will be used for the validation of TIO projections and for improvement forecasts for LWST subcontractor's whenever appropriate. Most of the detailed component numbers are based upon the WindPACT study work performed by GEC. There have been minor adjustments to match these numbers to those selected by program management for the baseline conditions. Where those conditions vary, they have been noted below. Accompanying this document will be two spreadsheets. One spreadsheet is an AEP calculator that was created by Lee Fingersh (National Wind Technology Center). This calculator allows adjustments for wind speed, weibull shape factor, shear, rotor diameter, hub height, air density, rotor Cp, and various efficiencies. This spreadsheet should be used when analyzing potential improvements in performance and operating conditions. The second spreadsheet is a COE calculator. It uses the baseline cost information developed by GEC under WindPACT with changes as noted below. This spreadsheet should be used when analyzing the impact of changes in component costs on overall COE. It also has cost elements for O&M, Levelized Replacement Cost and Land Lease Cost.

Operating Conditions and Parameters

	WindPACT (1999)	**Baseline Turbine (2002)**
Rotor Diameter	70 m	70 m
Rating	1,500 kW	1,500 kW
Hub Height	65 m	65 m
Operating Wind Class 4	5.8 ms at 10 m	5.8 ms at 10 m
Weibull K Factor	2	2
Base Wind Shear	1/7 (0.143)	1/7 (0.143)
Altitude	0 m	0 m
Air Density	1.225 kg/m^3	1.225 kg/m^3
Rotor Cp (This Cp has been reduced from the WindPACT study to more closely match the projected Cp of a machine of this size in 2002, as based on survey data.)	0.5	0.47
Conversion Efficiency (This conversion efficiency is actually represented as an efficiency surface in the spread sheet, and matches the profile of the WindPACT studies.)	0.95	0.95
Soiling Losses (Soiling Losses have been increased slightly to match with the combined losses used for projecting the 2002 Baseline)	2%	3.5%
Array Losses (The product of the conversion efficiency, soiling losses and array losses is a reduction in AEP of 13%. This matches the losses for the 2002 Baseline before application of the availability.)	5%	5%
Availability (Availability has been increased from the WindPACT 95%, to the 98% used for the 2002 Baseline Turbine projection. This more closely matches reported project numbers for recent installations.)	95%	98%

Cost of Energy Adjustments

The following adjustments have been made to the COE spreadsheet to match the numbers with those derived for the LWST 2002 Baseline Turbine.

Manufacturing Uncertainty
WindPACT (1999) $0
Baseline Turbine (2002) $162,000

A factor called manufacturing uncertainty has been added to the initial capital cost for the turbine; this number has been set at $162,000. It is included as an added markup to make WindPACT capital cost numbers consistent with a wide range of reported costs per kilowatt for large (100 MW and larger) projects reported in the 2002 timeframe. The WindPACT component cost data was developed based on quotes from vendors and cross checks with other industry data, where available, on a component-by-component basis. It is believed, however, that due to less than optimum production conditions, the advent of newer equipment, starts and stops in production due to uncertainties in the Production Tax Credit, exchange rate risks, and less than ideal timing of project starts that manufacturer costs or mark-ups are running above those assumed in WindPACT studies. As the LWST project proceeds this number could be reduced as better cost estimates are obtained.

Tower Costs
WindPACT (1999) $183,828
Baseline Turbine (2002) $101,000

The initial WindPACT tower was based on an 84-m hub height. For the Baseline Turbine, this hub height has been reduced to 65 m, consistent with the majority of recent projects. This has reduced the baseline tower costs to $101,000 from the original estimate of $183,828.

O&M
WindPACT (1999) $0.008/kWh
Baseline Turbine (2002) $0.007/kWh

The WindPACT O&M cost number was fixed at $0.008 per kWh. This was intended to limit O&M being varied during WindPACT studies, because these studies primarily focused on determining the impact of component design changes. For the Baseline Turbine the O&M number has been reduced to $0.007 per kWh (based on an estimate of $30,000 per turbine), to more closely match recent reports. For simplicity, the COE spreadsheet has reduced this calculation to $20 per kW per year. *It is important to note that O&M is a tax-deductible expense.* In the final COE calculation in the spreadsheet, the O&M number is multiplied by 0.6 to take into account the tax-deductible nature of the expense.

Levelized Replacement Cost
WindPACT (1999) $15/kW/turbine
Baseline Turbine (2002) $10.70/kW/turbine

Long-term replacement and overhaul costs from WindPACT were set at $15 per kW per turbine. For the 2002 Baseline Turbine this number was lowered to $10.70 per kW.

Land Cost
WindPACT (1999) $0
Baseline Turbine (2002) $0.00108/kWh/turbine

For the WindPACT study it was decided to include the lease cost of land in the fixed-charge rate, along with several other fixed costs. For the 2002 Baseline, the land-lease cost has been entered as a separate item in the spreadsheet and set at $0.00108 per kWh per turbine.

Fixed Charge Rate
WindPACT (1999) 10.6%
Baseline Turbine (2002) 11.85%

The WindPACT fixed charge rate (FCR) was set at 10.6%. This number was established at the beginning of the WindPACT project in late 1999. For the Low Wind Speed Technology project this number was adjusted to 11.85% to be more in line with data for projects at that time (2002). Additionally, the FCR was updated as a result of efforts to more closely align the pro forma cash-flow spreadsheet methodology with industry practices. The FCR is imputed from a standard case using the cash-flow spreadsheet. The FCR reflects finance charges and cost of money as well as other factors and in reality fluctuates over the years. For purposes of comparing competing technologies, however, it is necessary to freeze this number as has been done for LWST.

Appendix B. Baseline Cost of Energy Comparison Sheet (2002 $)

Baseline COE Comparison Sheet (2002 $)

Baseline Turbines:

- 108 kW—3-Bladed Upwind/Fixed Pitch Stall Controlled (from Wally T. report)
- 750 kW—3-Bladed Upwind/Variable Pitch Controlled (from WindPACT Turbine Rotor Design Study, June 2000 to June 2002)
- 1.5 MW—3-Bladed Upwind/Variable Pitch Controlled (from WindPACT Turbine Rotor Design Study, June 2000 to June 2002)
- 3.0 MW—3-Bladed Upwind/Variable Pitch Controlled (from WindPACT Turbine Rotor Design Study, June 2000 to June 2002)
- 5.0 MW—3-Bladed Upwind/Variable Pitch Controlled (from WindPACT Turbine Rotor Design Study, June 2000 to June 2002)

Rating (kW)	**108**	**750**	**1,500**	**3,000**	**5,000**
	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000
Component					
Rotor	19.0	101.90	247.53	727.93	1484.43
Blades	15.0	64.07	147.79	437.46	905.90
Hub	4.0	21.62	64.19	213.03	429.31
Pitch mechanism & bearings	0.0	16.21	35.55	77.44	149.22
Drive train, nacelle	44.8	255.63	562.77	1282.00	2474.26
Low speed shaft	1.5	8.43	19.86	56.26	120.90
Bearings	1.8	3.79	12.32	41.44	101.83
Gearbox	28.0	64.92	150.88	357.22	697.06
Mechanical brake, HS coupling, etc.	1.5	1.49	2.98	5.97	9.95
Generator	3.5	48.75	97.50	195.00	325.00
Variable speed electronics	0.0	50.25	100.50	201.00	335.00
Yaw drive & bearing	2.3	5.27	12.09	28.21	109.71
Main frame	2.5	21.45	63.99	192.12	433.63
Electrical connections	2.7	30.00	60.00	120.00	200.00
Hydraulic system	0.5	3.38	6.75	13.50	22.50
Nacelle cover	0.5	17.90	35.90	71.28	118.68
Control, safety system	13.0	10.00	10.20	10.49	10.78
Tower	18.0	69.66	183.83	551.42	1176.15
Turbine Capital Cost (TCC)	94.8	437.19	1004.33	2571.84	5145.62
Foundations	4.0	34.92	48.51	76.77	108.09
Transportation	3.5	26.59	51.00	253.41	1312.15
Roads, civil works	2.0	44.90	78.93	136.36	255.33
Assembly & installation	5.0	24.37	50.71	112.71	224.79
Electrical interface/connect	7.0	71.30	126.55	224.20	431.50
Permits, engineering	2.5	15.79	32.70	69.87	126.39

Rating (kW)	108	750	1,500	3,000	5,000
	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000	Baseline Component Costs $1,000
Balance of Station Cost (BOS)	24.0	217.87	388.41	873.31	2458.24
Project Uncertainty	0	0.00	0.00	0.00	0.00
Initial Capital Cost (ICC)	118.80	655.06	1392.74	3445.15	7603.86
Installed Cost per kW (cost in $)	1,100.00	873.41	928.50	1148.38	1520.77
Turbine Capital per kW sans BOS (cost in $)	877.78	582.92	669.55	857.28	1029.12
Levelized Replacement Costs (LRC) ($10.7/kW)	12.00	8.03	16.05	32.10	53.50
O&M $20/kW/Yr (O&M)	2.16	15.00	30.00	60.00	100.00
O&M ($/kWh)		0.0040	0.0037	0.0035	0.0033
Land ($/year/turbine)($0.00108/kWh)	0.25	2.43	5.20	11.20	19.58
Net 5.8 m/s Annual Energy Production MWh (AEP)	228.83	2219.73	4733.53	10265.32	18142.64
Net 6.67 m/s Annual Energy Production MWh (AEP)	305.04	2785.45	5840.23	12456.11	21743.67
Net Annual Energy Production MWh (AEP) (per WindPACT Study)	N/A	2254.463	4816.715	10371.945	18132.994
Fixed Charge Rate	11.85%	11.85%	11.85%	11.85%	11.85%
COE at 5.8 m/s $/kWh (see note 2)	0.1207	0.0437	0.0432	0.0475	0.0570
COE at 6.67 m/s $/kWh (see note 2)	0.0905	0.0349	0.0350	0.0391	0.0476
COE (per WindPACT study)(unknown wind velocity-IEC Class 2)		0.0431	0.0424	0.0470	0.0570

Notes:

1. The WindPACT study specified replacement costs at $15.00/kW/year, a 10.6% fixed charge rate, and O&M costs fixed at $0.008/kWh.

2. The 5.8 m/s and 6.67 m/s values for the 750 kW and larger turbines were calculated using wind data from “Alternative Design Study Report: WindPACT Advanced Wind Turbine Drive Train Designs Study,” and used the same spreadsheet Wally Thompson used for the 108 kW.

Appendix C. Detailed Task Outline

This is the revised task outline as determined during the July 27, 2005, kick-off meeting.

Main Objectives of the Program

- Decrease in capital cost for given capacity WTG gearbox
- Reduction in size and weight
- Potential increase in performance and/or efficiency

Accounting Task Codes in parentheses Assignments—7/27/05	**Timeline**	**Responsibility**
Task 1. Initial COE Projection—Deliverable 1	26 July 05	
1.1 (101) Review of stress comparisons—Micon 108 gearing 1.1.l Review of rating factors assumed		MEI, JRC, AJW
1.2 (102) Establish O&M costs for Micon 108 gearbox		WT, EUI
1.3 (103) Projection of rating factors and stress levels to be used in theoretical design of similarly rated Convoloid gear system 1.3.1 Project O&M costs as a function of operating stress level		MEI, JRC, WT
1.4 (104) Optimize design of Convoloid box for 108-KW capacity with reduced O&M costs		WT, MEI, JRC
1.5 (105) Analysis of cost factors for Convoloid gearbox 1.5.1 Compare to full-size (existing) involute box		TGW WT
1.6 (106) Calculate COE reduction based on cost, weight, and O&M factors for Convoloid box		
1.7 (107) Other		
Task 2—Kick-Off Meeting—Meet at TGW		
2.1 (201) Review plan details	27/28 July 05	BB
2.2 (202) Potential testing activities		MEI, TGW, DCE
2.3 (203) Future directions with NREL Project Team		BB
2.4 (204) Meeting summary		BB
Task 3—Gearbox Comparison Tests and Protocols		
3.1 (301) Refine Manufacturing Processes		
3.1.1 Reconcile grinding wheel dressing techniques for Convoloid gearing	19 Aug. 05	JRC, TGW, AJW
3.1.2 Measure and establish fixes for suspected Micon 108 housing distortions under 150%+ rated loading	10 July 05	MEI, TGW
3.1.3 Develop and execute on-machine tooth profile inspection protocol for Convoloid forms	9 Sept. 05	TGW, JRC, AJW
3.1.4 Develop and execute off-machine tooth profile and comprehensive gear inspection techniques for Micon 108 Convoloid gears		BB, TGW
3.1.5 Disassemble and inspect all Convoloid gears with new CMM software		TGW

Accounting Task Codes in parentheses Assignments—7/27/05	**Timeline**	**Responsibility**
3.1.6 Regrind and recheck all Convoloid gears until quality level is determined to be satisfactory (i.e., at least equal to the involute accuracy)	26 Aug. 05	TGW
3.2 (302) Compare stress levels at 200% rated load to insure bending strength life far surpasses that for surface durability	26 July 05	MEI, JRC, TGW
3.3 (303) Calculate the life of both boxes under their respective loads in a surface failure mode 3.3.1 Define "failure"		MEI, JRC, GEI MEI, JRC, GEI
3.4 (304) Analyze manufacturing sensitivities of Convoloid gearing and their effects on rating factors that should be used (at least initially) for time to failure calculations 3.4.1 Specific sliding 3.4.2 Absolute sliding 3.4.3 Film thicknesses 3.4.4 Center distance changes 3.4.5 Profile tolerances 3.4.6 Profile shifts 3.4.7 Spacing tolerances: 3.4.7.1 Individual	31 Oct. 05	GEI, MEI, JRC, AJW
3.4.7.2 Accumulative 3.4.8 Effects of crown 3.4.9 Effects of lead corrections 3.4.10 Gross misalignment 3.4.11 Bearing capacity		
3.4.12 Tangential and radial forces compared to involute designs	30 Sept. 05	WT, MEI, JRC
3.4.13 Possible constraints and/or enhancements that could impact the application of Convoloid gears in the future	26 July 05	TGW, MEI
3.4.13.1 Transition zone mismatches 3.4.13.2 Separating loads 3.4.13.3 Transition zone height overlaps	23 Aug. 05	MEI, TGW
3.5 (305) Relate testing protocol to expected loads for a Class 4 wind site 3.6 (306) Full analysis of housings, bearings, shafts, keys, and other components to determine safety factors of these components under protocol loads and speeds (to preclude premature failure) 3.7 (307) Description of planned test protocol 3.8 (308) Other		
Task 4—Testing of Involute and Convoloid Gear Sets		
4.1 (401) Instrumentation and data gathering system	26 July 05	DCE
4.2 (402) Parameter limits for automatic shut down	26 July 05	DCE, MEI
4.3 (403) Failure modes and precursors	26 Aug. 05	MEI, JRC, TGW
4.4 (404) Photography	20 Sept. 05	MEI
4.5 (405) Noise measuring	20 Sept. 05	MEI, DCE
4.6 (406) Execute test protocol	20 Sept. 05	DCE, TGW, GEI, MEI
4.7 (407) Other	18 May 06	AJW, JRC

Accounting Task Codes in parentheses Assignments—7/27/05	Timeline	Responsibility
4.10 (410) 3-Hole Test preliminary gearbox design 4.11 (411) 3-Hole Test project oversight 4.12 (412) 3-Hole Test—test fixture 4.13 (413) 3-Hole Test—test procedure 4.14 (414) 3-Hole Test—instrumentation 4.15 (415) 3-Hole Test—gearbox design 4.16 (416) 3-Hole Test—gearbox component detailing 4.17 (417) 3-Hole Test—gearbox component manufacture 4.18 (418) 3-Hole Test—gearbox assembly 4.19 (419) 3-Hole Test—test apparatus assembly 4.20 (420) 3-Hole Test—execute test protocol		
Task 5—Detailed Analysis of Test Results and Predictions		
5.1 (501) Comparison of Convoloid versus involute gear drive systems	03 Nov. 05	DCE, MEI, JRC, TGW, GEI
5.2 (502) Relationship of Convoloid gear performance to AGMA rating formulae and factors	2 Dec. 05	MEI, JRC, TGW, GEI
5.3 (503) Relationship of involute gear performance to AGMA rating formulae and factors	2 Dec. 05	MEI, JRC, TGW, GEI WT
5.4 (504) Trade-off summary—minimized COE for baseline turbine (projection of COE versus surface durability and fillet stress compromises)	15 Jan. 06	WT
5.5 (505) Extrapolation of data across a wide range of WTG gearboxes (see SOW for details)	30 Apr. 06	JRC, AJW, WT
5.6 (506) Test results report—Deliverable 3	15 June 06	MEI, GEI
5.7 (507) Other		
Task 6—Field Testing		
6.1 (601) Planning of Field Test Program 6.1.1 Establish number of units of each model 6.1.2 Location of field test(s) 6.1.3 Develop periodic inspection plan 6.1.4 Instrumentation design 6.1.5 Convoloid gearbox design 6.1.6 Involute gearbox design	Start 2 Jan. 06 Finish 15 Jan. 06	MEI, DCE, TGW, EUI, GEI
6.2 (602) Build instrumentation	7 Feb. 06	DCE
6.3 (603) Build Convoloid gearboxes	1 Feb. 06	TGW
6.4 (604) Build involute gearboxes	15 Jan. 06	TGW
6.5 (605) Ship gearbox/instrumentation packages to destination	15 Feb. 06	TGW
6.6 (606) Assemble gearboxes and instrumentation packages on site	1 Mar. 06	DCE, EUI
6.7 (607) Execute periodic inspection plan	15 Mar. 06	EUI
6.8 (608) Test results report—Deliverable 4—monthly start	15 Mar. 06	MEI, EUI, GEI
6.9 (609) Purchase long-lead items	3 Oct. 05	BB

Accounting Task Codes in parentheses Assignments—7/27/05	**Timeline**	**Responsibility**
Task 7—Detailed Design of Scaled-Up Gearbox Using Convoloid Gear Sets		
7.1 (701) Selection criteria for candidate gearbox	Start 15 June 06	TGW, MEI, EUI, GEI
7.2 (702) Prioritized list of objectives in scale-up effort	1 July 06	TGW, MEI, EUI, GEI
7.3 (703) Match 7.1 with 7.2	15 July 06	TGW, MEI, EUI, GEI
7.4 (704) Obtain actual gearboxes and/or drawings of candidates	15 Oct. 06	TGW, MEI, EUI, GEI
7.5 (705) Reverse-engineer and design Convoloid retrofit 7.5.1 Analyze Convoloid gear ratings using data developed in (5.2) above 7.5.2 Analyze bearing life, performance improvements; compare with existing Involute design 7.5.3 For the "selected" involute design using calculated stress levels, geometry, materials, and heat treatment, compare potential reductions in weight and cost of a "like-rated" Convoloid box	31 Dec. 06	JRC, AJW
7.6 (706) Other	1 Mar. 07	MEI, JRC, TGW, GEI
Task 8—Final Report, Deliverable 5	15 Apr. 07	MEI, GEI, BB, AJW

Appendix D. Gearing Life Analysis—Surface Failure Mode

4 November 2005

Definition of Surface Failure
Gear industry standards reference: AGMA 912-A04, ANSI/AGMA 1010-E95, ISO 10825:1995(E/F)

No definitive specification exists within the listed standards that explicitly states when a gear (or gear tooth) has failed in a surface failure mode. Additionally, combinations of different failure modes are common. Therefore determining the original cause requires years of experience. Unfortunately, many determinations are made after a gross destruction of a tooth flank. A summary of the most common surface failure modes is included at the end of this appendix.

The primary surface failure mode for which this study is concerned is a fatigue-type failure known as "macropitting." Therefore great care is required to prevent other failure modes (i.e., those due to insufficient lubrication, improper lubrication, mesh misalignment, manufacturing deviations, or improper material). The expected life of a surface in fatigue is directly related to contact stress and number of cycles.

As a surface is loaded, portions of the surface flex and eventually break away, leaving a "pit" or shallow depression in the surface. Most pits are small, but they can grow together and precipitate other failure modes—especially tooth breakage. The size of pitting and percentage of a tooth flank that is covered determine the definition of failure.

In some industries where transmission noise and smoothness is paramount, very little pitting is allowed. These industries specify failure as a very small percentage of the gearing contact face width. Other industries are concerned strictly with the transmission of rotational power, and therefore broaden the definition to the point of tooth breakage. For the wind turbine industry, the gearbox is an extremely important component of the turbine and, as such, must be very reliable. Failure of the gearbox creates extreme financial difficulties therefore Genesis defined failure as "any defect that will progress to the economic (operational) failure of the wind turbine gearbox."

Summary of Types of Common Surface Failure Modes
Pitting (Macropitting)

A contact fatigue failure occurs when fatigue cracks initiate either at the surface of the gear tooth or at a shallow depth below a loaded surface. Pits are formed when these cracks propagate back up to the surface and material separates. According to AGMA 925-A03, "Damage beginning on the order of 0.5 to 1.0 mm in diameter is considered to be a macropit." Four classifications have been observed: non-progressive, progressive, flake, and spall. Additional information and descriptive photos can be found in ANSI/AGMA 1010-E95, clause 6.1; and ISO 10825, clause 4.

Micropitting

Micropitting is a localized contact fatigue failure that exhibits a frosted, matte, or gray-stained appearance. The dynamic is the same as macropitting; however the pits are very fine. Micropitting occurs most frequently on surface-hardened gear teeth. Additional information and descriptive photos can be found in ANSI/AGMA 1010-E95, clause 6.2; and ISO 10825, clause 4.1.3.

Wear

This is a non-fatigue failure that is a degradation of a gear tooth involving removal or displacement of material due to mechanical, chemical, or electrical action. Specific classifications include adhesion, abrasion, polishing, corrosion, fretting corrosion, cavitation, erosion, electrical discharge, and rippling. Additional information and descriptive photos can be found in ANSI/AGMA 1010-E95, clause 3; and ISO 10825, clause 1.1, 1.2, 1.4, and 1.5.

Scuffing

This is a non-fatigue failure that involves severe adhesion between mating gear elements causing transfer of metal from one tooth surface to another due to welding and tearing. Degrees of scuffing are classified as mild, moderate, and severe. Additional information and descriptive photos can be found in ANSI/AGMA 1010-E95, clause 4; and ISO 10825, clause 2.

Plastic Deformation

This is a permanent deformation that occurs when the stress exceeds the yield strength of the material. Specific classifications include: indentation, cold flow, hot flow, rolling, tooth hammer, rippling, ridging, burr, root fillet yielding, and tip-to-root interference. Additional information and descriptive photos can be found in ANSI/AGMA 1010-E95, clause 5; and ISO 10825, clause 3.

Appendix E. Micon 108 Instrumentation Data Points

Micon 108 Lab Test Instrumentation Parameters

Note:
1. Involute gearbox is #1 gearbox (nearest the motor)
2. Convoloid gearbox is #2 gearbox

	Recorded in Data File	Alarms	Signal Type
Convoloid oil flow	Y		Pulses
Convoloid oil temp out	Y		
Convoloid oil temp in	Y		
Convoloid particles	Y		
Convoloid oil pressure at manifold	Y	Loss of Oil Pressure	DC Volts
Convoloid, outside high speed/intermediate separating force	Y		SG
Convoloid, outside low speed/intermediate separating force	Y		SG
Convoloid, outside, low speed bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid, outside, intermediate bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid, outside, high speed bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid, inside high speed/intermediate separating force	Y		SG
Convoloid, inside low speed/intermediate separating force	Y		SG
Convoloid, inside, low speed bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid, inside, intermediate bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid, inside, high speed bearing temperature	Y	Excessive Temperature	TC Isolated
Convoloid reservoir temperature			TC
Convoloid gearbox sump temperature	Y		TC
Convoloid magnetic chip detector	Y	Particles Detected	DC Volts
Convoloid filter pressure drop	Y	Excessive Filter Pressure Drop	
Convoloid X vibration	Y	Excessive Vibration	AC mV
Convoloid Y vibration	Y	Excessive Vibration	AC mV
Convoloid Z vibration	Y	Excessive Vibration	AC mV
Convoloid total vibration (calculated)	Y	Excessive Vibration	calc
Convoloid heat exchanger oil temp in			TC
Convoloid heat exchanger oil temp out			TC
Convoloid particle count 4u			DC Volts
Convoloid particle count 6u		High Count	DC Volts
Convoloid particle count 14u		High Count	DC Volts

High speed shaft RPM (motor RPM)	Y		Pulses
Circulating torque	Y		SG
Windup force strain gage	Y		SG
Windup force HP	Y		
Input wattage	Y		
Ambient temperature (computer side)	Y		TC

	Recorded in Data File	Alarms	Signal Type
Ambient temperature (test fixture side)	Y		TC
Smoke detector	Y	Smoke Detected	
Setpoint	Y		
Bypass valve	Y		
Hydraulic actuator pressure		Low Pressure	DC Volts

Involute oil flow	Y		Pulses
Involute oil temp out	Y		
Involute oil temp in	Y		
Involute particles	Y		
Involute oil pressure at manifold	Y	Loss of Oil Pressure	DC Volts
Involute, outside high speed/intermediate separating force	Y		SG
Involute, outside low speed/intermediate separating force	Y		SG
Involute, outside, low speed bearing temperature	Y	Excessive Temperature	TC Isolated
Involute, outside, intermediate bearing temperature	Y	Excessive Temperature	TC Isolated
Involute, outside, high speed bearing temperature	Y	Excessive Temperature	TC Isolated
Involute, inside high speed/intermediate separating force	Y		SG
Involute, inside low speed/intermediate separating force	Y		SG
Involute, inside, low speed bearing temperature	Y	Excessive Temperature	TC Isolated
Involute, inside, intermediate bearing temperature	Y	Excessive Temperature	TC Isolated
Involute, inside, high speed bearing temperature	Y	Excessive Temperature	TC Isolated
Involute reservoir temperature			TC
Involute Gearbox Sump Temperature	Y		TC
Involute magnetic chip detector	Y	Particles Detected	DC Volts
Involute filter pressure drop	Y	Excessive Filter Pressure Drop	
Involute X vibration	Y	Excessive Vibration	AC mV
Involute Y vibration	Y	Excessive Vibration	AC mV
Involute Z vibration	Y	Excessive Vibration	AC mV
Involute total vibration (calculated)	Y	Excessive Vibration	calc
Involute heat exchanger oil temp in			TC
Involute heat exchanger oil temp out			TC
Involute particle count 4u			DC Volts
Involute particle count 6u		High Count	DC Volts
Involute particle count 14u		High Count	DC Volts

Appendix F. Involute Gear Rating per AGMA 2001

Low Speed Gear Set

```
*******************************************************************************
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Micon 108
Low Speed Gearset

*******************************************************************************
                              *** ECHO OF INPUT ***
  1               1                        Data set number
  2Micon 108
  3Low Speed Gearset
  4
  5               2                2       In/Out =1 si =2 en 3 both units
  6               2                        =1 Short =2 Long output form
  7               1                        =1 External =2 Internal gear
  8               1                        =1 Single =2 Double helical
  9   20.000000                      deg   Normal pressure angle
 10   3.1750000                      in^-1 Normal diametral pitch
 11   10.000000                      deg   Helix angle
 12   13.976000                      in    Working center distance
 13              16               72       Pinion / Gear number of teeth
 14   5.9570000       23.261000      in    Pinion / Gear tip diameter
 15   5.1300000       5.1300000      in    net face width (gap not included)
 16              11                        AGMA Transmission Accuracy
 17   0.0000000       0.0000000      in    Pitch variation
 18  0.32500000     -0.62216448            addendum modification coefficient
 19  0.16000000E-01  0.16000000E-01        Tooth thinning for backlash(norm)
 20           10000            10000       Number of teeth on tool
 21   1.4700070       1.4700070            tool addendum (normalized)
 22 -0.62827565E-12 -0.62827565E-12        tool addendum modification coeff
 23  0.41000000E-01  0.41000000E-01        tool protuberance (normalized)
 24  0.41770000      0.41770000            tool tip radius (normalized)
 25  0.35955374E-01  0.35955374E-01        Stock allowance per side (norm)
 26   32.000000       32.000000      mu in surface finish (Ra)
 27   0.0000000       0.0000000      in    rim thickness (0= solid disk)
 28   10.000000                      in    Pinion bearing span
 29   2.0000000                      in    Pinion offset from ctr brg span
 30   296.66700                      rpm   Pinion speed (rpm)
 31   134580.05                     lbs in Pinion torque
 32   200.00000                      hours Design life
 33               1                1       number of mesh contacts
 34               2                2       Idler use =1 yes =2 no
 35               4                4       Surface hardness, =1 HV =2 HBN...
 36   58.000000       58.000000            Surface hardness values
 37               4                4       Core hardness, =1 HV =2 HBN...
 38   30.000000       30.000000            Core hardness values
 39               1                1       mat, =1 steel, =2 cast iron, ...
 40               1                1       Material sub classes
 41               3                3       Heat treatment =1 TH/NA =2 Flam..
 42               1                1       =1 quench & temper =2 Normalized
 43               0                0       pattern type =0 na =1 A =2 B
 44               2                2       material grade
 45   0.0000000       0.0000000      psi   Allow. contact stress number
```

```
 46   0.0000000        0.0000000      psi   Allow. bending stress number
 47  0.30000000       0.30000000            Poissons ratio
 48   30000000.        30000000.      psi   Modulus of elasticity
 49               2                         Lapped adjusted assy =1 yes =2 no
 50               1                         Lead crowned/corrected=1 yes=2 no
*******************************************************************************
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Low Speed Gearset

*******************************************************************************
  51               3                         =1 open =2 commer =3 prec =4 xtra
  52               2                         Point stress calc =1 hpstc =2 tip
  53   0.0000000                             Pitting resistance service factor
  54   0.0000000                             Bending strength service factor
  55   1.0000000                             Pitting resistance safety factor
  56   1.0000000                             Bending strength safety factor
  57   1.0000000        1.0000000            Reliability factor
  58   1.0000000                             Overload factor (clause 9)
  59   0.0000000                      lbs    Maximum tangential load
  60   0.0000000                             Load distribution factor
  61   0.0000000                             Load Distribution factor at yield
  62               2                         Application =1 conser =2 Indus
  63   0.0000000                             Size factor (clause 20)
  64   0.0000000                             Temperature factor (clause 19)
  65   0.0000000                             Surface condition factor
  66   0.0000000                             Dynamic factor (clause 8)
  67   0.0000000        0.0000000            Life factor Pitting
  68   0.0000000        0.0000000            Life factor Bending
  69  0.85000000       0.85000000            Life factor Pitting @ 10^10
  70  0.90000000       0.90000000            Life factor Bending @ 10^10

 General Geometry Error Messages  See User's Manual for More Information
 41) NOTE: This is a Low axial contact ratio (LACR) set. Axial contact
           ratio is (  0.90)

 *** GEAR GEOMETRY FACTOR J ERROR MESSAGES ***
   1) WARNING: LACR pinion J factor   0.56161526 is greater than
               conventional helical pinion J factor   0.55454096. The
               conservative J value is used for analysis.

 Rating Routine Error Messages  See User's Manual for More Information
 13) WARNING: Actual Unit Load ( 29691.0) exceeds allowable unit load for
              the Pinion.
 14) WARNING: Actual Unit Load ( 30306.1) exceeds allowable unit load for
              the Gear.
 17) WARNING: Calculated bending safety factor for the Pinion ( 0.906)
              is less than the input value.
 18) WARNING: Calculated bending service factor for the Gear ( 0.925)
              is less than the input value.

 Effective Case Error Messages  See User's Manual for More Information
  5) NOTE: Minimum case depth is calculated using contact stress above
           the maximum value recommended in the standard
```

```
******************************************************************************
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Low Speed Gearset

******************************************************************************
                           SINGLE HELICAL    PINION                GEAR
  ** GEAR GEOMETRY **
  Number of Teeth ........  (-)                  16                   72
  Norm. Dia. Pitch(Module)1/in(mm)                      3.1750(  8.0000)
  Normal Pressure Angle ..  deg                        20.0000
  Helix Angle ............  deg                        10.0000
  Op. Center Distance ....  in                         13.9760
  Outside Diameter .......  in                5.9570              23.2610
  Face Width .............  in                5.1300               5.1300
  Effective Face Width ...  in                          5.1300
  Gear Ratio .............  (-)                         4.5000

  ** GEAR GEOMETRY - NORMALIZED **
  Addendum Mod. Coeff ....  (-)               0.3250              -0.6222
  Tooth Thinned for B/L ..  (-)               0.0160               0.0160
  Stock Allow./Tooth Flank  (-)               0.0360               0.0360

  ** TOOL GEOMETRY - NORMALIZED **
  Add. Mod. Coeff of Tool   (-)               0.0000               0.0000
  Protuberance of Tool ...  (-)               0.0410               0.0410
  Addendum of Tool .......  (-)               1.4700               1.4700
  Tool Tip Radius ........  (-)               0.4177               0.4177
  Number of Teeth on Tool   (-)                10000                10000

  ** LOADING DATA **
  Design Life ............ hours                           200.
  Pinion Torque (input) .. lb in                    134580.0544
  Speed .................. rpm                296.67                65.93
  Pitch Line Velocity ....ft/min                       394.7190
  Max Tang. Load (input) .  lb                           0.0000
  Type of Practice .......                           INDUSTRIAL
  Type of Service ........                   PRECISION ENCLOSED GEARING
  ReliabilityFactor(input)  (-)               1.0000               1.0000

  ** ADDITIONAL INPUTS **
  Type of Gearset ........                             EXTERNAL
  Bearing Span ...........  in                         10.0000
  Distance "s1" ..........  in                          2.0000
  Rim Thickness ..........  in                0.0000               0.0000
  Dynamic factor .........  (-)                           N/A
  Transmission Accuracy No  (-)                           11
  Abs. Pitch Variation ...  (-)                           N/A
  Lead Correction or Crown                                YES
  Lapped or Adjusted .....                                NO
  Number of Contacts .....  (-)                   1                    1
  Idler ..................                      NO                   NO
  Spur Loading ...........                                N/A
```

```
*****************************************************************************
                          SINGLE HELICAL   PINION                GEAR
  ** INPUT FACTORS **
  Load Distribution Factor  (-)                      0.0000
  Dynamic Factor .........  (-)                      0.0000
  Surface Condition Factor  (-)                      0.0000
  Overload Factor-Default   (-)                      1.0000
  Size Factor ............  (-)                      0.0000
  Temperature Factor .....  (-)                      0.0000
  Load Dist. Ftr Overload   (-)                      0.0000
  Pitting Stress Cycle Fac  (-)             0.0000              0.0000
  Bending Stress Cycle Fac  (-)             0.0000              0.0000
  Pitt Stress Cycle @10^10  (-)             0.8500              0.8500
  Bend Stress Cycle @10^10  (-)             0.9000              0.9000

  ** MATERIAL DATA **
  Material ...............                  STEEL               STEEL
  Material Type ..........               PLAIN STEEL         PLAIN STEEL
  Material Grade or Class                   GRADE 2             GRADE 2
  Heat Treatment .........              CARB & HARDENED     CARB & HARDENED
  Induction Hard. Pattern                    N/A                 N/A
  Quench .................              QUENCH & TEMPER     QUENCH & TEMPER
  Surface Finish ......... mu in            32.0000             32.0000
  Modulus of Elasticity ..  psi          30000000.           30000000.
  Poisson's Ratio ........  (-)             0.3000              0.3000
  Allow.Cont.Stress(input)  psi                 0.                  0.
  Allow.Bend.Stress(input)  psi                 0.                  0.
  Core Hardness Number ...  (-)              30.00               30.00
  Core Hardness Scale ....                ROCKWELL C          ROCKWELL C
  Surface Hardness Number   (-)              58.00               58.00
  Surface Hardness Scale .                ROCKWELL C          ROCKWELL C
  Brinell Core Hardness ..  (-)             287.16              287.16
  Brinell Surface Hardness  (-)             620.29              620.29
  Allow Contact Str. No ..  psi            225000.             225000.
  Allow Bending Str. No ..  psi             65000.              65000.
  Elastic Coeff .......(lb/in^2)^.5                 2290.6039

  ** EFFECTIVE CASE DATA **
  Process Fact / Core Coef  psi           6400000.            6400000.
  Maximum Effective Case .  in              0.0861              0.1260
  Min. Eff./ Total Case ..  in              0.0512              0.0512
  Heavy Minimum ..........  in              0.0722              0.0722
  Normal Minimum .........  in              0.0444              0.0444
```

```
******************************************************************************
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******************************************************************************
                         SINGLE HELICAL    PINION                 GEAR
  ** PRESSURE ANGLES **
  Inv. of Norm.Press Angle  (-)                      0.014904
  Std. Trans. Pressure Ang  deg                       20.2836
  Normal Op. Press Angle .  deg                       18.9227
  Trans. Op. Press Angle .  deg                       19.1898
  Inv. of Trans. Press Ang  (-)                      0.015570
  Trans. Tip Press Angle .  deg              36.3183                 21.7897
  Inv. of T. Tip Press Ang  (-)             0.101191                0.019461

  ** PITCHES **
  Transverse Dia. Pitch .. 1/in                        3.1268
  Normal Base Pitch ......  in                         0.9298
  Transverse Base Pitch ..  in                         0.9424
  Axial Pitch ............  in                         5.6982

  ** HELIX ANGLES **
  Op. Helix Angle ........  deg                        9.9331
  Base Helix Angle .......  deg                        9.3913

  ** DIAMETERS **
  Ref. Pitch Diameter ....  in                5.1171                 23.0270
  Operating Pitch Diameter  in                5.0822                 22.8698
  Root Diameter ..........  in                4.4442                 21.7573
  Root Diameter (ref. 908)  in                4.3820                 21.6951
  Base Diameter ..........  in                4.7998                 21.5991

  ** TOOTH GEOMETRY **
  Tooth Whole Depth ......  in                0.7564                  0.7518
  Norm Tooth Thk (ref dia)  in                0.5642                  0.3471
  Norm Top Land Thickness   in                0.1537                  0.2613

  ** CONTACT RATIOS **
  Trans Contact Ratio ....  (-)                        1.5783
  Axial Contact Ratio ....  (-)                        0.9003
  Total Contact Ratio ....  (-)                        2.4786

  ** MISCELLANEOUS FACTORS **
  Gear Ratio Factor ......  (-)                        0.8182
  Effective Protuberance .  (-)               0.0072                  0.0072
  Minimum Contact Length .  in                         7.9638
  Fractional Part of m_f .  (-)                        0.9003
  Adjusted No of Teeth ...  (-)              16.2468                 73.1107
  Fractional Part of m_p .  (-)                        0.5783

  ** LINE OF ACTION DATA **
  Points Along LOA     A         B         C         D         E         F
  Dist(c_1-c_6)in   0.2766    0.8216    0.8353    1.2191    1.7641    4.5939
  (gamma_a-e) ...  -0.6688   -0.0163    0.0000    0.4595    1.1120
  Rol.Ang(eqs)deg   6.6043   19.6161   19.9410   29.1043   42.1161
```

```
*******************************************************************************
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*******************************************************************************
                           SINGLE HELICAL   PINION                  GEAR
  ** MESH FACTORS **
  Mesh Align Corr Factor .  (-)                       1.0000
  Surface Condition Factor  (-)                       1.0000
  Mesh Alignment Factor ..  (-)                       0.1307
  Lead Correction Factor .  (-)                       0.8000
  Pinion Proportion Factor  (-)                       0.1276
  Pinion Proportion Modif   (-)                       1.1000
  Load Distribution Factor  (-)                       1.2168

  ** AGMA 908 DATA (normalized) **
  I-Factor ...............  (-)                       0.2435
  J-Factor ...............  (-)             0.5545              0.4814
  Working Center Distance.  (-)                      44.3738
  Number of Teeth ........  (-)            16.0000             72.0000
  Effective Face Width ...  (-)                      16.2877
  X-Factor ...............  (-)             0.3250             -0.6222
  Number of Teeth (tool)..  (-)         10000.0000          10000.0000
  X-Factor (tool) ........  (-)             0.0000              0.0000
  Tool Addendum ..........  (-)             1.4700              1.4700
  Tool Tip Radius ........  (-)             0.4177              0.4177
  Effective Protuberance .  (-)             0.0072              0.0072
  Tooth Thinning for B/L .  (-)             0.0160              0.0160
  Strength Conc. Factor ..  (-)             1.3837              1.7088

  ** STRESS FACTORS SUMMARY **
  Load Dist Factor-Ovrload  (-)                       1.1439
  Number of Stress Cycles   (-)        0.35600E+07         0.79111E+06
  Reliability Factor .....  (-)             1.0000              1.0000
  Overload Factor ........  (-)                       1.0000
  Hardness Ratio Factor ..  (-)             1.0000              1.0000
  Size Factor ............  (-)                       1.0000
  Temperature Factor .....  (-)                       1.0000
  Yield Strength Factor ..  (-)             0.7500              0.7500
  Stress due to Wmax .....  psi           64162.83            59849.13
  Allow. Yield Strength ..  psi          105612.70           105612.70
  Max. Tangential Load ...  lb                     52961.5269
  Pitting Stress Cycle Fac  (-)             1.0595              1.1526
  Bending Stress Cycle Fac  (-)             1.0366              1.2187
  Dynamic Factor .........  (-)                       1.0501
  Transmission Accuracy No  (-)                           11
  Abs. Value of Pitch Var.  in                          N/A
  Calculated Driver Power   hp                      633.7375
  Member Torque .......... lb in         134580.05           605610.24
  Max. Pitch Line Vel ....ft/min                  10000.0000
  Tangential Load ........  lb                    52961.5269
```

```
******************************************************************************
                          SINGLE HELICAL   PINION                GEAR

                        **************************
                        *** MAIN RATING VALUES ***
                        **************************

                            ** PITTING **
  Allowable Transmitted Power at
    Unity Service Factor .  hp            643.8985            762.0390

  Allowable Power at
    input Service Factor .  hp            N/A                 N/A
  Service Factor (input) .  (-)                     0.0000
  Service Factor (calc) ..  (-)           N/A                 N/A

  Contact Load Factor ....  psi                  2482.8078
  Allow Contact Load Fact   psi          2522.6158           2985.4576
  Contact Stress Number ..  psi                236505.0210

  Allowable Power at
    input Safety Factor ..  hp            643.8985            762.0390
  Safety Factor (input) ..  (-)                     1.0000
  Safety Factor (calc) ...  (-)               1.01                1.10

                            ** BENDING **
  Allowable Transmitted Power at
    Unity Service Factor .  hp            574.0478            585.9389

  Allowable Power at
    input Service Factor .  hp            N/A                 N/A
  Service Factor (input) .  (-)                     0.0000
  Service Factor (calc) ..  (-)           N/A                 N/A

  Unit Load .............   psi                 32778.3329
  Allowable Unit Load ....  psi         29691.0480          30306.0823
  Bending Stress Number ..  psi         74381.8711          85680.9104

  Allowable Power at
    input Safety Factor ..  hp            574.0478            585.9389
  Safety Factor (input) ..  (-)                     1.0000
  Safety Factor (calc) ...  (-)               0.91                0.92

                            ** Power Summary **
  Input Power ...........   hp                    633.7375
  Allowable Transmitted Power for
    input Service Factor .  hp                       N/A
  Allowable Transmitted Power for
    input Safety Factor ..  hp                    574.0478
```

High Speed Gear Set

```
*****************************************************************************
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*****************************************************************************
                              *** ECHO OF INPUT ***
  1               1                           Data set number
  2Micon 108 Test
  3High Speed Gearset
  4
  5               2               2           In/Out =1 si =2 en 3 both units
  6               2                           =1 Short =2 Long output form
  7               1                           =1 External =2 Internal gear
  8               1                           =1 Single =2 Double helical
  9   21.980833                         deg   Normal pressure angle
 10   5.7760500                         in^-1 Normal diametral pitch
 11   10.000000                         deg   Helix angle
 12   9.8425000                         in    Working center distance
 13              16              96           Pinion / Gear number of teeth
 14   3.3090000       17.105000         in    Pinion / Gear tip diameter
 15   3.1500000       3.1500000         in    net face width (gap not included)
 16              11                           AGMA Transmission Accuracy
 17   0.0000000       0.0000000         in    Pitch variation
 18  0.38010000     -0.39320913              addendum modification coefficient
 19  0.12916389E-01  0.12916390E-01           Tooth thinning for backlash(norm)
 20           10000           10000           Number of teeth on tool
 21   1.3634389       1.3634389               tool addendum (normalized)
 22  0.99936876E-12  0.99936876E-12           tool addendum modification coeff
 23  0.69312600E-01  0.69312600E-01           tool protuberance (normalized)
 24  0.37544325      0.37544325               tool tip radius (normalized)
 25  0.60453910E-01  0.60453910E-01           Stock allowance per side (norm)
 26   32.000000       32.000000         mu in surface finish (Ra)
 27   0.0000000       0.0000000         in    rim thickness (0= solid disk)
 28   10.000000                         in    Pinion bearing span
 29   3.0000000                         in    Pinion offset from ctr brg span
 30   2050.0000                         rpm   Pinion speed (rpm)
 31   19468.000                        lbs in Pinion torque
 32   200.00000                         hours Design life
 33               1               1           number of mesh contacts
 34               2               2           Idler use =1 yes =2 no
 35               4               4           Surface hardness, =1 HV =2 HBN...
 36   58.000000       58.000000               Surface hardness values
 37               4               4           Core hardness, =1 HV =2 HBN...
 38   30.000000       30.000000               Core hardness values
 39               1               1           mat, =1 steel, =2 cast iron, ...
 40               1               1           Material sub classes
 41               3               3           Heat treatment =1 TH/NA =2 Flam..
 42               1               1           =1 quench & temper =2 Normalized
 43               0               0           pattern type =0 na =1 A =2 B
 44               2               2           material grade
 45   0.0000000       0.0000000         psi   Allow. contact stress number
 46   0.0000000       0.0000000         psi   Allow. bending stress number
 47  0.30000000      0.30000000               Poissons ratio
 48   30000000.       30000000.         psi   Modulus of elasticity
 49               2                           Lapped adjusted assy =1 yes =2 no
 50               1                           Lead crowned/corrected=1 yes=2 no
```

```
********************************************************************************
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********************************************************************************
 51               3                             =1 open =2 commer =3 prec =4 xtra
 52               2                             Point stress calc =1 hpstc =2 tip
 53   0.0000000                                 Pitting resistance service factor
 54   0.0000000                                 Bending strength service factor
 55   1.0000000                                 Pitting resistance safety factor
 56   1.0000000                                 Bending strength safety factor
 57   1.0000000        1.0000000                Reliability factor
 58   1.0000000                                 Overload factor (clause 9)
 59   0.0000000                         lbs     Maximum tangential load
 60   0.0000000                                 Load distribution factor
 61   0.0000000                                 Load Distribution factor at yield
 62               2                             Application =1 conser =2 Indus
 63   0.0000000                                 Size factor (clause 20)
 64   0.0000000                                 Temperature factor (clause 19)
 65   0.0000000                                 Surface condition factor
 66   0.0000000                                 Dynamic factor (clause 8)
 67   0.0000000        0.0000000                Life factor Pitting
 68   0.0000000        0.0000000                Life factor Bending
 69  0.85000000       0.85000000                Life factor Pitting @ 10^10
 70  0.90000000       0.90000000                Life factor Bending @ 10^10
```

```
******************************************************************************
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******************************************************************************
                           SINGLE HELICAL   PINION                GEAR
  ** GEAR GEOMETRY **
  Number of Teeth ........  (-)                 16                      96
  Norm. Dia. Pitch(Module)1/in(mm)                    5.7761(  4.3975)
  Normal Pressure Angle ..  deg                      21.9808
  Helix Angle ............  deg                      10.0000
  Op. Center Distance ....  in                        9.8425
  Outside Diameter .......  in              3.3090              17.1050
  Face Width .............  in              3.1500               3.1500
  Effective Face Width ...  in                        3.1500
  Gear Ratio .............  (-)                       6.0000

  ** GEAR GEOMETRY - NORMALIZED **
  Addendum Mod. Coeff ....  (-)             0.3801              -0.3932
  Tooth Thinned for B/L ..  (-)             0.0129               0.0129
  Stock Allow./Tooth Flank  (-)             0.0605               0.0605

  ** TOOL GEOMETRY - NORMALIZED **
  Add. Mod. Coeff of Tool   (-)             0.0000               0.0000
  Protuberance of Tool ...  (-)             0.0693               0.0693
  Addendum of Tool .......  (-)             1.3634               1.3634
  Tool Tip Radius ........  (-)             0.3754               0.3754
  Number of Teeth on Tool   (-)              10000                10000

  ** LOADING DATA **
  Design Life ............ hours                        200.
  Pinion Torque (input) .. lb in                 19468.0000
  Speed .................. rpm             2050.00               341.67
  Pitch Line Velocity ....ft/min                  1509.2454
  Max Tang. Load (input) .  lb                       0.0000
  Type of Practice .......                       INDUSTRIAL
  Type of Service ........                 PRECISION ENCLOSED GEARING
  ReliabilityFactor(input)  (-)             1.0000               1.0000

  ** ADDITIONAL INPUTS **
  Type of Gearset ........                          EXTERNAL
  Bearing Span ...........  in                       10.0000
  Distance "s1" ..........  in                        3.0000
  Rim Thickness ..........  in              0.0000               0.0000
  Dynamic factor .........  (-)                         N/A
  Transmission Accuracy No  (-)                          11
  Abs. Pitch Variation ...  (-)                         N/A
  Lead Correction or Crown                              YES
  Lapped or Adjusted .....                              NO
  Number of Contacts .....  (-)                  1                    1
  Idler ..................                      NO                   NO
  Spur Loading ...........                              N/A
```

```
******************************************************************************
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******************************************************************************
                          SINGLE HELICAL   PINION              GEAR
  ** INPUT FACTORS **
  Load Distribution Factor  (-)                      0.0000
  Dynamic Factor .........  (-)                      0.0000
  Surface Condition Factor  (-)                      0.0000
  Overload Factor-Default   (-)                      1.0000
  Size Factor ............  (-)                      0.0000
  Temperature Factor .....  (-)                      0.0000
  Load Dist. Ftr Overload   (-)                      0.0000
  Pitting Stress Cycle Fac  (-)             0.0000             0.0000
  Bending Stress Cycle Fac  (-)             0.0000             0.0000
  Pitt Stress Cycle @10^10  (-)             0.8500             0.8500
  Bend Stress Cycle @10^10  (-)             0.9000             0.9000

  ** MATERIAL DATA **
  Material ...............                  STEEL              STEEL
  Material Type ..........               PLAIN STEEL        PLAIN STEEL
  Material Grade or Class                  GRADE 2            GRADE 2
  Heat Treatment .........               CARB & HARDENED    CARB & HARDENED
  Induction Hard. Pattern                    N/A                N/A
  Quench .................              QUENCH & TEMPER    QUENCH & TEMPER
  Surface Finish ......... mu in            32.0000            32.0000
  Modulus of Elasticity ..  psi          30000000.          30000000.
  Poisson's Ratio ........  (-)             0.3000             0.3000
  Allow.Cont.Stress(input)  psi                 0.                 0.
  Allow.Bend.Stress(input)  psi                 0.                 0.
  Core Hardness Number ...  (-)              30.00              30.00
  Core Hardness Scale ....                ROCKWELL C         ROCKWELL C
  Surface Hardness Number   (-)              58.00              58.00
  Surface Hardness Scale .                ROCKWELL C         ROCKWELL C
  Brinell Core Hardness ..  (-)             287.16             287.16
  Brinell Surface Hardness  (-)             620.29             620.29
  Allow Contact Str. No ..  psi            225000.            225000.
  Allow Bending Str. No ..  psi             65000.             65000.
  Elastic Coeff .......(lb/in^2)^.5               2290.6039

  ** EFFECTIVE CASE DATA **
  Process Fact / Core Coef  psi           6400000.           6400000.
  Maximum Effective Case .  in              0.0325             0.0675
  Min. Eff./ Total Case ..  in              0.0278             0.0278
  Heavy Minimum ..........  in              0.0368             0.0368
  Normal Minimum .........  in              0.0265             0.0265
```

```
****************************************************************************
AMERICAN GEAR MANUFACTURERS ASSOCIATION    Time: 15:47:40    Date: 2008/11/17
Version 1.07 GEAR RATING PER AGMA 2001-C95 Data Set:   1     Page  5 of  7
Micon 108 Test
High Speed Gearset

****************************************************************************
                          SINGLE HELICAL   PINION                    GEAR
  ** PRESSURE ANGLES **
  Inv. of Norm.Press Angle  (-)                        0.019999
  Std. Trans. Pressure Ang  deg                         22.2870
  Normal Op. Press Angle .  deg                         21.9490
  Trans. Op. Press Angle .  deg                         22.2547
  Inv. of Trans. Press Ang  (-)                        0.020883
  Trans. Tip Press Angle .  deg               38.1365                   24.0839
  Inv. of T. Tip Press Ang  (-)              0.119522                  0.026641

  ** PITCHES **
  Transverse Dia. Pitch .. 1/in                          5.6883
  Normal Base Pitch ......  in                           0.5044
  Transverse Base Pitch ..  in                           0.5110
  Axial Pitch ............  in                           3.1322

  ** HELIX ANGLES **
  Op. Helix Angle ........  deg                          9.9977
  Base Helix Angle .......  deg                          9.2664

  ** DIAMETERS **
  Ref. Pitch Diameter ....  in                 2.8128                   16.8768
  Operating Pitch Diameter  in                 2.8121                   16.8729
  Root Diameter ..........  in                 2.5186                   16.3148
  Root Diameter (ref. 908)  in                 2.4667                   16.2629
  Base Diameter ..........  in                 2.6027                   15.6160

  ** TOOTH GEOMETRY **
  Tooth Whole Depth ......  in                 0.3952                    0.3951
  Norm Tooth Thk (ref dia)  in                 0.3228                    0.2148
  Norm Top Land Thickness   in                 0.0580                    0.1206

  ** CONTACT RATIOS **
  Trans Contact Ratio ....  (-)                          1.5345
  Axial Contact Ratio ....  (-)                          1.0057
  Total Contact Ratio ....  (-)                          2.5402

  ** MISCELLANEOUS FACTORS **
  Gear Ratio Factor ......  (-)                          0.8571
  Effective Protuberance .  (-)                0.0133                    0.0133
  Minimum Contact Length .  in                           4.8879
  Fractional Part of m_f .  (-)                          0.0057
  Adjusted No of Teeth ...  (-)               16.2468                   97.4810
  Fractional Part of m_p .  (-)                          0.5345

  ** LINE OF ACTION DATA **
  Points Along LOA      A         B         C         D         E         F
  Dist(c_1-c_6)in    0.2375    0.5107    0.5325    0.7486    1.0217    3.7276
  (gamma_a-e) ...   -0.5539   -0.0410    0.0000    0.4057    0.9187
  Rol.Ang(eqs)deg   10.4586   22.4847   23.4458   32.9586   44.9847
```

```
********************************************************************************
                           SINGLE HELICAL    PINION              GEAR
  ** MESH FACTORS **
  Mesh Align Corr Factor .  (-)                         1.0000
  Surface Condition Factor  (-)                         1.0000
  Mesh Alignment Factor ..  (-)                         0.1069
  Lead Correction Factor .  (-)                         0.8000
  Pinion Proportion Factor  (-)                         0.1139
  Pinion Proportion Modif   (-)                         1.1000
  Load Distribution Factor  (-)                         1.1857

  ** AGMA 908 DATA (normalized) **
  I-Factor ..............   (-)                         0.2867
  J-Factor ..............   (-)               0.6114              0.5716
  Working Center Distance.  (-)                        56.8508
  Number of Teeth ........  (-)              16.0000             96.0000
  Effective Face Width ...  (-)                        18.1946
  X-Factor ..............   (-)               0.3801             -0.3932
  Number of Teeth (tool)..  (-)           10000.0000          10000.0000
  X-Factor (tool) ........  (-)               0.0000              0.0000
  Tool Addendum ..........  (-)               1.3634              1.3634
  Tool Tip Radius ........  (-)               0.3754              0.3754
  Effective Protuberance .  (-)               0.0133              0.0133
  Tooth Thinning for B/L .  (-)               0.0129              0.0129
  Strength Conc. Factor ..  (-)               1.4023              1.4993

  ** STRESS FACTORS SUMMARY **
  Load Dist Factor-Ovrload  (-)                         1.1154
  Number of Stress Cycles   (-)          0.24600E+08         0.41000E+07
  Reliability Factor .....  (-)               1.0000              1.0000
  Overload Factor ........  (-)                         1.0000
  Hardness Ratio Factor ..  (-)               1.0000              1.0000
  Size Factor ...........   (-)                         1.0000
  Temperature Factor .....  (-)                         1.0000
  Yield Strength Factor ..  (-)               0.7500              0.7500
  Stress due to Wmax .....  psi             43363.59            43390.32
  Allow. Yield Strength ..  psi            105612.70           105612.70
  Max. Tangential Load ...  lb                     13845.6693
  Pitting Stress Cycle Fac  (-)               0.9790              1.0512
  Bending Stress Cycle Fac  (-)               1.0015              1.0340
  Dynamic Factor .........  (-)                         1.0921
  Transmission Accuracy No  (-)                             11
  Abs. Value of Pitch Var.  in                            N/A
  Calculated Driver Power   hp                        633.4825
  Member Torque .......... lb in            19468.00           116808.00
  Max. Pitch Line Vel ....ft/min                     10000.0000
  Tangential Load ........  lb                     13845.6693
```

```
                          SINGLE HELICAL   PINION               GEAR

                       *************************
                       *** MAIN RATING VALUES ***
                       *************************

                              ** PITTING **
  Allowable Transmitted Power at
    Unity Service Factor .  hp            829.8249            956.6122

  Allowable Power at
    input Service Factor .  hp            N/A                 N/A
  Service Factor (input) .  (-)                      0.0000
  Service Factor (calc) ..  (-)           N/A                 N/A

  Contact Load Factor ....  psi                   1823.5296
  Allow Contact Load Fact   psi          2388.7166           2753.6838
  Contact Stress Number ..  psi                 192468.2357

  Allowable Power at
   input Safety Factor ..  hp             829.8249            956.6122
 Safety Factor (input) ..  (-)                       1.0000
 Safety Factor (calc) ...  (-)                1.14                1.23

                             ** BENDING **
 Allowable Transmitted Power at
   Unity Service Factor .  hp             778.8305            751.5908

 Allowable Power at
   input Service Factor .  hp             N/A                 N/A
 Service Factor (input) .  (-)                       0.0000
 Service Factor (calc) ..  (-)            N/A                 N/A

 Unit Load .............  psi                    25388.3423
 Allowable Unit Load ....  psi          31213.5137          30121.8140
 Bending Stress Number ..  psi          52949.6347          56645.8193

 Allowable Power at
   input Safety Factor ..  hp             778.8305            751.5908
 Safety Factor (input) ..  (-)                       1.0000
 Safety Factor (calc) ...  (-)                1.23                1.19

                             ** Power Summary **
 Input Power ...........  hp                        633.4825
 Allowable Transmitted Power for
   input Service Factor .  hp                         N/A
 Allowable Transmitted Power for
   input Safety Factor ..  hp                       751.5908
```

Appendix G. Comparison of Stress Levels at 200% Rated Load

13 October 2005

Summary of Gear Bending Strength and Surface Durability Ratings

Reference industry standards: ANSI/AGMA 2001-E04

An analysis of all gear elements was performed to confirm that the gear tooth bending strength exceeds the expected surface durability under the applied test loads (200% of normal rated load). Using the gear industry helical gear rating standard, each involute gear set was studied to determine its safety factor with respect to the applied test load. Stresses for the Convoloid gear form have been calculated using an incremental method based upon actual tooth contact and stress allowables from the industry standard. Load distribution factors, dynamic factors, and allowable stresses are similar in both studies.

Results of the Stress Analysis

Involute

High Speed Gear Set	Pitting	Bending-Pinion	Bending-Gear
Pinion Torque (lb-in)	15,750		
Load Distribution Factor	1.25		
Dynamic Factor	1.05		
Stress Number (psi)	174,376	43,634	46,683
Allowable Stresses (psi)	225,000	65,000	65,000
Stress Cycle Factors	0.982	1.004	1.037
Safety Factors	1.267	1.496	1.443

Low Speed Gear Set	Pitting	Bending-Pinion	Bending-Gear
Pinion Torque (lb-in)	94,500		
Load Distribution Factor	1.30		
Dynamic Factor	1.05		
Stress Number (psi)	204,893	56,127	64,558
Allowable Stresses (psi)	225,000	65,000	65,000
Stress Cycle Factors	1.060	1.037	1.219
Safety Factors	1.164	1.200	1.227

Convoloid

The torques, load distribution factors, dynamic factors, allowable stresses, and stress cycle factors used to calculate the stresses are the same as those used by Mr. Don McVittie in his reports HS Inv Opt for Pitting AGMA and LS Inv Opt for Bending AGMA.

High Speed Gear Set	Pitting	Bending-Pinion	Bending-Gear
Pinion Torque (lb-in)	15,750		
Stresses	145,889	32,737	34,722
Load Distribution Factor	1.25		
Dynamic Factor	1.05		
Factored Stresses (psi)	167,137	42,967	45,573
Allowable Stresses (psi)	225,000	65,000	65,000
Stress Cycle Factors	0.982	1.004	1.037
Factored Allowable Stresses (psi)	220,950	65,260	67,405
Safety Factors	1.322	1.519	1.479

Low Speed Gear Set	Pitting	Bending-Pinion	Bending-Gear
Pinion Torque (lb-in)	94,500		
Stresses	159,223	36,290	37,002
Load Distribution Factor	1.30		
Dynamic Factor	1.05		
Factored Stresses	186,025	49,536	50,507
Allowable Stresses	225,000	65,000	65,000
Stress Cycle Factors	1.060	1.037	1.219
Factored Allowable Stresses	238,500	67405	79,235
Safety Factors	1.282	1.361	1.569

Summary

In all instances, contact stress safety factors (pitting) are lower than bending stress safety factors. Therefore, the gearing should fail in pitting before a tooth breakage failure should occur. The Convoloid stresses are always lower than the corresponding involute stresses. The torque values used here are the same as those used in submitted by Mr. Don McVittie in his Test Design Review of 18 March 2004. These torques are slightly more than 200% of the rated torque (i.e., that required to produce 108 kW at an field generator speed of 1,220 rpm), but they correspond to the appropriate loads experienced by the gearbox after taking into account efficiency losses throughout the wind turbine system. Test speed is to be 1,780 rpm.

Appendix H. Micon 108 Gearing Life Calculation

13 October 2005

Summary of Gear Surface Durability Life

Reference industry standards: ANSI/AGMA 2001-C95

Reference study documents: Task 3.2—COMPARISON OF STRESS LEVELS AT 200% RATED LOAD

The equation for allowable life was derived from AGMA 2001-C95 from equations 4 and 47, and Figure 17. Equation 27 was used to confirm the equality statement in equation 4 for the involute gearing. The stresses applied to each gear element were reported in the referenced document. The time of test, as required in the test protocol, is 200 hours.

Results of the Pitting Life Analysis

Involute

High Speed Gear Set	
Test protocol stress:	174,376 psi
Allowable stress number:	225,000 psi
Calculated Stress Cycle factor for pitting:	0.775004
Life:	8,872 hours
Low Speed Gear Set:	
Test protocol stress:	204,893 psi
Allowable stress number:	225,000 psi
Calculated Stress Cycle factor for pitting:	0.910636
Life:	2,985 hours

Convoloid

High Speed Gear Set	
Test protocol stress:	167,137 psi
Allowable stress number:	225,000 psi
Calculated Stress Cycle factor for pitting:	0.742831
Life:	18,916 hours
Low Speed Gear Set:	
Test protocol stress:	186,025 psi
Allowable stress number:	225,000 psi
Calculated Stress Cycle factor for pitting:	0.826778
Life:	16,756 hours

Appendix I. Component Analysis at 240% Rated Load

TASK (3.6)
18 January 2006 (rev. A)

Summary of Component Safety Factors
NREL CONTRACT #ZAM-5-33200-12

Calculations based upon increased test procedure loads (i.e., 240% torque, 200 hours life, 2,700 rpm HS shaft speed). Bearing lives are catalog L_1 (99% reliability). Key stresses include the 2.0 overload factor as represented in AGMA6001. Shaft stresses are calculated per AGMA6001.

BEARINGS

High Speed Pinion: Tapered Roller Bearings (# 32312, # 33215)

- Applied Power: 834 hp
- Speed: 2,700 rpm
- Calc. Torque: 19,468 lb-in (applied torque is 240% operational torque)
- Minimum Bearings L_1 = 213 hours

Intermediate Pinion: Spherical Roller Bearings (# 22319CJ)

- Applied Power: 834 hp
- Speed: 448.8 rpm
- Calc. Torque: 117,119 lb-in (applied torque is 240% operational torque)
- Minimum Bearings L_1 = 318 hours

Low Speed Pinion: Cylindrical Roller Bearings (# NCF 2948)

- Applied Power: 834 hp
- Speed: 100 rpm
- Calc. Torque: 525,629 lb-in (applied torque is 240% operational torque)
- Minimum Bearings L_1 = 2,282 hours

KEYS
Note: All key calculations based upon AGMA6001 inclusion of 2.0 overload factor.

High Speed Shaft (Involute)

- Applied Torque: 19,468 lb-in
- Key Size: ½ in x ½ in x 3 ¾ in
- Tensile Yield Strength: 61,000 psi (applied torque is 240% operational torque)

- Minimum Shear Safety Factor = 1.03
- Minimum Compressive S.F. = 1.29

High Speed Shaft (Convoloid)

- Applied Torque: 19,468 lb-in
- Key size: ¾ in x ¾ in x 3 in
- Tensile Yield Strength: 61,000 psi (applied torque is 240% operational torque)
- Minimum Shear Safety Factor = 1.70
- Minimum Compressive Safety Factor = 2.09

Intermediate Pinion

- Applied Torque: 117,119 lb-in
- Key Size: 5/8 in x 1.1 in x 3.15 in
- Tensile Yield Strength: 130,000 psi (applied torque is 240% operational torque)
- Minimum Shear Safety Factor = 1.34
- Minimum Compressive S.F. = 1.01

Low Speed Shaft

- Applied Torque: 525,629 lb-in
- Key Size: 1 ¼ in x 2.2 in x 5.19 in
- Tensile Yield Strength: 75,000 psi (applied torque is 240% operational torque)
- Minimum Shear Safety Factor = 1.35
- Minimum Compressive Safety Factor = 1.03

SHAFT STRESS ANALYSIS—INTERMEDIATE PINION

The location of concern is the intermediate gear shoulder (0.09 radius) where a keyway bisects the shoulder. AGMA6001 barely addresses this case, indicating:

> "Experimental verification is preferred for superposition of stress concentration factors. Without verification, the smaller values should be used."

Therefore the more severe condition has been analyzed (shoulder with radius). As with the key stress analysis, a 2.0 overload factor is included in the peak load stress calculation (but not for the fatigue factors). This results in a (raw) safety factor that is twice the indicated value, providing some latitude for the overlap of stress concentration factors.

Applied Torque: 117,119 lb-in	% Alternating: 25%
Applied Moment: 75,753 lb-in	% Alternating: 100%
Applied Shear Force: 19,372 lb	% Alternating: 100%
Shaft Diameter: 3.979 in.	Shoulder Dia.: 4.570
Radius: 0.09 in	Surface Finish: 63 rms

Peak Load Safety Factor = 1.28
Fatigue Safety Factor = 1.00

By creating a smoother radius (to 32 rms from 63 rms) the fatigue safety factor increases to 1.06.

The AGMA standard assumes a stress cycle factor of 1.0 for all cycles above 10^6. Operating at a shaft speed of 448.8 rpm, this number of cycles is reached in 37 hours.

Appendix J. Manufacturing Sensitivities of Convoloid Gearing and Their Effects on Rating Factors

18 November 2005

Reference industry standards: ANSI/AGMA 2001-E04, AGMA 908-B89, AGMA 925-A03

The mesh action of the Convoloid tooth form is similar to the involute form in that both exhibit a conjugate action. This action creates a smooth transfer of power from one component to the other. Because of geometry differences between the Convoloid and the involute tooth forms, however, calculation methods for key factors used to rate a gear set are modified. Many rating factors are the same and react to manufacturing deviations similarly, but differences exist where factors are based upon tooth profile geometry. Section numbers refer to task numbers in the Detailed Task Summary (*see* Appendix C)

3.4.1. Specific Sliding—There are two different accepted ways of calculating specific sliding—one commonly is used by the lubrication industry, and the other is used by the gearing industry. The gearing industry defines specific sliding at a point on the pinion as the sliding velocity, divided by the roll velocity of that point on the pinion. To maintain consistency, Genesis chose the latter method in its analysis of this parameter, and determined the following results.

Table J.1. Comparison of the Specific Sliding Velocities for Convoloid Versus Involute Gear Sets Used in the Micon 108 Wind Turbine Gearbox

High Speed Gear Set	**Specific Sliding**			
	Pinion SAP	**Pinion O.D.**	**Gear SAP**	**Gear O.D.**
Involute	-1.4487	0.5586	-1.2656	0.5916
Convoloid	-0.7979	0.5168	-1.0696	0.4438

Low Speed Gear Set	**Specific Sliding**			
	Pinion SAP	**Pinion O.D.**	**Gear SAP**	**Gear O.D.**
Involute	-2.2828	0.6578	-1.9226	0.6954
Convoloid	-1.0241	0.5702	-1.3269	0.5059

From this summary data, it can be seen that the relationship of the sliding to rolling velocities for Convoloid gear sets are reduced throughout the mesh. This reduced “slip” of one gear tooth with respect to the other is highly desirable, reducing the risk of scoring and enhancing film-thickness characteristics.

3.4.2. Absolute Sliding Velocities—Values for the sliding velocity are computed from the difference in rolling velocities of the pinion and gear.

Table J.2. Comparison of the Absolute Sliding Velocities for Convoloid Versus Involute Gear Sets Used in the Micon 108 Wind Turbine Gearbox

High Speed Gear Set	Absolute Sliding Velocity (in/sec)			
	Pinion SAP	Pinion O.D.	Gear SAP	Gear O.D.
Involute	-64.1472	106.3861	-106.3861	64.1472
Convoloid	-70.0374	106.8713	-106.8713	70.0374

Low Speed Gear Set	Absolute Sliding Velocity (in/sec)			
	Pinion SAP	Pinion O.D.	Gear SAP	Gear O.D.
Involute	-20.6614	36.9765	-36.9765	20.6614
Convoloid	-24.5015	36.0520	-36.0520	24.5015

From this summary data, it can be seen that the sliding velocities are slightly greater for the Convoloid; however, the relationship of the sliding ratio to the rolling ratio (specific sliding) is smaller. This slight increase in absolute sliding velocity is thought not to be detrimental, due to the relative curvature of the Convoloid gearing.

3.4.3. Film Thickness—Film thickness is directly related to the curvature of the elements in contact and their sliding velocities. In the traditional involute tooth form, the two mating parts have a convex curvature (similar to two cylinders rolling against each other). This contact exhibits a constantly changing radius of curvature for each element, creating a varying film thickness over the tooth profile. In gears with small numbers of teeth, this condition causes widely varying sliding velocities. In contrast, the Convoloid profile creates a situation where one mating tooth portion is convex and the complementary portion is concave (similar to one cylinder rolling inside another) and the two profiles have a constant relative curvature. Thus, the oil film thickness can be consistent throughout the profile and can exhibit a capacity for greater film thickness.

3.4.4. Center Distance Changes—Within normal center distance tolerances common today on modern CNC machinery, Convoloid gearing sensitivity to center distance changes will not affect operation or rating expectations. However, this tolerance must not allow the center distance to be closer than the calculated value. Outside this generally accepted range, however, changes will detrimentally affect both of these parameters.

3.4.5. Profile Tolerances—Analysis of this parameter from the involute and Convoloid perspectives does not appear to show any significant differences in treatment. Until proven otherwise, standard AGMA and ISO quality tolerances will be mirrored from the involute to the Convoloid; that is, for Class 12 involute gears, the Convoloid tolerance of this given parameter will be replicated to its involute counterpart.

3.4.6. Profile Shifts—Convoloid design protocols do not use profile shifting as is done commonly with involute designs. The pitch diameter is established in a manner similar to involute practice, and Convoloid curvatures and transition zone geometry are struck from that point according to specific formulae.

3.4.7. Spacing Tolerances—Analysis of this parameter from the involute and Convoloid perspectives does not appear to show any significant differences in treatment. Until proven otherwise, standard AGMA and

ISO quality tolerances will be mirrored from the involute to the Convoloid; that is, for Class 12 involute gears, the Convoloid tolerance of this given parameter will be replicated to its involute counterpart.

3.4.8. Effects of Crown—Although Convoloid gearing can be crowned in a way very similar to involute gearing, the characteristic “twisted tooth” syndrome initially looks to be more pronounced with the Convoloid tooth form than with the involute form. “Twisted teeth” are characterized by exact leads at the pitch diameter with positive lead readings in the dedendum, and negative lead readings in the addendum on the same flank of the same tooth, or vice versa. This area is of special interest, and should be studied carefully going forward to better access its corrective procedures, affects on stress concentration factors (KHB), and other important parameters.

3.4.9. Lead Correction Factors—Analysis of this parameter from the involute and Convoloid perspectives does not appear to show any significant differences in treatment. Until proven otherwise, standard AGMA and ISO quality tolerances will be mirrored from the involute to the Convoloid, that is, for Class 12 involute gears, the Convoloid tolerance of this given parameter will be replicated to its involute counterpart.

3.4.10. Gross Misalignment—Analysis of this parameter from the involute and Convoloid perspectives does not appear to show any significant differences in treatment. Until proven otherwise, standard AGMA and ISO quality tolerances will be mirrored from the involute to the Convoloid, that is, for Class 12 involute gears, the Convoloid tolerance of this given parameter will be replicated to its involute counterpart.

3.4.11. Bearing Capacity—Although the radial and separating forces of Convoloid gearing closely approximate those in same sized involute gearing under the same load/speed spectrums, when Convoloid theory is applied to increase power density of a gear drive system to carry a certain load/speed spectrum, gear center distances and gear sizes are reduced as compared to the involute design (and maintain the same or lower Hertz and bending stresses), thus increasing the loads on the supporting bearing system. The bearing industry has recently developed processes and procedures to significantly increase the power density of their products. It is thought that these advancements will economically accommodate these anticipated increased loads, providing the industry with well-balanced gear drive systems (from a stress and reliability standpoint).

3.4.12. Tangential, Thrust, and Radial Forces—Tangential and thrust forces are calculated in the same manner for Convoloid and involute gear sets with similar gear geometry (e.g., operating pitch diameter and helix angle). Radial (or separating) forces for Convoloid gears are comparable to involute gears with larger pressure angles.

3.4.13. Possible Constraints / Enhancements—Preliminary calculations involving the Convoloid tooth form reveal opportunities for reducing the size of wind turbine gearboxes by 15% to 20% via center distance reduction. However, limitations exist with the power density of available bearings. Efforts are underway to determine the availability of enhancements by various bearing manufacturers to solve this problem.

Appendix K. Shakedown Protocol—Micon 108 in Preparation for Executing the Test Protocol

November 17, 2005

In preparation for executing the test protocol, a preliminary test—or shakedown—is required to confirm proper operation of equipment and test instrumentation, filter the test oil to protocol cleanliness, and familiarize personnel with the equipment. This procedure assumes all equipment and instrumentation has been assembled and tested in the static state. The purpose of this process is to find any discrepancies and take necessary actions in support of the test protocol.

1. Familiarize test personnel with instrumentation and test protocol. Set up and start manual test log.
2. Apply marking compound to teeth flanks—each tooth should be uniquely labeled. Per test protocol document of September 7, 2005.
3. Obtain and label two 3-oz samples of new oil from drum for later testing.
4. Clean the test oil by operating the oil pumps and oil heaters for 6 hours, or until the oil cleanliness reaches -/15/12 and the oil temperature reaches 150° F, prior to startup.
5. With the motor operating, but with 3% to 5% (228 lb to 380 lb-in) load, run the oil pumps until an -/15/12 oil cleanliness is again achieved.
6. Take one 3-oz oil sample from each gearbox and perform a preliminary patch test.
7. Bring the load up to 10% torque (760.4 lb-in) and operate for 1 hour. Confirm that instrumentation operates properly. Continuously monitor particulates; if they exceed -/18/13, shut the system down and determine the reason.
8. Increase load to 50% torque (3,801.8 lb-in) and operate for 1 hour.
9. Stop operation and inspect wear pattern. Photograph the results.
10. Restart and slowly increase load to 100% torque (7,603.6 lb-in) and operate for 1 hour.
11. Stop operation and inspect and photograph wear pattern.
12. Repeat Step 8 for 150% (11405.3 lb-in) and 200% (15,207.1 lb-in) load with inspections and photographs performed after each test. Note any abnormalities.

Note: The bulk oil temperature should not exceed 165° F, measured at inlet to gearbox, at any time. Nor should any bearing temperature exceed 190° F (an alarm will sound). If either of these conditions exists, stop the test run and contact the instrumentation engineer.

If there is a shutdown due to mechanical problems during this shakedown procedure, the procedure will be restarted with zero loads after the problem is repaired. If there is a shutdown due to instrumentation or lubrication problems, then the protocol will be repeated from the last successful load step. (Repeat one load step.)

Appendix L. Micon 108 Test Protocol

September 7, 2005

References

- Previous draft of test memo for objectives and general description of the test
- TGW layouts for test setup
- DCE specs for instrumentation

DCE specs for instrumentation

1) Preliminary checks:
  a) Gear accuracy checking and numbering
    i) Number and mark with a vibrating pencil all of the teeth on all pinions and gears. Number clockwise looking upwind toward the rotor end of the gearbox.
    ii) Identify and mark which flanks are the loaded flanks of the involute and Convoloid sets.
    iii) Check the contact patterns of the loaded flanks of the Convoloid sets in the test stand after finish grinding.
    iv) Document the contact with tape impressions and mark the pinion and gear teeth used so the same teeth can be used at assembly and during testing.
    v) Chart the accuracy of the involute sets on M&M.
    vi) Prior to charting lead, spray diemakers' ink on half of face width to document the thickness of the ink.
  b) **Torsional stiffness of the involute gearbox:** To estimate the appropriate torsional offset between the two gearboxes at no load, the involute gearbox will be pretested to 100% torque as follows.
    i) Mount the gearbox on the test fixture, with low speed shaft Ringfeder coupling tightened to specification.
    ii) Lock the low speed shaft to the test fixture with a torque arm and Ringfeder coupling.
    iii) Attach a 10-in diameter 360° protractor disc to the input shaft.
    iv) Torque the input shaft to 120% of nominal torque, recording torsional deflection at 5%, 15%, 30%, 60%, 90%, and 120% of nominal torque. (Test torque will be 200% of nominal torque.)
    v) Carefully reduce torque to zero, recording the deflection at 10% of nominal torque and zero torque.
    vi) Plot the results of the static torque test and report them to Genesis. Genesis will provide an estimate of the preliminary no-load torsional offset between the test and slave gearboxes from the static torque test results.

c) Contact patterns and gearbox cleanliness

i) Tooth marking

ii) Paint all of teeth of both gear boxes with diemakers' ink (Dykem red or blue) at final assembly. Touch up any teeth of the involute gearbox where marking is damaged in the torsional stiffness test.

iii) Select four teeth at 90° spacing on each gear of both gearboxes. Paint white dashes on the tips of those teeth, one dash at 0°, two at 90°, etc., for permanent identification. Number clockwise looking upwind toward the rotor end of the gearbox.

d) Preliminary test runs

i) Minimum load
ii) Torque up gearboxes to eliminate backlash and assemble couplings in high speed connecting shaft with gearboxes offset as specified by Genesis.
iii) The circulating/filter system should be run for a significant period of time until the oil cleanliness reaches -/15/12 before the gearboxes are rotated and loaded.
(a) Bring test rig up to speed and increase torque to 10% of nominal torque (5% of test torque)
(b) Run at 10% torque for 2 hours minimum or until the oil particle count is reduced to -/15/12 per ISO 4406.
(c) After 2 hours, stop operation, inspect, and record by digital photography the contact patterns of the four marked teeth on each gear of both gearboxes. Include a scale in photos for dimensional reference. Assign a unique file name to each photo to clearly identify.
(d) Use numbers and paint marks on teeth for identification.
(e) Repaint four teeth used for contact pattern checks.

Load steps: Repeat the procedure above at 30%, 60%, 90%, 120%, and 150% of nominal torque, with the same minimum run time, cleanliness requirement and recording contact patterns.

2) 200% load test

a) Re-apply marking compound.
b) Torque up gearboxes to eliminate backlash and assemble couplings in high speed connecting shaft with gearboxes offset as specified by Genesis.
c) Bring test rig up to speed and increase torque to 50% of nominal torque (25% of test torque)**.** Run at 50% torque for 5 minutes.
d) Increase torque to 100% and run for 5 minutes.
e) Increase torque to 150%.
f) After 1 hour, stop operation, inspect, and record by digital photography the contact patterns of the four marked teeth on each gear of both gearboxes. This counts as the first hour of the 200-hour test.
g) Restart at zero torque, increasing torque gradually to 200% over one minute.
h) After 24 hours, stop operation and record contact patterns and tooth condition of the marked teeth and any damaged teeth. Take an oil sample at each stoppage. Conduct a patch test of the oil and compare to ISO cleanliness requirements. Repeat this step every 24 hours of running time until 200 hours at 200% torque test is complete.

i) Sweep sump with a magnet prior to gear contact checks. Document debris on magnet with photos; collect and save debris.

3) Conditions which require stopping the test and inspection of the gear boxes.

a) Loss of test torque
b) Sudden temperature rise
c) Increase in noise level
d) Unusual noise (bangs, thumps, etc.)
e) Surface fatigue pitting—where macro pitting exceeds one half the face width of any teeth
f) Unusual contact patterns

4) Oil Samples

a) Two oil samples are to be taken each day from the same location from each gearbox and using the same procedures.
b) Oil samples must be specifically marked with subject gearbox and time of test. Any additional observations are to be documented.
c) Check and document oil cleanliness using patch test. Documentation is to include digital photo (with reference scale), additional observations, and original patch.
d) Retain one oil sample from each box for future oil tests.
  i) Possible further analysis by an outside laboratory (e.g., Herguth Labs)
  ii) Further tests could involve viscosity, particle analysis, and chemistry

5) Data Storage and Reports

a) Test data must be downloaded from the data-loggers daily, and be stored in at least two different forms and/or locations.
b) General observations of the test are to be documented (noting elapsed time of the test) during each shut-down period and at random intervals during the test.
c) Test samples must be appropriately labeled with, at minimum, subject gearbox and time of test, and be stored in a manner that will preserve the integrity of the sample.
d) Daily reports
  i) Verify daily that all collected data is reasonable and has been saved.
  ii) Summarize all measured parameters.
  iii) Note any significant events or changes.
  iv) Include several photos of contact patterns.
  v) Notes reference contact inspections.
e) Final Report
  i) Purpose of the test
  ii) Daily summary record of all measured parameters and observations
  iii) Complete list of all contact pattern and patch test photos taken
  iv) Conclusions

Appendix M. Test Protocol—Addendum A: Continuation of Test

September 7, 2005

Description

Without having a definitive failure mode, additional testing is required. Analysis of components allows an increase of load to 240% of rated torque, and a potential speed increase (up to 2,700 rpm).

1) Initial application of increased load and speed
   a) Follow standard protocol for normal startup to 200% torque (429,200 lb-in) and 1,768 rpm (60 Hz).
   b) Allow monitoring system and lubrication to stabilize for 1 hour at minimum.

2) Increase torque to 220% of rated load (472,100 lb-in) with existing motor speed and allow oil levels and temperatures to stabilize (minimum of 15 minutes).

3) Increase torque to 240% of rated load (515,000 lb-in low speed shaft torque—1,000 uE on the load cell) with existing motor speed and allow oil levels and temperatures to stabilize (minimum of 15 minutes).

4) Increase speed to 2,000 rpm (67.9 Hz) and maintain torque at the 240% level and allow oil levels and temperatures to stabilize (minimum of 15 minutes). Additional increases in speed are not advised due to the high speed shaft coupling speed limitations.

5) Planned shutdowns
   a) Scheduled shutdowns should occur Wednesday afternoon (1500 hours) and Saturday morning (0700), and should follow the shutdown procedure.
   b) Saturday shutdowns must be followed with securing the equipment for the weekend.

6) Emergency shutdown. If at any time during the test cycle any loud or unusual noises are heard in the test equipment, bearing temperatures go above 185º, or excessive vibration of any of the equipment occurs, then perform an emergency shutdown and contact the test director.

7) Inspections
   a) Inspections will be conducted the beginning of week on Monday morning (0900 hours) and midweek on Wednesday afternoon (1500 hours). At this time, perform detailed internal gear inspections of both gearboxes.
   b) Oil samples will be obtained from each gearbox each day at approximately 1500 hours following the established protocol.
   c) Computer data will be downloaded each day, and a new data sequence started for each computer at approximately 1500 hours.
   d) After inspections, if the gearboxes can satisfactorily continue to run, execute the startup procedure and return to 2,000 rpm and 240% load.
   e) The testing cycle continues until the inspector reports a defined failure has occurred. This failure could be advanced pitting, tooth cracks or breakage, or some other failure mode.

Appendix N. Description of Start-Up and Shutdown Procedures—Micon 108

Start-Up Procedure

To ensure proper lubrication and prevent improper overloads, this procedure was followed for every start-up.

- Main lubrication pumps are activated.
- Allow oil level to rise to desired oil-level mark. This level was established just above the low speed bearing race allowing the bearing rollers to dip into the oil, but below the level of the labyrinth oil seals. Oil level was moderated by adjusting ball valves installed in the oil drain hoses. Because a constant flow of oil was desired (to ensure clean oil and to remove heat from the gearbox), the amount of oil released through the drain had to equal the amount supplied. The drain valves required constant monitoring due to the variation in oil flow as the viscosity varied with temperature.
- Once the oil level was acceptable, the main drive motor was energized. The motor was controlled through a variable frequency drive that allowed for a soft start. Full speed was achieved over a 15-second time frame.
- With the drive motor at full speed (1,768 rpm—measured), torque gradually was added to the system by increasing pressure in the hydraulic cylinder. The resulting force from the cylinder induces a twist in the common low speed shaft, resulting in a system torque. The amount of force (in units of strain—measured by a strain gage installed in line with the cylinder) was increased to a predetermined value and required continuous monitoring until stable. The cylinder was located 48 in from the centerline of the low speed shaft; therefore a force of 9,144 lb was required. (equating to a micro-strain value of 886) for the 200% load condition (16,250 lb.-in. torque at the high speed shaft). The force of the cylinder caused the bottom of the torque arm structure to raise approximately 1¼ in from the at-rest position.
- Shut off oil reservoir heaters. Continue oil level monitoring. Monitor for any sounds that are not a part of the normal operation.

Shutdown Procedure

The system was shut down once a day for oil sample collection and gear tooth flank inspection.

- Retrieve 3 oil samples from each lubrication system (Convoloid and involute). One sample will be used immediately for oil particle analysis, one to be used for external oil analysis, and one sample will be saved in case further analyses are required.
- Gradually reduce the system torque until no load is applied to the system. The hydraulic cylinder linkage should be loose.
- De-energize the main motor. The variable frequency drive reduces the motor speed to zero over a 15-second time frame.
- When the system is static, shut off the lubrication pump motors. Turn on the oil reservoir heaters (set to 160°F).

Appendix O. Micon 108—Detail of Test

Test Setup

Two gearboxes were designed for this test. In using a "four-square" test, the two drives must have the same ratios. To achieve this result with gearboxes of differing geometries, a non-hunting tooth arrangement is required. Tooth counts for the involute were 16/96 (high speed) and 16/72 (low speed). Tooth counts for the Convoloid were 13/78 and 12/54. The resulting overall ratio is 27.00:1.

The original gearboxes were designed for operation in a 108-kW (electrical power) wind turbine, therefore this is the basis for the torque application for the test. Once the efficiencies of the generator were accommodated, the reference 100% torque was 24,800 Nm (219,500 lb.-in.) at the low speed shaft. The main drive motor provided approximately 25 hp at 1,767 rpm. Therefore, there was a 5.5% power loss through both gearboxes combined, equating to a 97.5% efficiency for each drive. See Appendix T for a detailed discussion of the efficiency.

As stated in the body of this report, the two test gearboxes were rebuilt to like-new conditions. The first drive was assembled with high-quality (minimum AGMA Class 12) involute gearing. The second drive employed gearing with the Convoloid technology. All material used for the gearing was 4320 alloy steel that exceeded AGMA Class 2 material specifications, and was carburized and hardened prior to final tooth grind.

Preliminary testing conducted in June 2005 revealed variations in the gear mesh contact patterns once torque application exceeded 100% of the rated load. Subsequent testing of the gearbox housing for stiffness when subjected to axial forces showed a significant deflection (0.007 in to 0.009 in at the bearing bores). This deflection was reduced dramatically when the cover plate was taper-pinned to the housing, essentially becoming part of the load-carrying structure. Following this modification, tooth contact pattern shifts were not a cause for concern.

Additionally, the early Convoloid test pinions showed hard contact and early micropitting around the transition zone. Verification of the profile of all gear elements by Brown & Sharpe (North Kingstown, RI) indicated excess material in this region. Subsequent profile development was conducted due to the tooth form error that caused micropitting during the 150% load test, a Convoloid a non-heat treated "dummy" pinion was reground to new data as provided by Dr. John Colbourne (Genesis Partners LP). This data series allowed for a greater concentration of points around the high-curvature transition zone. Subsequent results from B&S showed the gear profile to closely follow the electronic master with only minor errors. Additional modifications to the profile were added to cause a slight relief around the transition zone and to increase the curvature within the transition zone and minimize measurement errors within the zone.

Note that the cycle of grinding, shipping to B&S for verification, and return of the parts required a large time investment. Investigations are ongoing for a local or in-plant method for checking the Convoloid profile. Much promise exists with the introduction of freeform grinding and verification software that will address non-involute tooth profiles more readily.

A final check of the "dummy" pinion showed a profile that was very close to the electronic master. Subsequent verification of the high speed and low speed test pinions were not as close as the “dummy” pinion, but the profile was thought to be within an acceptable range (Figure P.1), especially when the high-curvature areas in the transition zone and at the tip were removed from the curve fit. There was some concern that additional grinding would affect the strength of the part by removing too much of the heat treatment case depth. After careful analysis, the resulting case depth after grind was found to be acceptable. The gears were not verified for profile, as the grinding machinery was deemed very reliable. Tooth lead and crown were checked on the traditional gear measurement machines.

Upon assembly into the housings, all meshes (both involute and Convoloid) were painted with a marking compound to determine their no-load contact. With the exception of the Convoloid high speed pinion, all contacts patterns were acceptable. The Convoloid high speed pinion, when centered on the active flank, showed contact nearer the outside edge (nearer the bearing) (Figure P.2). This type of contact causes hard edge loading and premature failure of the gear teeth. Regrinding the high speed pinion with a 0.001-in helix modification brought the contact pattern into the center of the face width (Figure P.3). This pinion was not re-verified by B&S due to time constraints and the feeling that, by using the same profile geometry data, the profile should be acceptably accurate to the previous check.

Crown Specification

Design specifications for crown on the Convoloid gears were calculated generally by AGMA standards, listed below.

- For the HS Gear: End relief only (0.0004 in) starting 0.56 in from each gear edge face
- For the LS Gear: End relief only (0.0003 in) starting 0.75 in from each gear edge face
- For the HS Pinion: 0.0008in circular crown over a 3.5-in reference face
- For the LS Pinion: 0.0010-in circular crown over a 5.5-in reference face

These values can be refined using ISO 6336 and AGMA 927 where load intensity, tooth geometry, and other important factors are considered. Reading the Convoloid tooth patches, and taking into account the tooth twisting phenomenon observed, the calculation method will be revised for future Convoloid pairs.

Tooth Twisting

Tooth twisting is a phenomenon that typically occurs when grinding both involute and non-involute forms with a single flank grinding wheel. It manifests itself, for example, as a positive lead error in the addendum of the gear, accurate lead at the pitch line, and negative lead error in the dedendum creating a cross lead on the same flank of the same tooth. Depending on manufacturing process and gear geometry, the lead errors could be reversed from those indicated above. Machine tool manufacturers of gear grinding machines have developed software to correct these situations, such that lead errors generally result in acceptable low to very low values.

Tip Relief

From observations made toward the end of the test run, tip relief on the Convoloid low speed gear appears to be correct because there is no micropitting in the low speed pinion root. The Convoloid high speed gear, however, has too much tip relief, as evidenced by use of only about 50% of the dedendum of the Convoloid high speed pinion, and only about 50% contact (in the upper half) of the high speed gear addendum. The methodology used to calculate tip relief in Convoloid gears must be studied and redefined. The test data, pictures, and measurements of the 108 test gears help to serve the objective, as would closer study of the AGMA/ISO practice in this critical area (*see* Figure P.18). Genesis partners developed new formulae which take into account the results and refines the end and tip relief calculations for future testing.

Test Observations

200% x 200-Hour Test

The presence of "smears" on the active surface of the gear teeth (easily visible in Figure O.13) indicates that some small particles went through the mesh. No damage could be felt due to these marks, and old marks tended to fade as the test progressed. Oil flowed through the heat exchangers after the filters, therefore it is thought that particles flushed from the exchangers could have caused the marks. These particles would have been soft aluminum, therefore explaining the lack of damage to the tooth surface (pictures and figures located in Appendix P).

Once grinding was complete and the gearboxes fully assembled, the shakedown procedure was begun on November 22, 2005. After the oil was filtered to acceptable limits, the 50% and 100% loads were applied—with inspections following each application of load. The teeth were fully "blued" prior to the test, therefore the full-load contact was readily apparent. More of the bluing was removed as the loads were increased during the shakedown procedure. All contact patterns were acceptable. Shakedown was completed on November 28, (year). Contact patterns for both the 150% and 200% were acceptable, and the test protocol was begun immediately.

Contact for the involute gear meshes (at 200% load) showed even contact across the face widths (Figure P.4, Figure P.5). Some bluing remains near the tops of the teeth due to tip relief. Very faint bluing is visible at the edges of the face widths alluding to a proper amount of lead crowning having been applied during manufacture. Note: As indicated later, this perceived 200% load actually only was 77% of rated load.

Contact for the Convoloid gear meshes (at 200% load) shows full contact along the transition zone. However, there is a significant portion (~20%) of the tooth flank with bluing in the high speed mesh (Figure P.6). A high percentage of the tip (due to tip relief) is still blue in comparison with the involute gear set. Also, at the ends of the high speed face width, significant bluing remains (due to crowning). Therefore, because contact in these areas would have removed the bluing, there might not be full contact in the entire active tooth flank. As a consequence, a smaller area of contact is required to carry the load leading to higher contact stresses. The wear pattern in the low speed mesh covers the full face width (Figure P.7).

There is some bluing remaining on the right side of the addendum and on the left side of the dedendum (as shown on the low speed gear—Figure P.8). During grinding of the Convoloid pinions, it was noticed that when a lead crown was applied to the tooth flank, the addendum and dedendum exhibited a slight relative "twist." Thus, the lead of the addendum and dedendum were slightly different.

Initially, the Convoloid gearbox was noticeably noisier that the involute gearbox. This perception was attributed to the partial contact exhibited in the wear patterns. It was thought that the crowning and end relief were incorrect. Upon application of the “true” 200% load, however, the contact spread to the entire face width. Also, perceived noise from the Convoloid decreased below that perceived by the involute.

As early as the first inspection after 18 hours of operation at the test load, fine micropitting began to appear on either side of the transition zone. The test continued as planned, with a special sensitivity towards closely monitoring the micropitting progression. Testing at 77% of the rated load continued up to the 179-hour point. Very slight micropitting was beginning to occur in the dedendum of the involute high speed pinion, located at the point of the beginning of tip relief of the mating gear. The low speed pinion showed distress where the edges of the gear contact the flank. In the Convoloid gearbox, micropitting progressed slowly around the transition zone of the high speed pinion, and began in the dedendum side of the high speed gear transition zone. Slight micropitting is visible on the low speed pinion. Noise for the Convoloid decreased slightly as the test progressed.

On December 6, it was discovered that the calculation for micro-strain on the Lebow torque sensor was incorrect. Consequently, instead of applying 200% of the rated load, the gearboxes were experiencing only 77%. Thus the preceding analysis applies for this much lower load. Once the discrepancy was found (after 179 hours of operation), the gearboxes were subjected to the proper 150% and 200% load for 1 hour in accordance with the shakedown procedure. Inspection of the gear meshes did not reveal any adverse effects on the teeth. The contact pattern for the high speed gear set in both boxes improved with the increased load, and the noise within the Convoloid gearbox decreased. Both low speed meshes showed signs of distress due to edge loading.

The gearboxes showed acceptable contact, therefore the test protocol was begun anew using the proper 200% load. After 21.85 hours of operation at the true 200% load (approximately 2.4 million cycles on the high speed pinion), micropitting on the involute high speed pinion is evident on all teeth (Figure P.10). Additionally, there are signs of distress due to edge contact near the center of the bearing span. The low speed pinion is showed signs of distress along both edges, and in the dedendum (Figure P.11). The high speed gear and low speed gear show minimal wear.

The Convoloid high speed pinion showed a narrowing of the transition zone due to a slight widening of the micropitting (Figure P.12). Full contact is being made, as the bluing now is beginning to wear on the side nearest the bearing. Some distress due to edge loading was visible on the opposite side of the face width.

The Convoloid high speed gear is showing light micropitting around the transition zone, with the heaviest in the dedendum (Figure P.13). There still is a significant portion of the profile (at the tip) that is not making contact. This fact causes a higher than expected stress in other areas of contact.

The Convoloid low speed pinion shows micropitting advancing along the face width (Figure P.14). Distress is evident at both edges and at the tip. The low speed gear is showing faint signs of distress at along the transition zone and at the bottom of the dedendum (Figure P.15).

After 91.25 hours of operation at the true 200% load (approximately 9.8 million cycles on the high speed pinion), micropitting is progressing slowly on the involute high speed and low speed pinions. No wear is evident on the gears. The increase in micropitting seems to correspond with an increase in noise from the involute gearbox. Figure P.16 shows a close-up of the micropitting in the high speed pinion.

Slow growth of micropitting on the Convoloid high speed pinion is narrowing the width of the transition zone. Otherwise there is no dramatic change in the micropitting. Of special interest is that, during run-in, marks appeared on the tooth flanks due to debris in the oil. Most of these have not precipitated wear, but one mark seems to be leaving an impression on the mating flanks of the gear (Figure P.17). These marks later began to fade, removing concern that they would precipitate an unwanted failure. Noise from the Convoloid gearbox continued to decrease, and was comparable with the involute gearbox at this time.

Micropitting is consistently evident on the high speed gear at the dedendum side of the transition zone (Figure P.18). The photo in Figure P.18 also shows very well the area of no contact at the tooth tip due to excessive tip relief. This area represents a significant portion of the contacting flank, yet is not carrying any load, causing the rest of the tooth flank to see higher stresses than calculated.

At the conclusion of testing, the gearboxes had been subjected to approximately 179 hours of the 77% load and 235 hours of the true 200% load. Significant micropitting is evident in the involute high speed pinion near the root and along one edge (Figure P.19, Figure P.20) and appears to expand at a significant rate. This wear appears to have caused a degradation of the profile leading to a significant increase in noise from the involute gearbox. Conversely, the high speed gear shows little wear (Figure P.21). The low speed pinion shows light micropitting around the edges and at the peaks of grinding grooves (Figure P.22). As with the high speed gear, the low speed gear shows minimal wear (Figure P.23). Vibration levels within the involute gearbox increased approximately 6% during the time of the 200% test.

The Convoloid gear sets show definite micropitting on all tooth surfaces around the transition zone. However, in contrast to the rapidly spreading wear of the involute high speed pinion, the micropitting seems to have retarded its advance during the last 100 hours of the test. The Convoloid high speed pinion shows distinct micropitting on both sides of the transition zone (Figure P.24), but there still is a definite area of no contact in this region, as is expected by the Convoloid geometry. The high speed gear shows slight micropitting wear along the dedendum side of the transition zone (Figure P.25). The addendum side remains free of wear. This figure also shows the large area of non-contact.

The low speed pinion shows distinct micropitting around all edges, except in the root (Figure P.26). This "picture frame" phenomenon requires additional study. The low speed gear also shows faint signs of micropitting except at the tip (Figure P.27). Some shades of bluing remain at the tip of the low speed gear, but it is evident that contact occurred here. Thus, an appropriate amount of tip relief must have been applied. Vibration levels within the Convoloid gearbox decreased approximately 9% during the time of the 200% test.

Oil samples taken from both gearboxes were extremely clean. Towards the end of the test (approximately 165 hours), metal particles were observed in the involute sample (although not at a dangerous level) and they continued to appear until the test was shut down. Also note that there was more entrained air in the involute samples, as compared to the Convoloid samples.

Micon 108 Extended Test (240% load)

(Pictures and figures located in Appendix P.)

Having completed the testing protocol (200 hours at 200% load and 50% faster speed) during December, it was determined that the test results were not definitive. Extensive discussions early in the month resolved that the test should continue until a failure occurred.

To reduce the length of the test, investigations were conducted to determine whether the applied load and the motor speed could be increased. Analysis of the limiting factors (gear tooth life, key stress, and shaft strength) confirmed that and increase of load to 240% of the operating torque was viable. Speed of the test motor was to be increased to as high as 2,700 rpm which increases the number of load cycles in a shorter period of time. An addendum to the test protocol was developed (Appendix M). For reference, 240% of the rated load is 515,000 lb-in torque versus the standard 214,500 lb-in.

Upon completion of the original test, the high speed pinion of the Convoloid gearbox was removed and sent to Brown and Sharpe (B&S) for inspection. Of particular interest was comparing the area of contact with a portion of a tooth where no contact or wear has occurred. On either side of the transition zone, micropitting has removed material, yet the geometry within the zone is identical to the untouched region.

Testing resumed on January 23 and followed the test protocol addendum. Once the oil and bearing temperatures stabilized with a 200% load and 1,768 rpm, the load was increased to 220% and then to 240%. No abnormal increases in temperature were observed and sound levels did not noticeably increase. The rubber cover for the constant velocity coupling nearest the Convoloid gearbox, however, was observed to slide away from the mounting shoulder. The test was halted and the cover was moved back to its proper position. After the test was restarted the coupling appeared to run smoothly.

After confirming that the system was stable, motor speed was increased to 2,000 rpm. As the speed increased, noise from both gearboxes seemed to increase. The involute gearbox exhibited a higher frequency due to a higher number of teeth on its high speed pinion (16) compared to the Convoloid (13). Oil levels in both gearboxes did not seem to be affected by the increase in speed and bearing temperatures were satisfactory.

The motor speed was increased to 2,150 rpm. Evidently, the frequency of the vibrations from the involute high speed gear mesh were close to a system harmonic, because the sound from the involute gearbox increased dramatically, requiring the test personnel to wear hearing protection. The torque arm structure is different because the involute drive in that the motor also must be supported, and this structure could act as a sounding board. After the speed was increased to 2,350 rpm, the noise level—although at a higher frequency—dropped. Due to concerns with the coupling no additional speed increases were conducted.

After approximately 1 ½ hours at the new test speed of 2,350 rpm (at 240% load), the rubber coupling cover nearest the Convoloid gearbox failed. The test immediately was shut down and examination of the coupling revealed a 1 ½-in wide, 6-in long strip of the coupling that had been removed. The coupling manufacturer emphatically stated that the rubber cover was required for operation. A replacement cover was found and shipped to a repair facility in Portland, Oregon. After the coupling was repaired and the test equipment was reassembled, the test was resumed.

The manufacturer indicated a maximum speed of 2,000 rpm for the test coupling. Investigations into prior communications with the coupling manufacturer did not reveal any speed limitations for the coupling, although previous discussion had been focused on operation at 1,800 rpm. Due to the speed limitations of the coupling, the test was restarted with an operation speed of 2,000 rpm and 240% load.

After 70 hours of operation (302 total test hours), micropitting on the involute high speed pinion teeth (Figure P.28) appeared to have spread in the pinion root and along one edge (*c.f.* Figure P.19). Wear on the Convoloid teeth (Figure P.29) did not appear much changed from the end of the previous test (*c.f.* Figure P.24).

Vibration levels in both gearboxes appear to be changing as the test progresses. The involute gearbox is seeing a gradual increase in vibration. Conversely, vibration in the Convoloid gearbox is slowly decreasing. Spectrum analysis of these vibrations reveals a substantial difference in the two gearboxes. High speed mesh frequencies at 2,000 rpm are 533 Hz for the involute and 433 Hz for the Convoloid. Figure P.30 and Figure P.31 compare the high speed mesh frequency for the involute and Convoloid respectively. The fundamental frequency for the high speed mesh is 19.2 dB higher with the involute gearbox. Additionally, an audio sound spectrum shows a very distinct difference in noise level between the two gearboxes (Figure P.32). The involute gearbox is 12.8 dB higher than the Convoloid gearbox, with the primary noise occurring at the high speed gear mesh frequency.

Mr. Williston of Genesis investigated the speed limitations of the constant velocity (CV) joint that ties together the high speed shafts of the two gearboxes. Discussions with Mr. Fred Standfest of Universal Technical Services (UTS) who was instrumental in the design selection of the CV couplings revealed that the mechanical components of the coupling were viable up to 5,000 rpm. However, the joint itself is limited by the rubber "boot" that retains the lubricating grease. Mr. Standfest was comfortable with a 50-rpm increase in speed, but cautioned that any additional speed increases should be closely monitored—especially coupling temperature, as this directly affects the solidity of the boot. Subsequent testing at increased speeds revealed that the involute side coupling boot began to "billow" or expand at speeds as low as 2,100 rpm. To prevent a potential coupling failure, motor speed was reduced to 2,050 and testing progressed at this speed.

Audible noise emanating from the test apparatus slowly rose to approximately 92 dB by mid-month and remained at this level through the end of the month. Vibration and noise levels continued to be closely monitored to provide early warning signs for impending failure. The computer monitoring system was designed with warning and alarm points for excessive temperature in any bearing, high particulate counts, or excessive vibrations. Spectrum analysis of vibrations from each gearbox allowed monitoring specific frequencies and harmonics of the drive system. As discussed in previous reports, amplitude levels for the involute high speed gear mesh are significantly higher than the Convoloid.

One key vibration appeared mid-month that was not tied to any mechanical device in either gearbox. This vibration (68.3 Hz) was exactly two times the high speed shaft rotational frequency (34.15 Hz at 2,050 rpm). Upon closer inspection, audible noise relating to this vibration was thought to be coming from the involute side CV joint. This joint has not had any alteration and had the original rubber boot. These couplings are not designed for field maintenance, and after discussion with the manufacturer it was deemed that no action was necessary. On March 1, 2006, noise levels and unexplained fluctuations in an intermediate shaft bearing temperature in the Convoloid gearbox prompted the test monitor to shut down the test. Subsequent inspections revealed a failure of one of the intermediate bearings in the Convoloid gearbox.

It is important to note that the Convoloid gearing did not show a progression in wear during the last 200 hours of operation. Conversely, the involute gearbox experienced a continual progression in micropitting along the flanks of both pinions. The following figures show the relative wear from the initial contact patch, to conclusion of the 200-hour test, to the last inspection prior to the bearing failure.

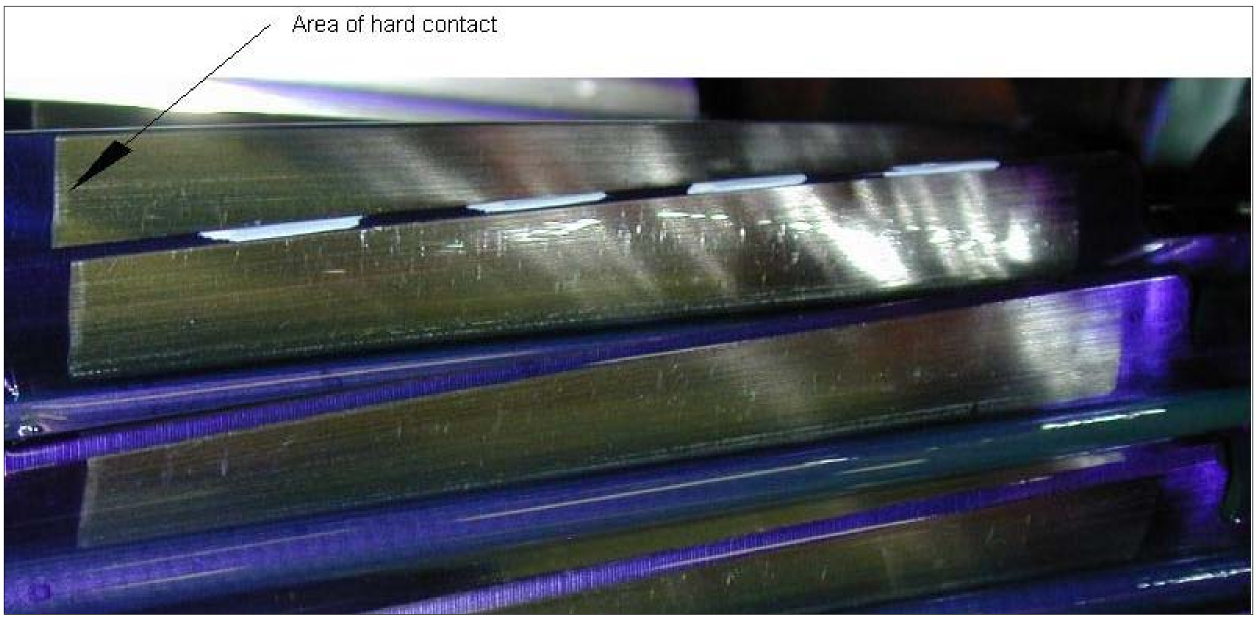


Figure O.1. Involute high speed pinion—initial 200% loading; nearly full contact as the bluing has been nearly uniformly removed; hard contact is visible on the left side of the contact patch; paint marks on the tooth tip are used for identification to verify the same tooth was view in subsequent inspections

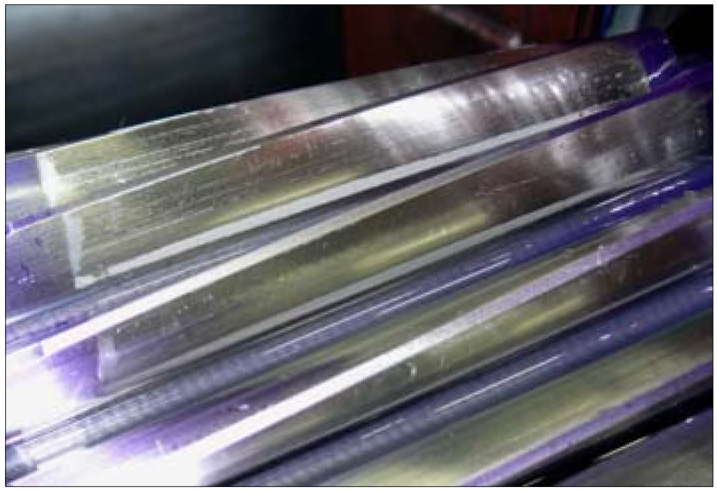


Figure O.2. Involute high speed pinion—235 hours at 200% load; micropitting (dull gray area) in dedendum and at the edge of contact

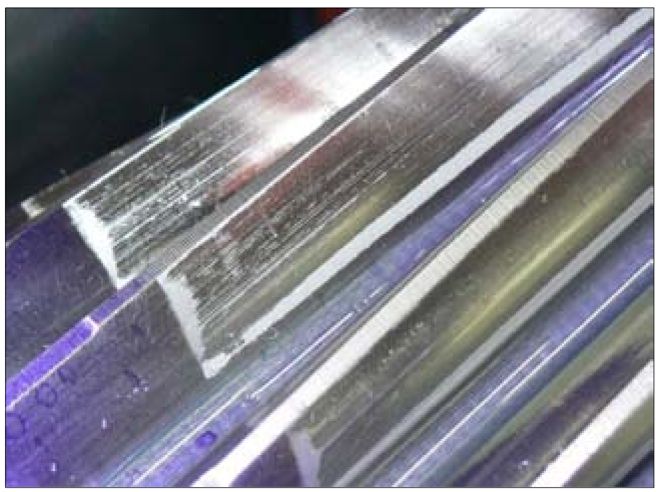


Figure O.3. Involute high speed pinion—235 hours at 200% load; micropitting (dull gray area) in dedendum and at the edge of contact

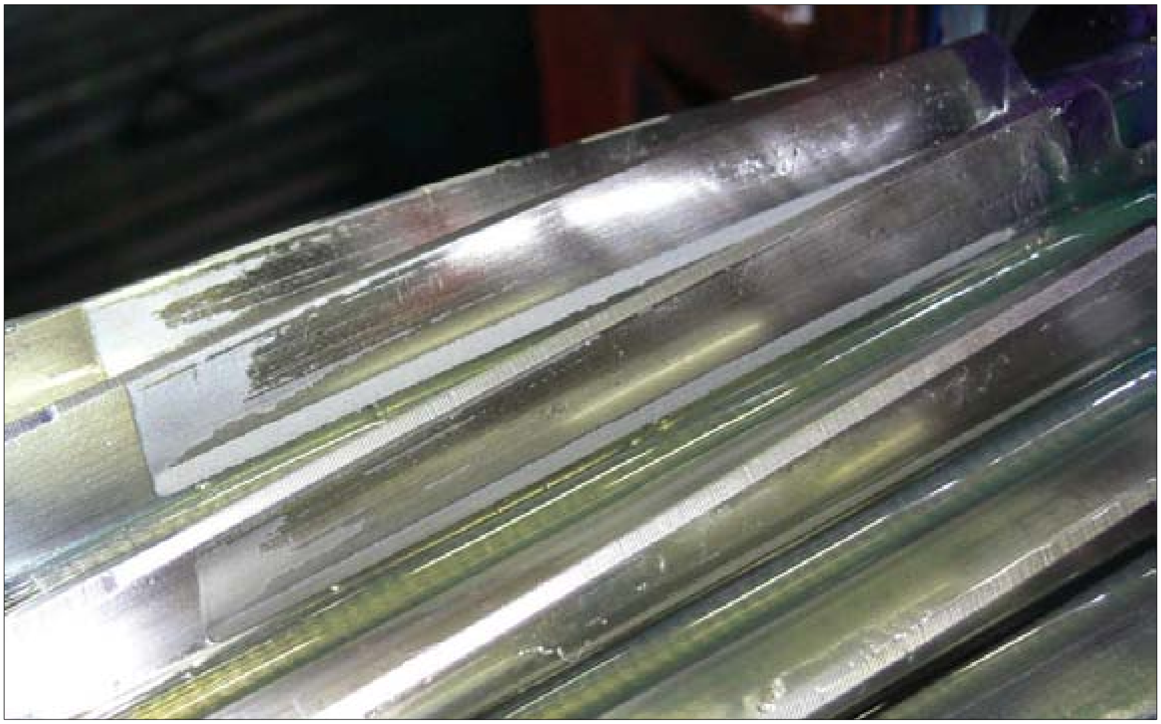


Figure O.4. Involute high speed pinion showing extensive micropitting; 763 hours total test time (89 x 10^6 cycles). (235 hours at 200% load and 1,767 rpm motor speed, then the load was increased to 240% and speed increased to 2,050 rpm)

Wear on the mating involute high speed gear shows minimal wear in comparison with run-in.

Figure O.5 Involute high speed gear at shakedown

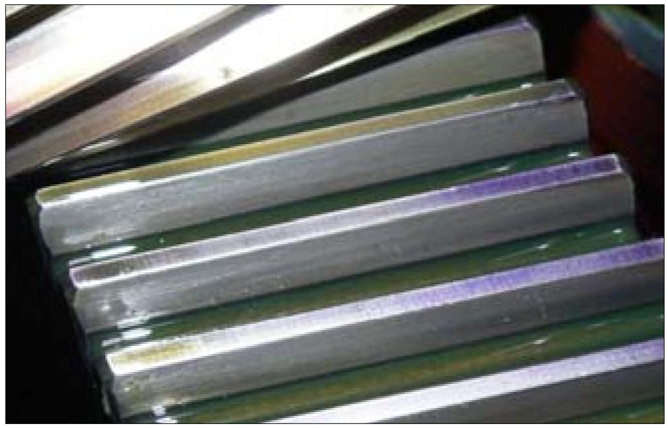


Figure O.6. Involute high speed gear at shakedown after 763 hours; minimal wear is visible

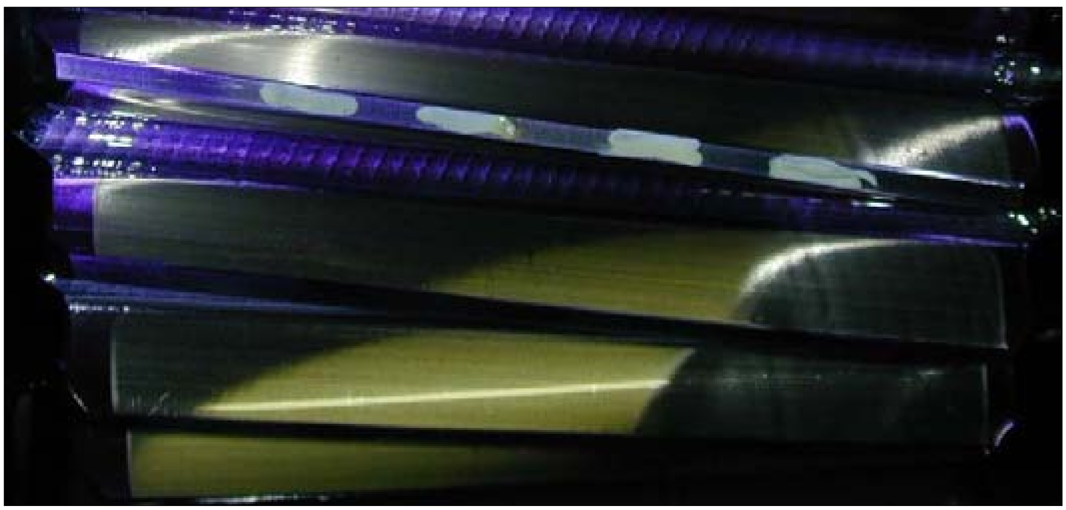


Figure O.7. Involute low speed pinion at 200% shakedown; the contact patch reveals a consistent contact pattern across the contact flank; note paint marks on teeth tips for inspection purposes

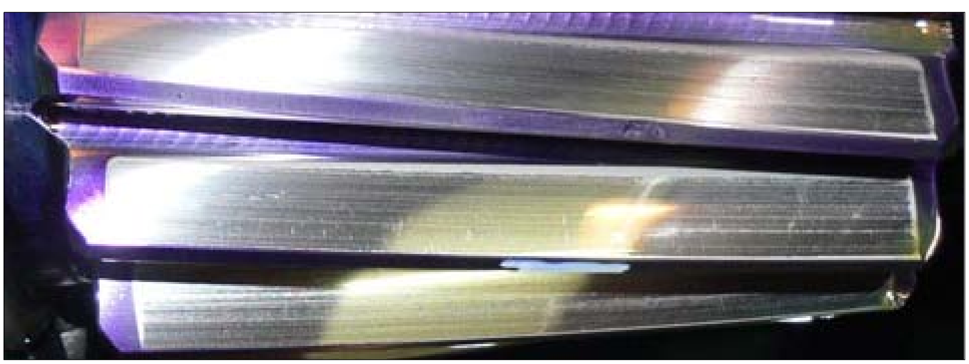


Figure O.8. Involute low speed pinion after 235 hours at 200% rated load; light micropitting is visible on all four edges of contact with a heavier concentration in the dedendum; faint micropitting also is visible on the grinding peaks (faint lines that run horizontal in the active flank)

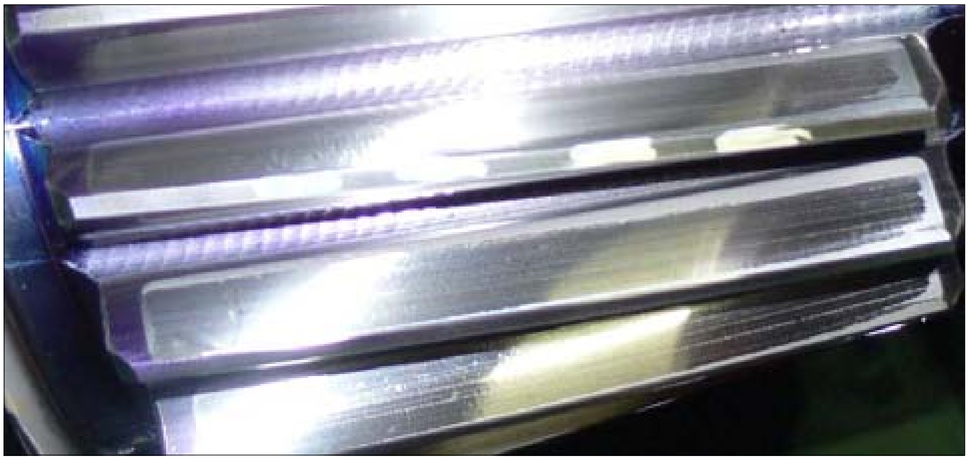


Figure O.9. Involute low speed pinion after 763 total test hours; extensive micropitting is visible in the dedendum and at the right side

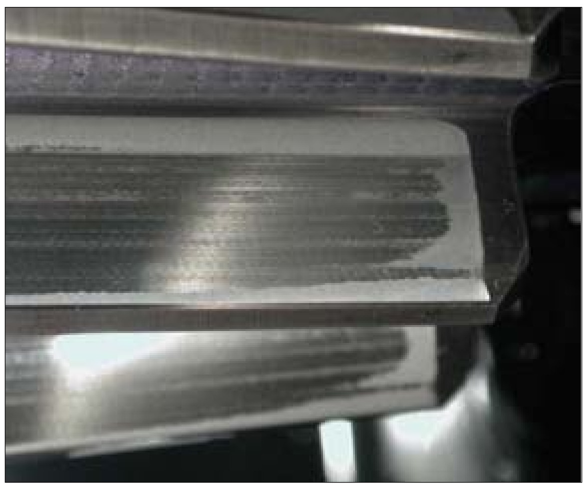


Figure O.10. Close-up of micropitting on the right side of the involute low speed pinion after 702 total test hours

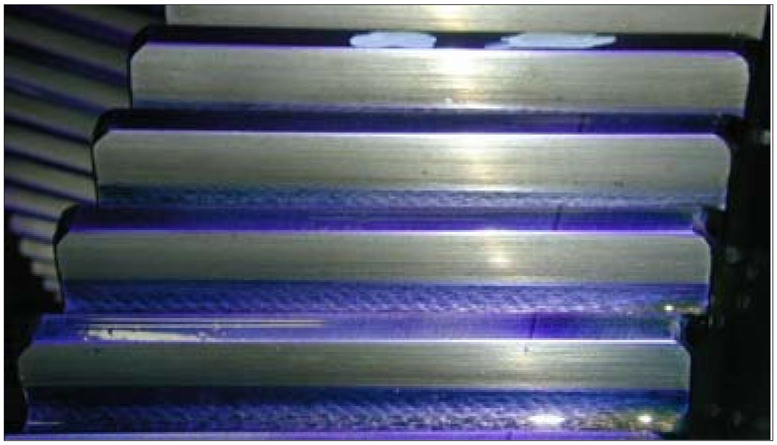


Figure O.11. Involute low speed gear after 200% shakedown

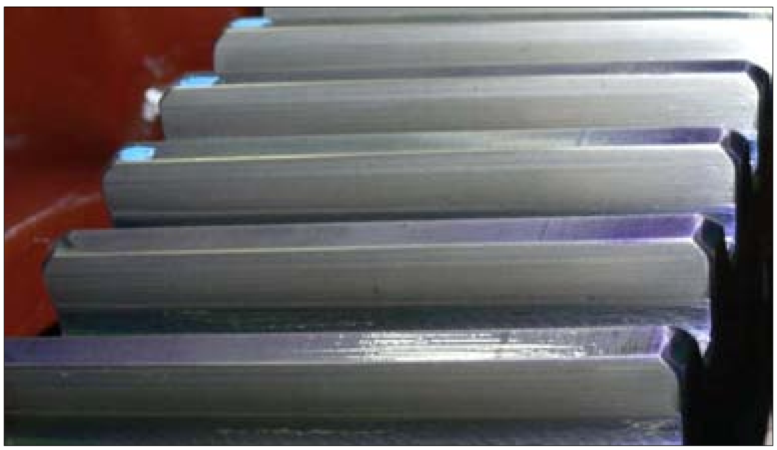


Figure O.12. Involute low speed gear after 763 total test hours; minimal wear is evident

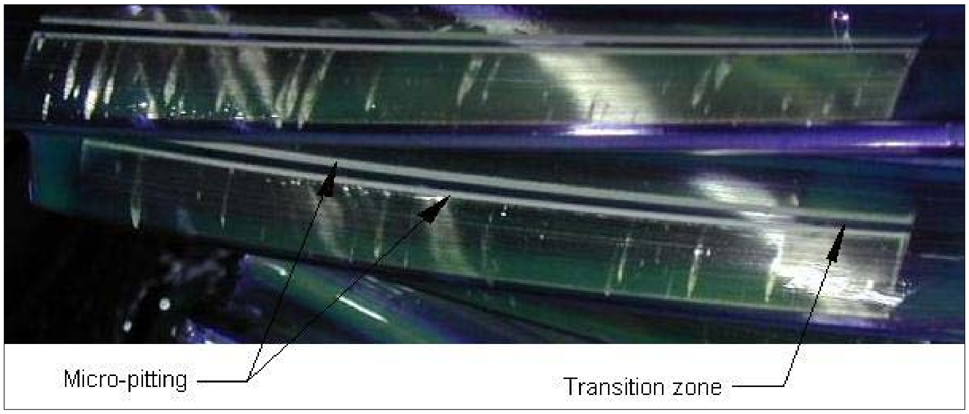


Figure O.13. Convoloid high speed pinion after shakedown of 1 hour operation at 200% rated load; micropitting on either side of the transition zone appeared during approximately 160 hours of testing at 77% rated load; radial marks are due to aluminum particles from the heat exchanger that passed through the gear mesh; no impression could be felt nor was there any damage to the tooth flank

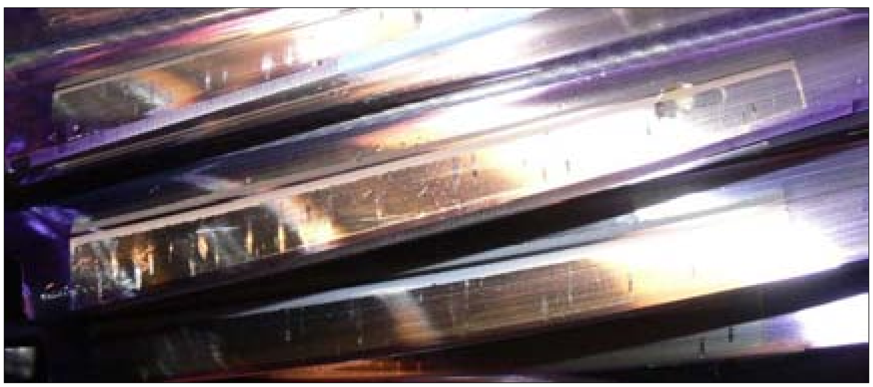


Figure O.14. Convoloid high speed pinion after 235 hours at 200% rated load

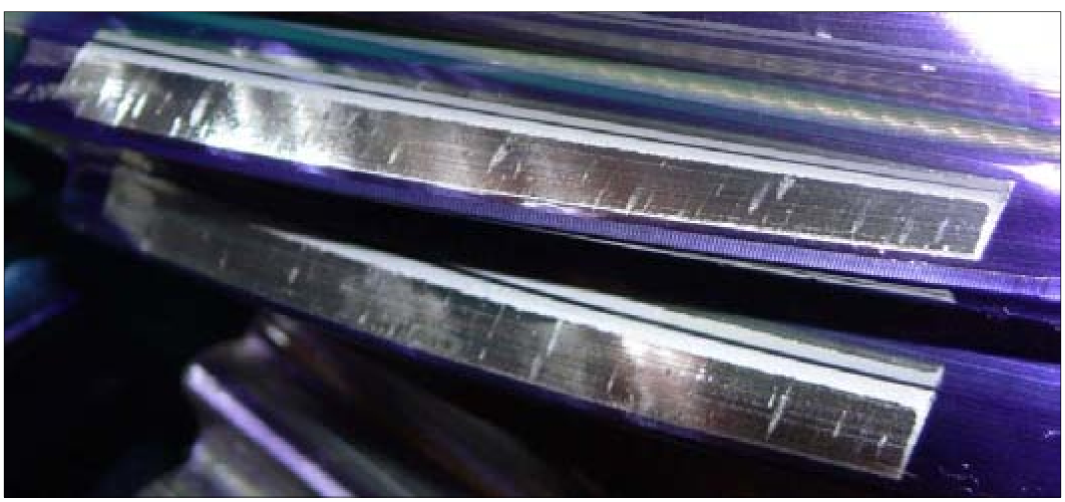


Figure O.15. Convoloid high speed pinion after 360 total test hours; micropitting around the transition zone is broader than after shakedown

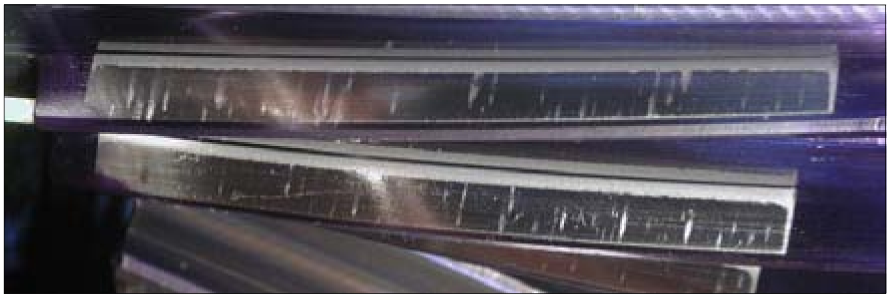


Figure O.16. Convoloid high speed pinion after 538 total test hours; micropitting around the transition zone shows minimal progression

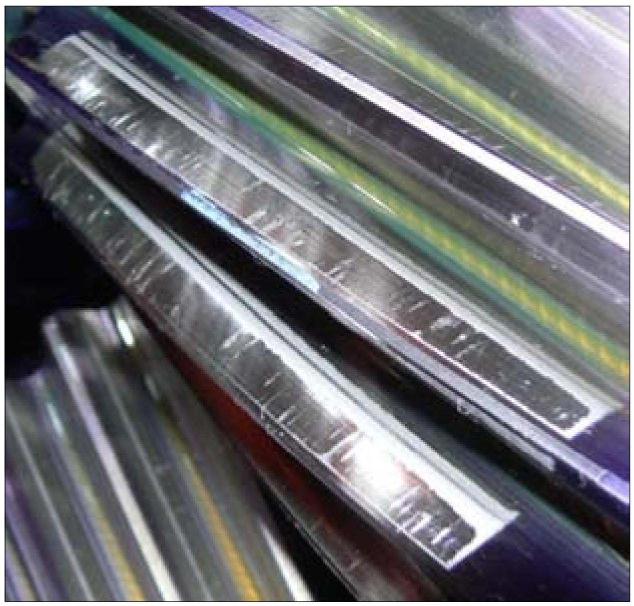
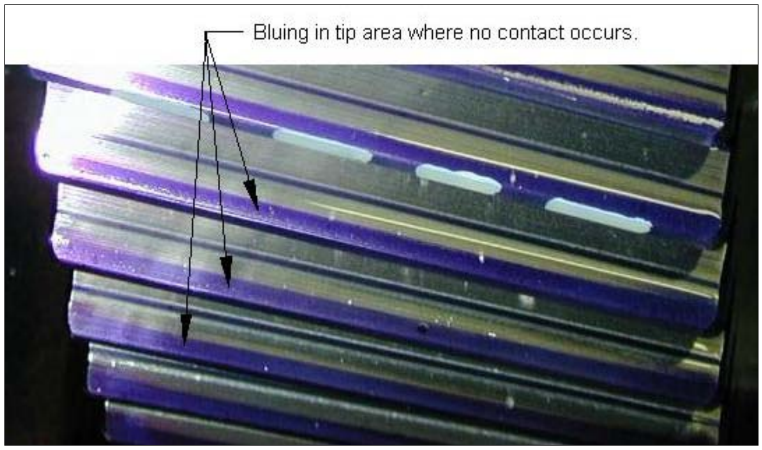


Figure O.17. Convoloid high speed pinion after 763 total test hours; micropitting zones on either side of the transition zone have broadened



Figures O.18. Convoloid high speed gear at 200% shakedown

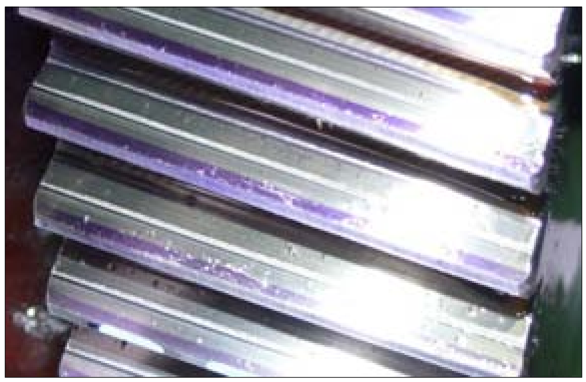


Figure O.19. Convoloid high speed gear at 200% shakedown after 763 total test hours; micropitting on the dedendum side of the transition zone corresponds with wear in the mating pinion; note the large amount of bluing still visible at the tooth tips

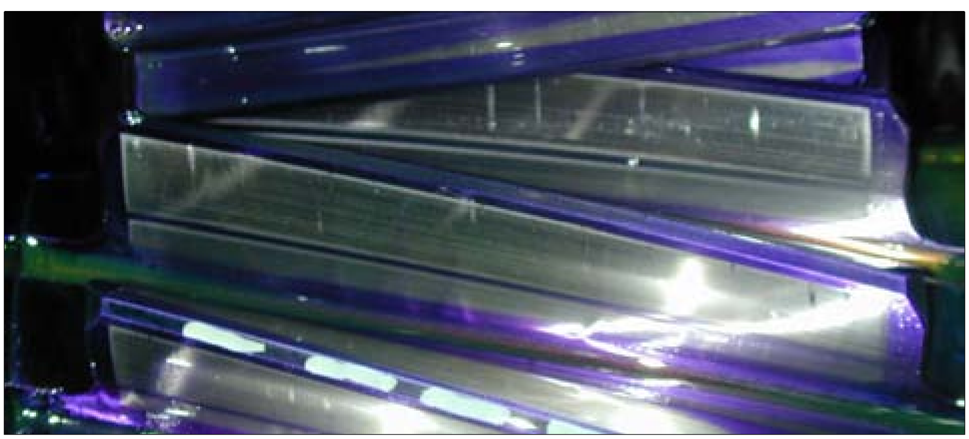


Figure O.20. Convoloid low speed pinion after shakedown of 1 hour operation at 200% rated load; micropitting at the either side of the transition zone appeared during approximately 160 hours of testing at 77% rated load

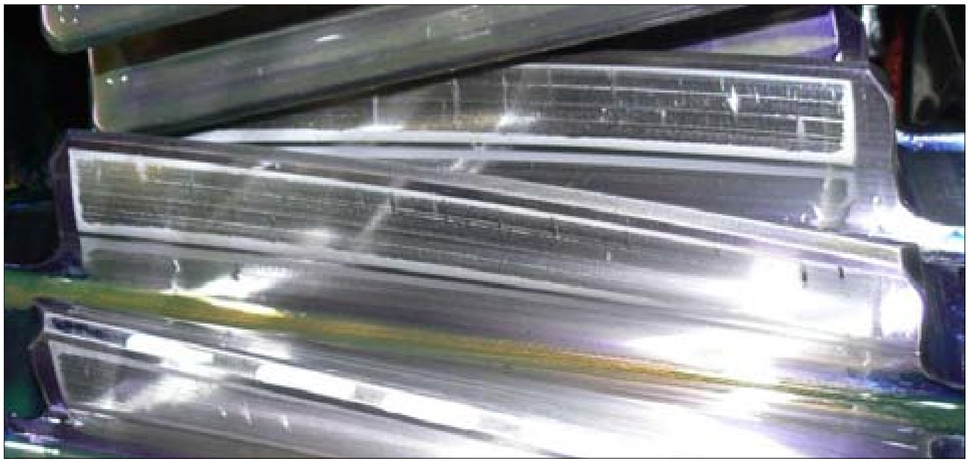


Figure O.21. Convoloid low speed pinion after 763 total test hours

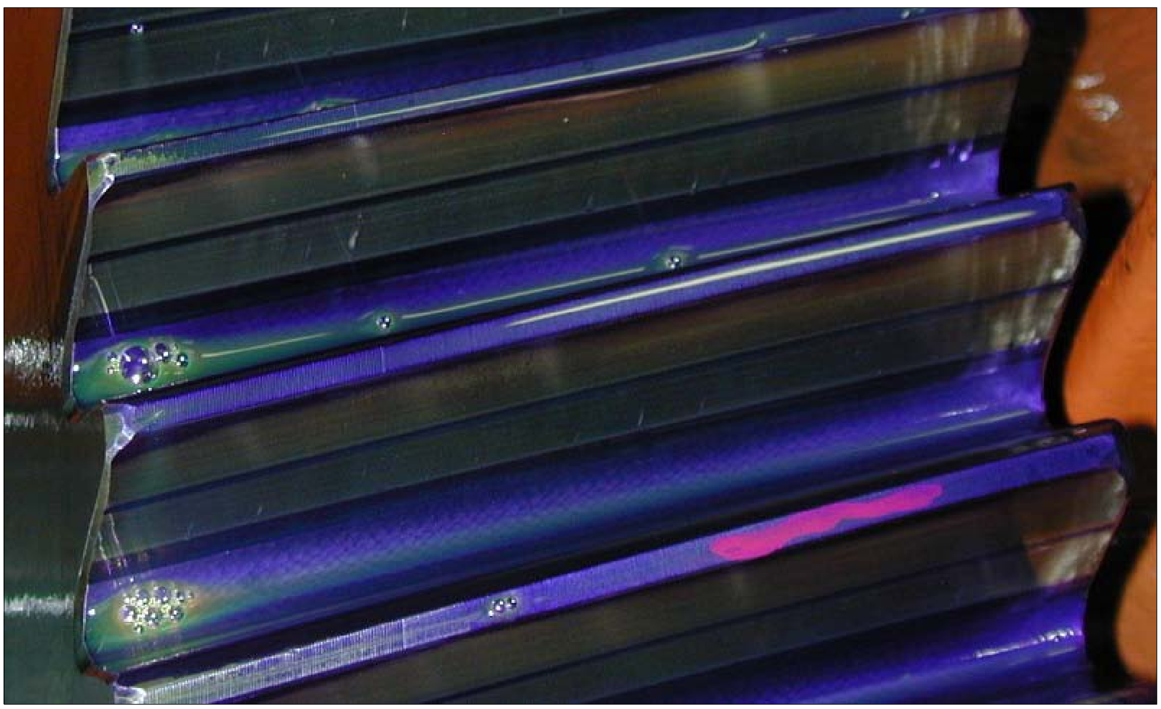


Figure O.22. Convoloid low speed gear after 200% shakedown

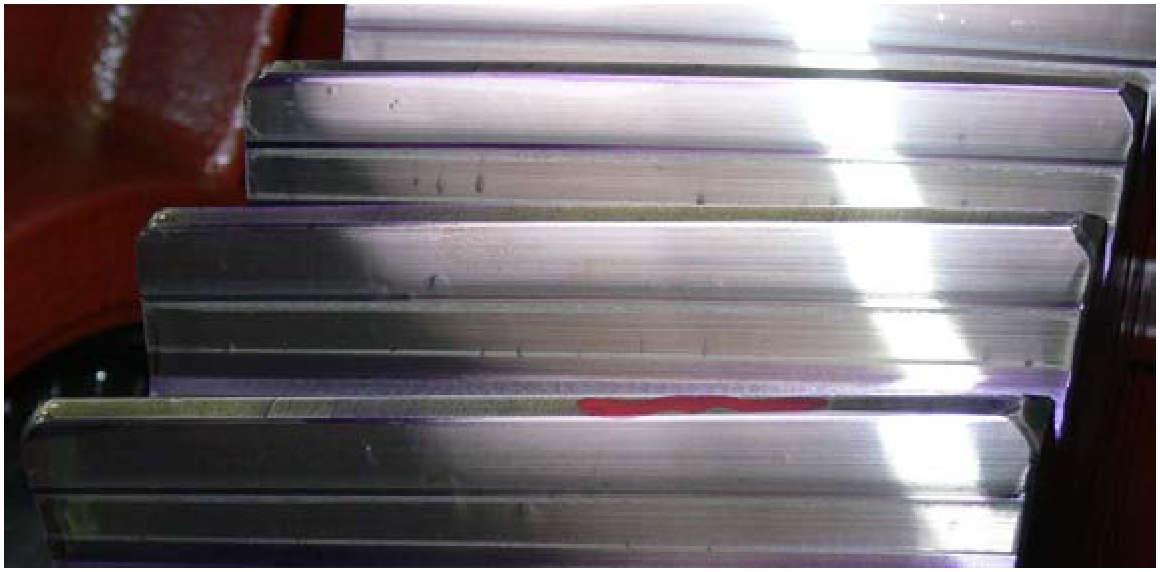


Figure O.23. Convoloid low speed gear after 763 total test hours; light micropitting is visible at the edges of the dedendum side of the transition zone; no micropitting is evident at the tip, nor is there much bluing

On March 1, 2006, noise levels and unexplained fluctuations in an intermediate shaft bearing temperature in the Convoloid gearbox prompted the test monitor to shut down the test. Significant damage was present on the rollers and cage (Figure O.24). Pieces of the cage and roller material passed through gear meshes and other bearings (Figure O.25). Significant quantities of metal were recovered in the gearbox sump, oil reservoir, and primary filter. A day prior to the bearing failure, magnet inspection of both gearbox sumps did not reveal any metal particles. After the failure, extensive quantities were found inside the gearbox housing.

Analysis of temperature and vibration data taken during the final day of operation, shows a distinct increase in vibration levels approximately 8 hours before shutdown (Figure O.26, Figure O.27). An observation by the operator that was written in the test log at the time of the increased vibration indicates that there was “a change in the sound” of the Convoloid gearbox. However, testing continued until a more dramatic failure occurred.

Figure O.24. Failed spherical roller bearing in Convoloid gearbox after 763 total test hours; note pitted edges of the rollers and deformation of the cage; metallic particles are present at the bottom of the housing bore

A meeting was conducted at The Gear Works (TGW) in Seattle, Washington, to discuss the test results. In summary, testing of the existing gearboxes was halted pending analysis of the bearings and feasibility of correcting damage due to debris in the gear mesh. Harry Halloran of EUI emphasized that completion of a laboratory test showing a definitive comparison of the two gearing types was imperative. The amount of time in test: 179 hours at 77% of rated load; 235 hours at 200% load; 528 hours at 240% load; total test time: 942 hours.

Figure O.25. Pieces of the failed bearing

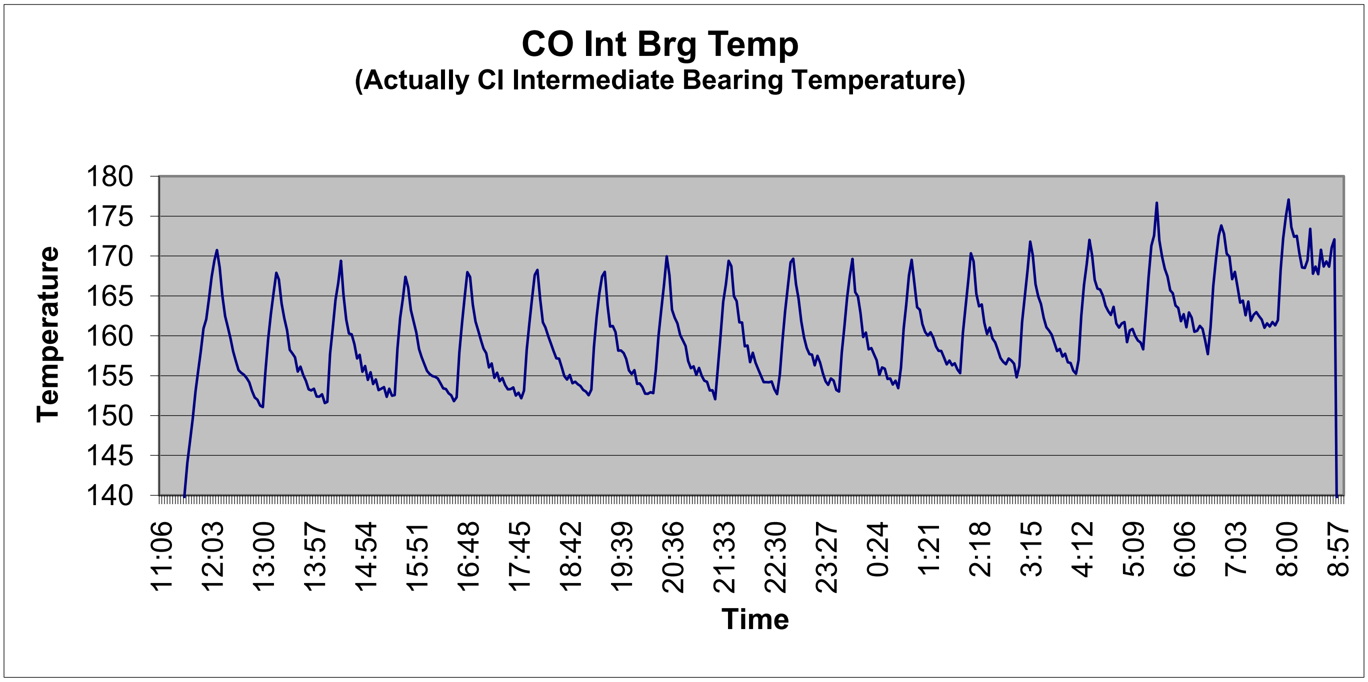


Figure O.26. Bearing temperature chart for the failed Convoloid intermediate bearing; note the rise in temperature after 2:18 a.m.; the jagged "saw-tooth" shape is due to cycling of the heat exchanger fan

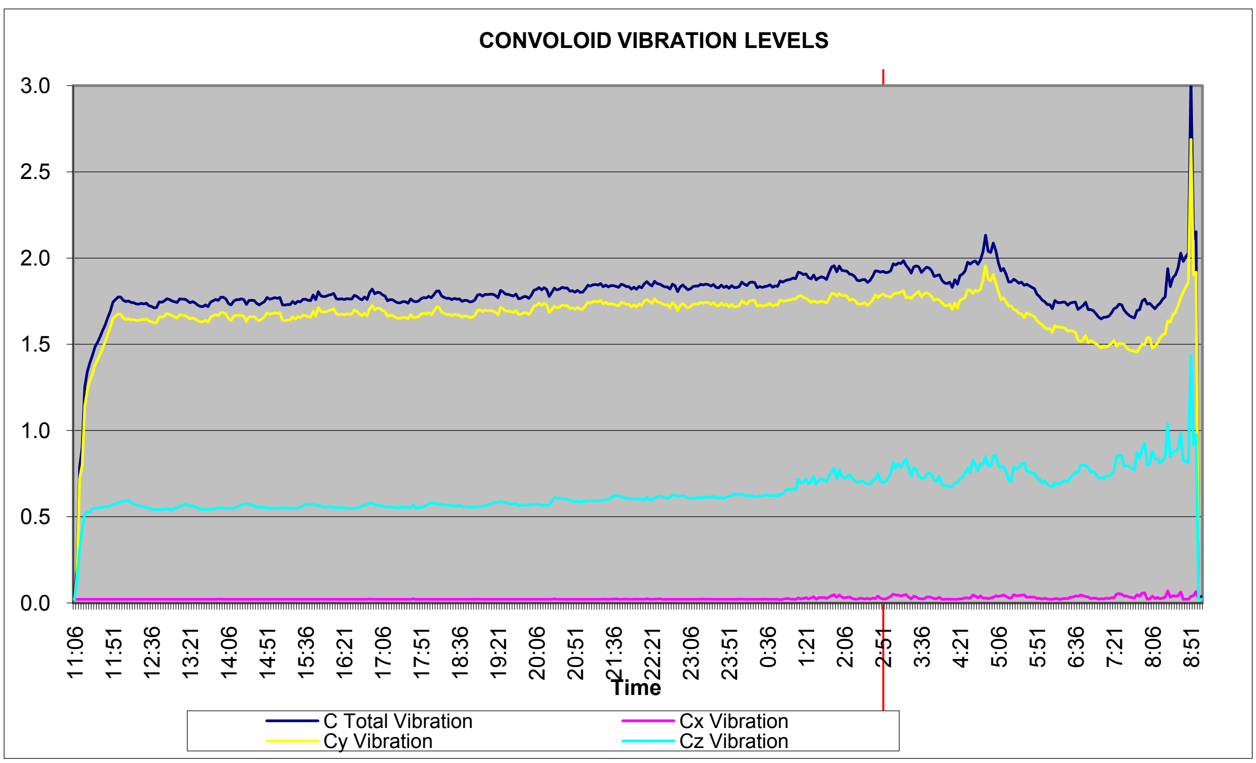


Figure O.27. Vibration levels for the Convoloid gearbox for the last day of operation; note the increase in vibration at approximately 1:00 a.m.

Appendix P. Micon 108 200% x 200-Hour Detail of Test Figures and Photos

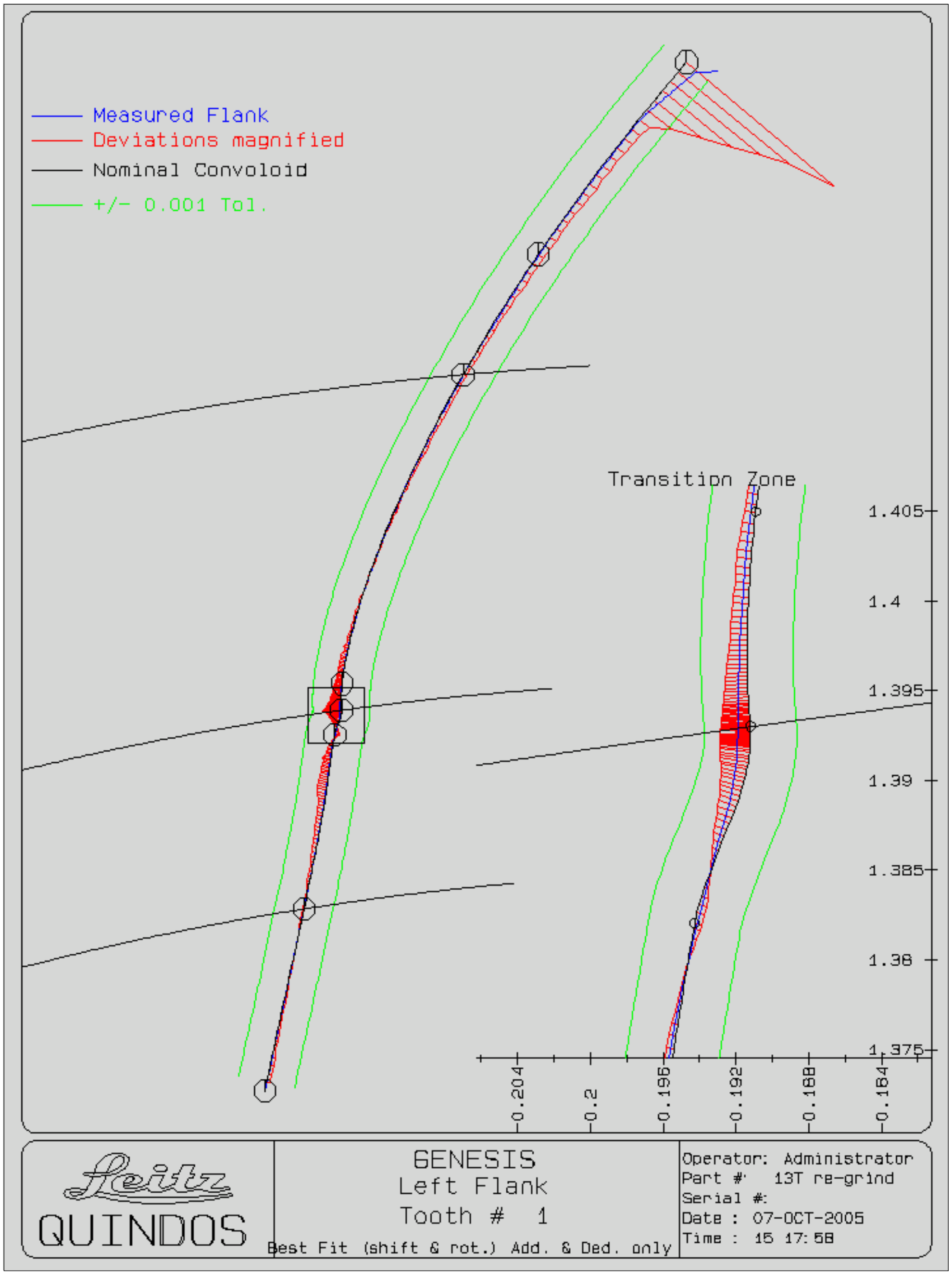


Figure P.1. Profile plot of Convoloid high speed pinion showing very small variations; the top of the pinion shows the tip relief

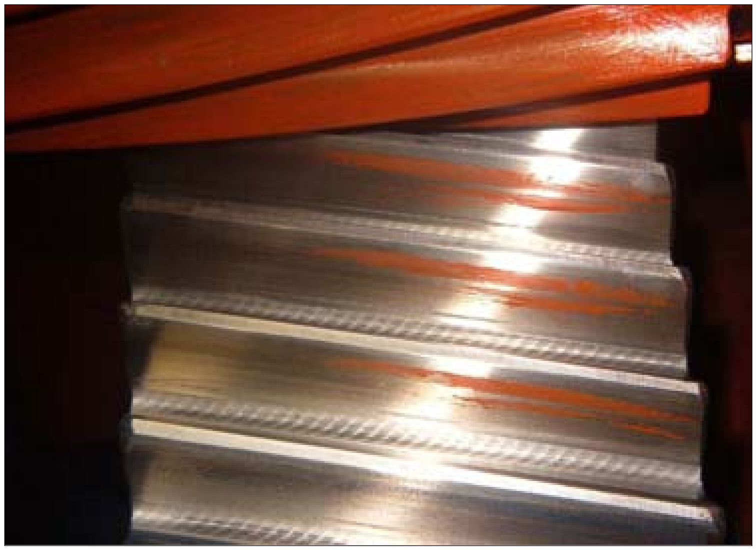


Figure P.2. Convoloid High speed gear mesh no-load contact patch prior to lead modification; gearing has been assembled into the test housing

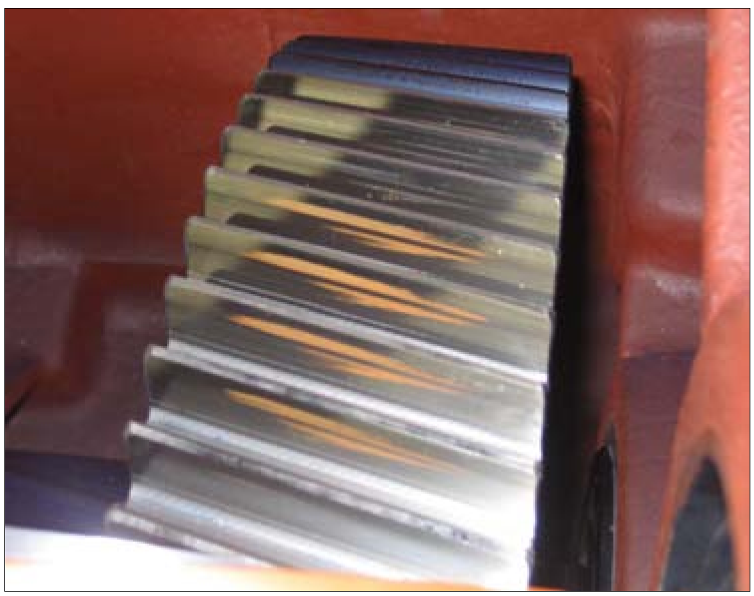


Figure P.3. Convoloid high speed gear mesh no-load contact patch after lead modification

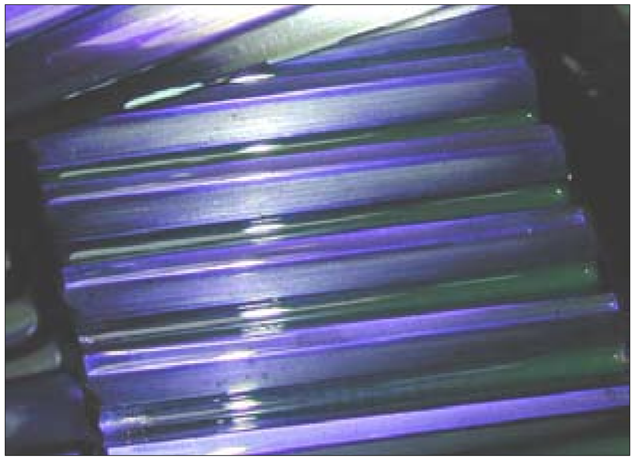


Figure P.4. HS Involute, 77% shakedown contact

Figure P.5. LS Involute, 77% shakedown contact

Figure P.6. HS Convoloid, 77% shakedown contact

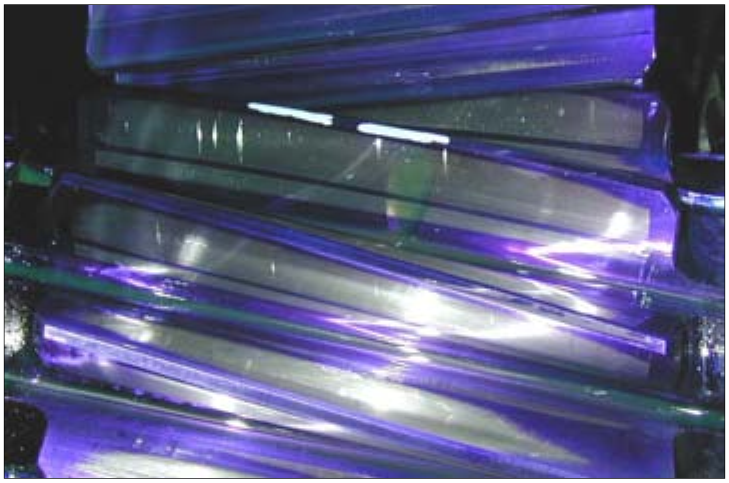


Figure P.7. LS Convoloid, 77% shakedown contact

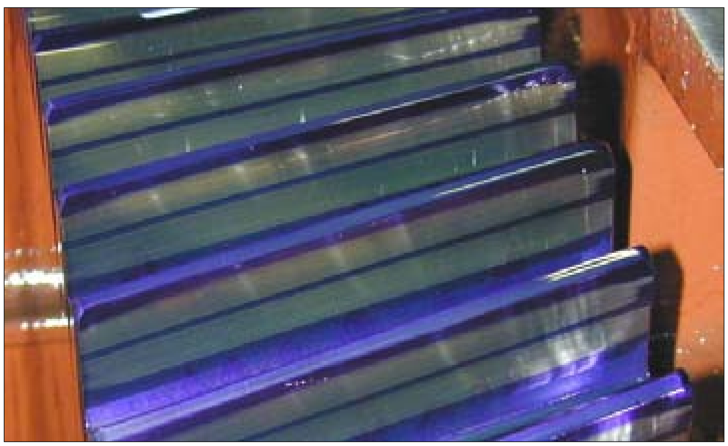


Figure P.8. LS Convoloid gear showing “twist”; note the blue area at the right side of the addendum and at the left side of the dedendum (very faint)

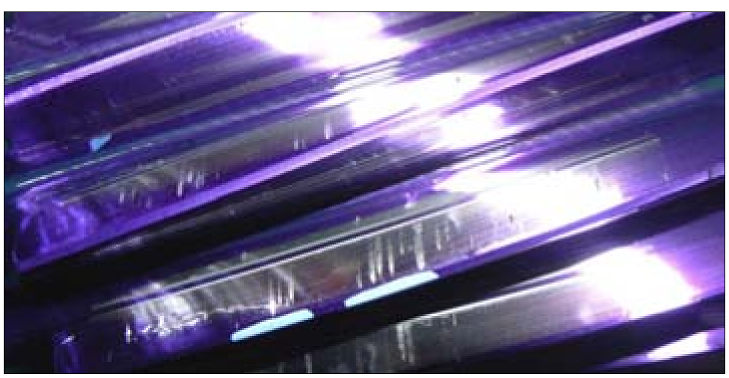


Figure P.9. HS Convoloid pinion showing micropitting along the transition zone (63.7 hours—(approx. 6.8 million cycles) at 77% load)

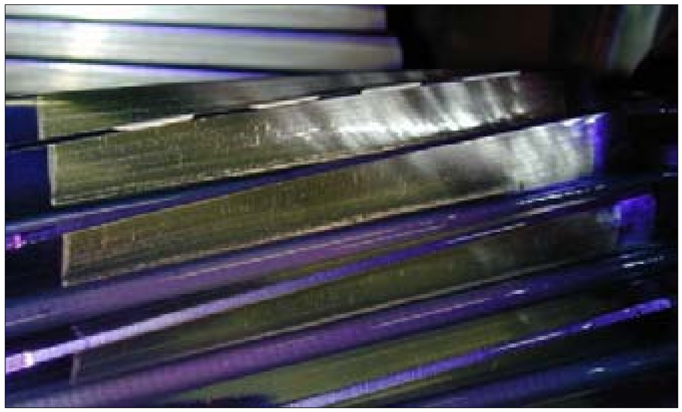


Figure P.10. Involute HS pinion with micropitting in the dedendum and edge distress (true 200% load after 21.9 hours)

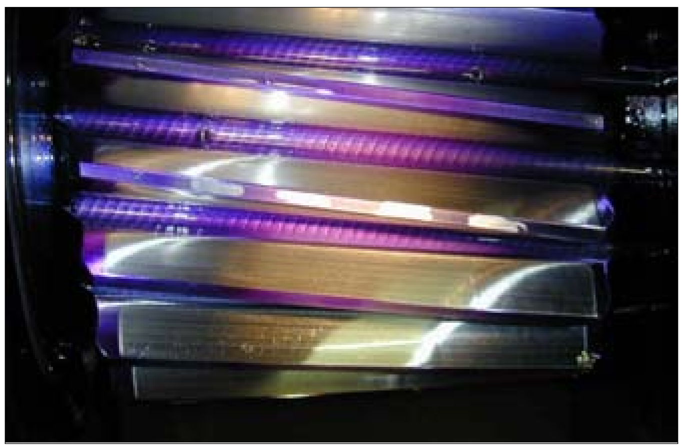


Figure P.11. Involute LS pinion showing edge distress (21.9 hours at 200% load)

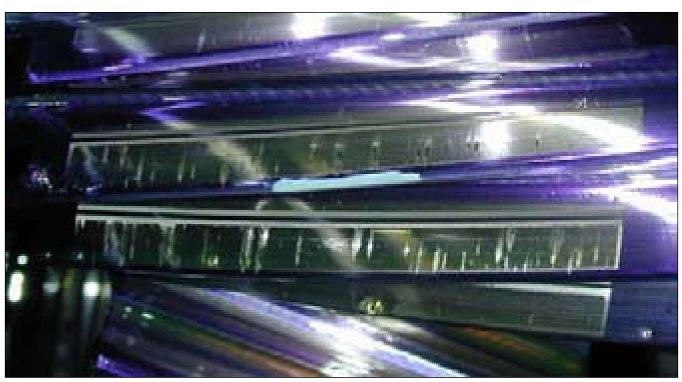


Figure P.12. Convoloid HS pinion with full face contact, showing transition zone micropitting and edge distress (21.9 hours at 200% load)

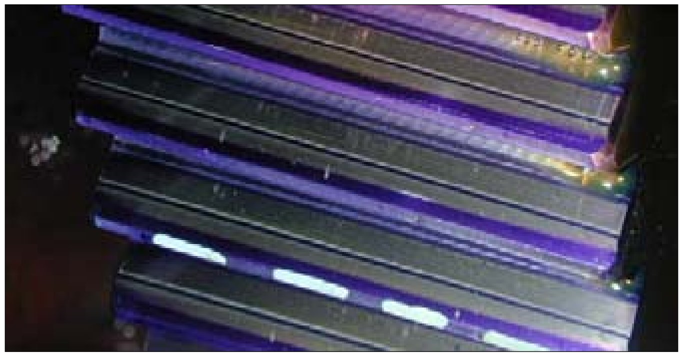


Figure P.13. Convoloid HS gear showing slight micropitting (21.9 hours at 200% load)

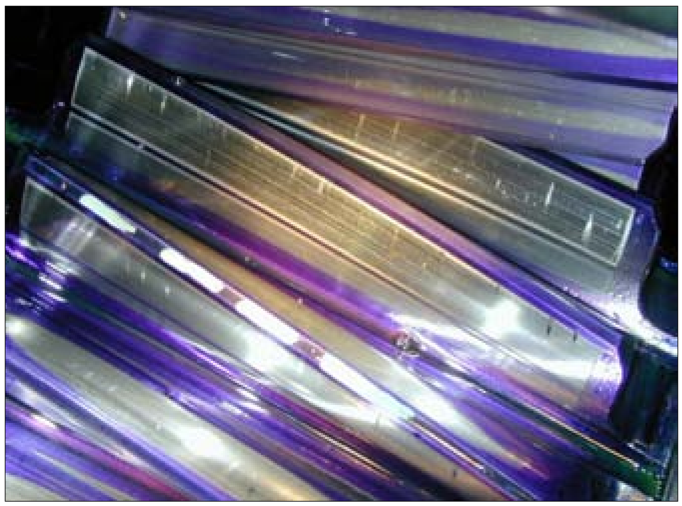


Figure P.14. Convoloid LS pinion showing micropitting (21.9 hours at 200% load)

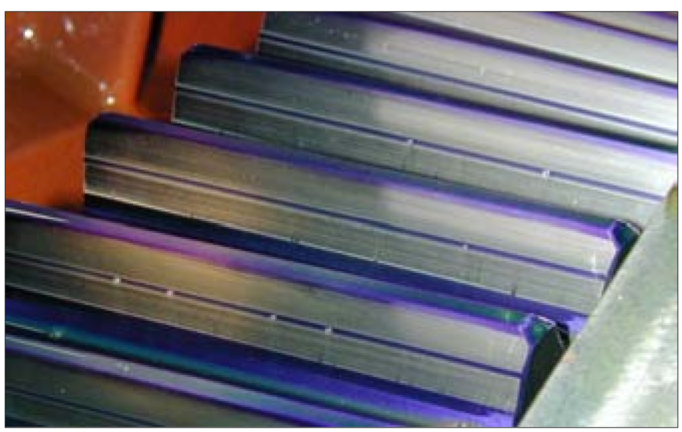


Figure P.15. Convoloid LS gear with faint signs of distress(21.9 hours at true 200% load)

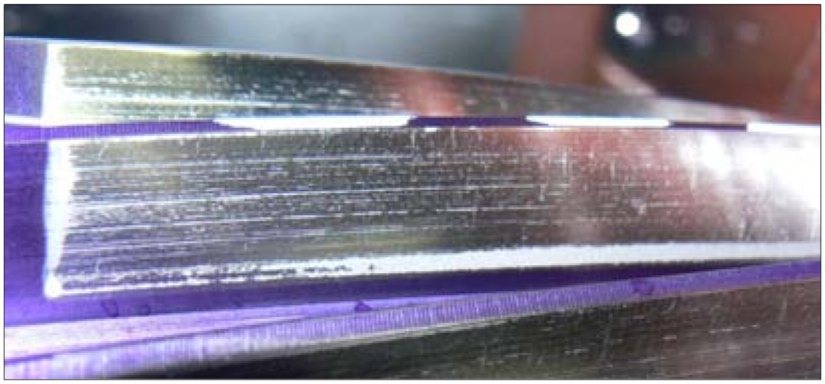


Figure P.16. Close-up of the Involute HS pinion micropitting (91.2 hours at 200% load)

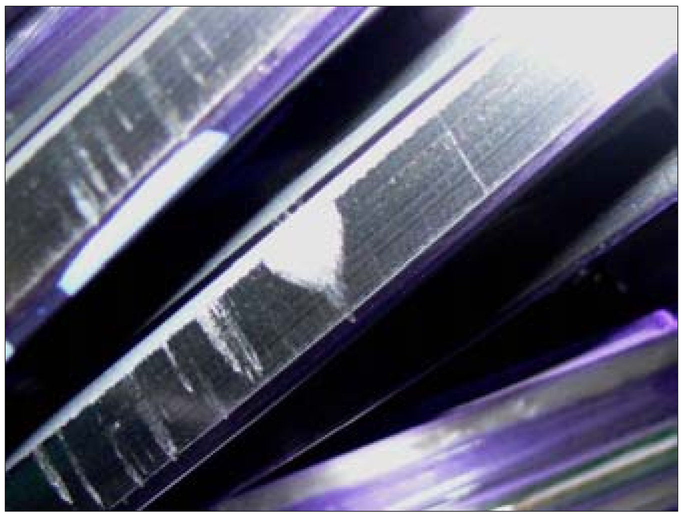


Figure P.17. Close-up of oil-debris damage on the Convoloid HS pinion; this mark began to fade at the next inspection (91.2 hours at 200% load)

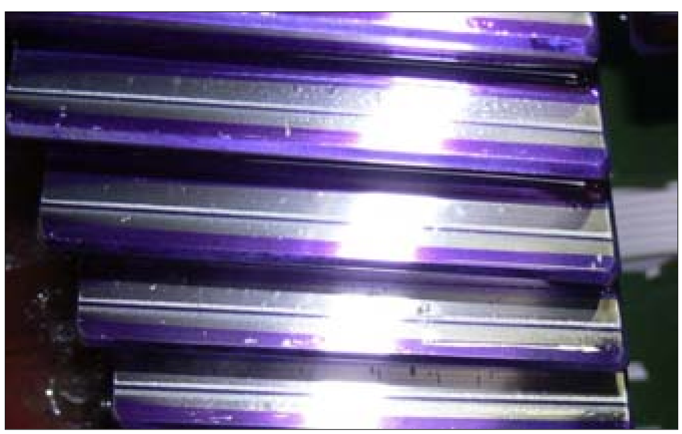


Figure P.18. Micropitting in the Convoloid HS gear; note the extensive area of non-contact at the tip of the teeth, this is due to excessive tip relief applied during manufacturing (91.2 hours at 200% load)

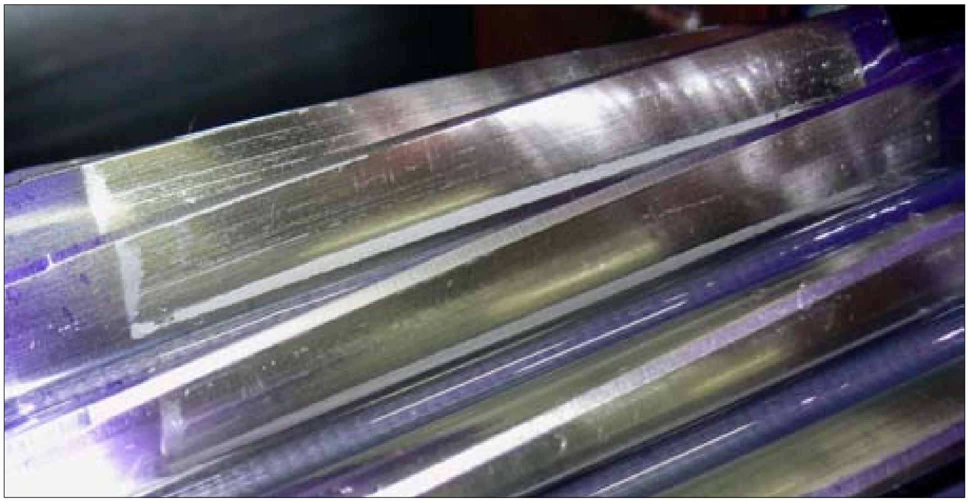


Figure P.19. Involute HS pinion at the conclusion of the 200% testing; note the extensive micropitting in the dedendum and at the edges; some micropitting is visible along the tip; this wear is progressing (235.8 hours at 200% load)

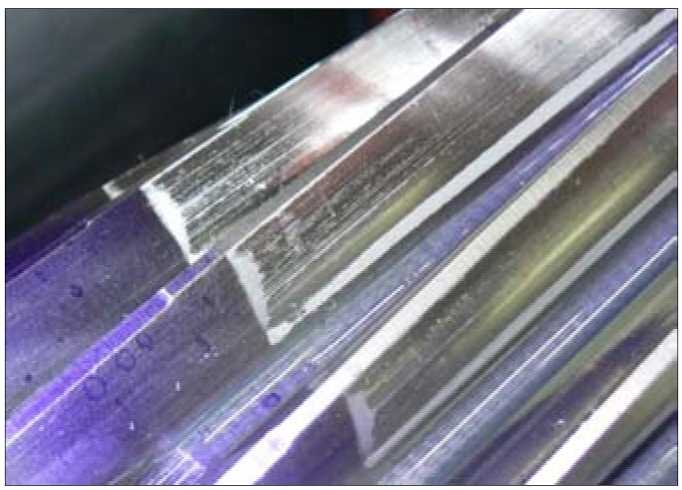


Figure P.20. Close-up of involute HS pinion with micropitting (235 hours at 200% load)

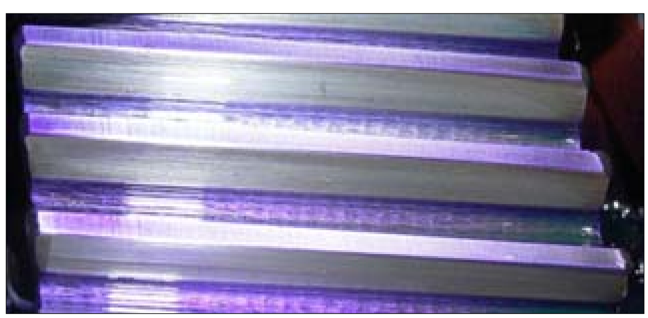


Figure P.21. Involute HS gear with no wear at the conclusion of the 200% test (235 hours at 200% load)

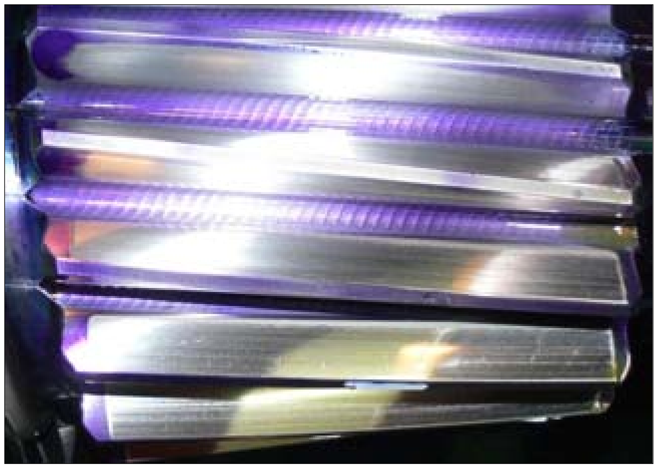


Figure P.22. Involute LS pinion at the conclusion of the 200% test showing micropitting wear around edges and at grinding peaks (235 hours at 200% load)

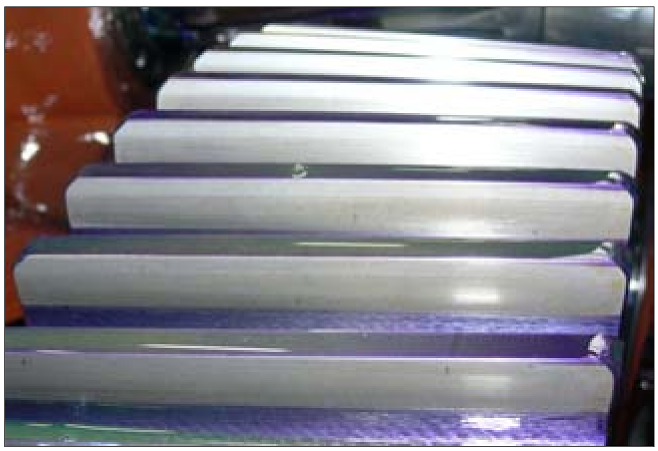


Figure P.23. Involute LS gear at the conclusion of the 200% test showing no wear (235 hours at 200% load)

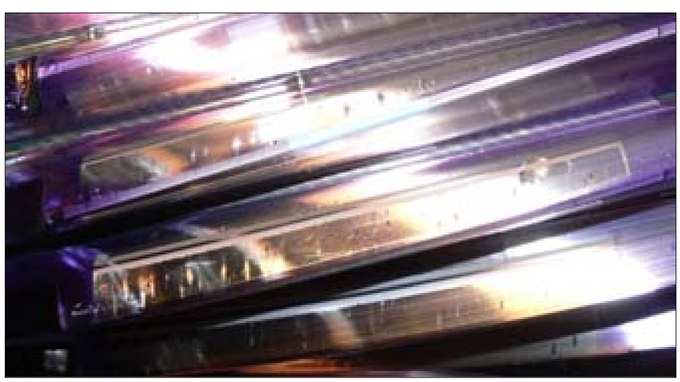


Figure P.24. Convoloid HS pinion at conclusion of the 200% test. Note that although there is extensive micropitting on both sides of the transition zone, it is not rapidly progressing (235.8 hours at 200% of the rated load)

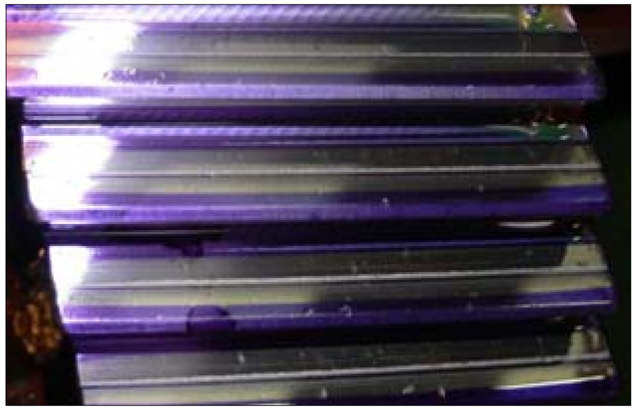


Figure P.25. Convoloid HS gear at the conclusion of the 200% test; micropitting is evident in the dedendum side of the transition zone, but the addendum side is free of wear (235 hours at 200% load)

Figure P.26. Convoloid LS pinion at the conclusion of the 200% test; micropitting is present at all edges except for the bottom of the dedendum (235 hours at 200% load)

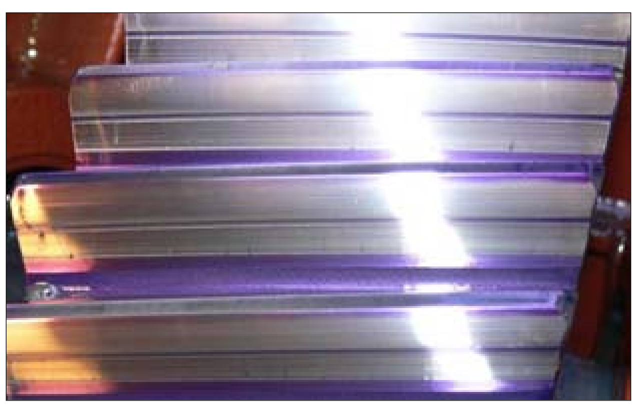


Figure P.27. Convoloid LS gear at the conclusion of the 200% test; light micropitting is evident around the transition zone and at the root; good tip relief causes the wear pattern to fade away at the tip; 235 hours at 200% load

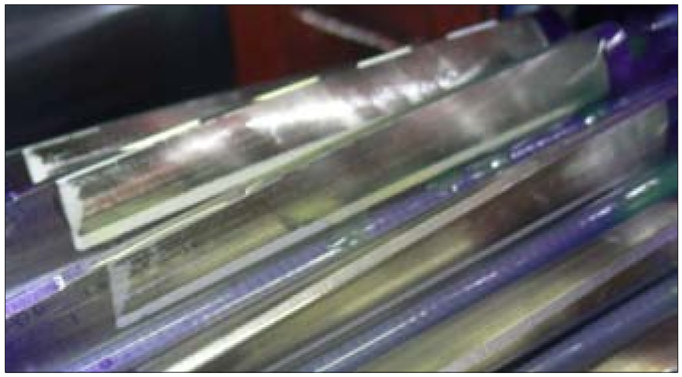


Figure P.28. Involute HS pinion (235 hours at 200% load; 67 hours at 240% load)

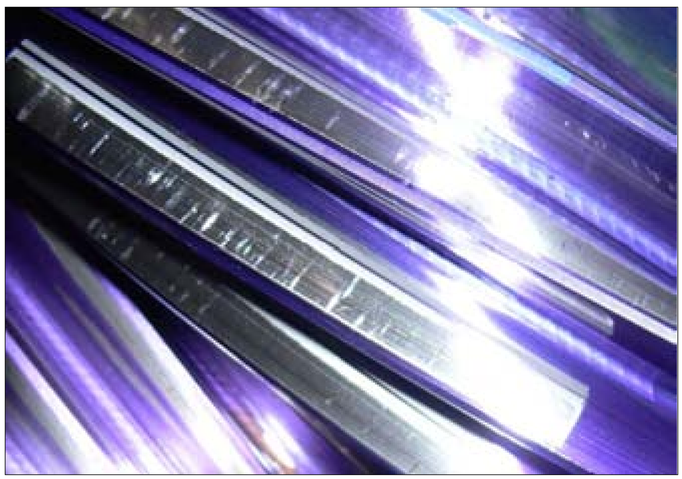


Figure P.29. Convoloid HS pinion (235 hours at 200% load; 67 hours at 240% load)

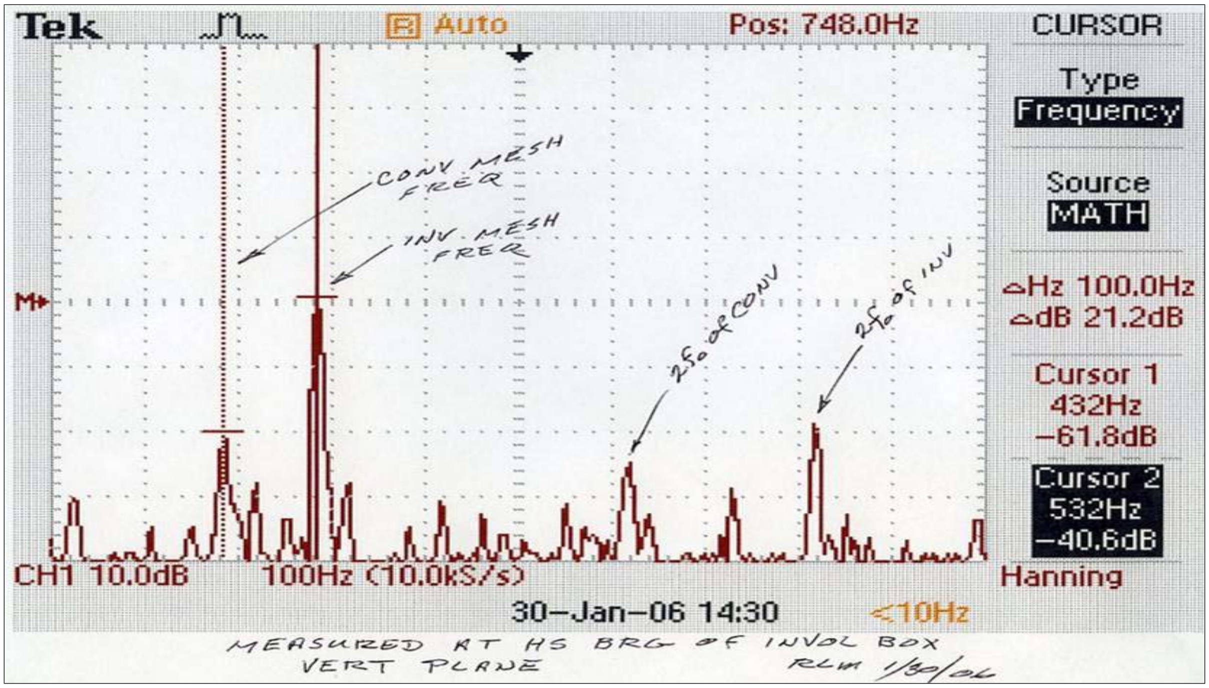


Figure P.30. Vibration spectrum for involute gearbox as measured at the high speed bearing; value for the involute high speed gear mesh (533 Hz) is shown in the black box (Cursor 2); transmitted vibration from the Convoloid gearbox (433 Hz) is listed as Cursor 1; secondary peaks are harmonics

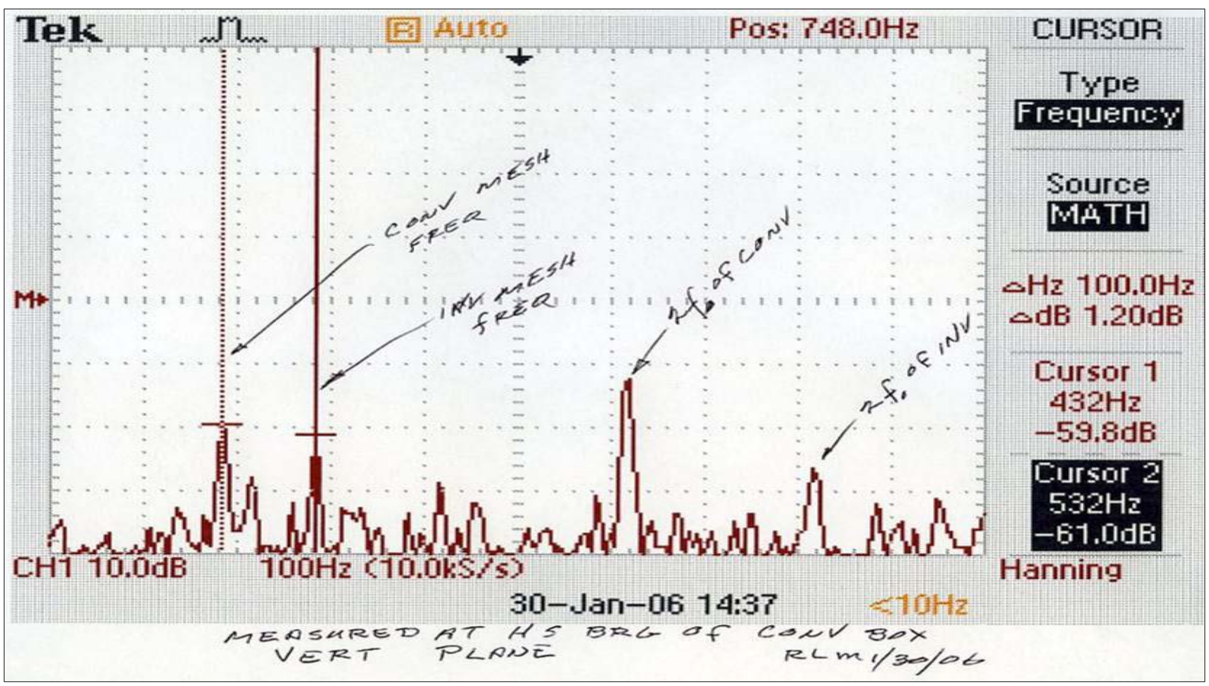


Figure P.31. Vibration spectrum for Convoloid gearbox as measured at the high speed bearing; value for the Convoloid high speed gear mesh (433 Hz) is shown in as Cursor 1; transmitted vibration from the involute gearbox (533 Hz) is listed as Cursor 2

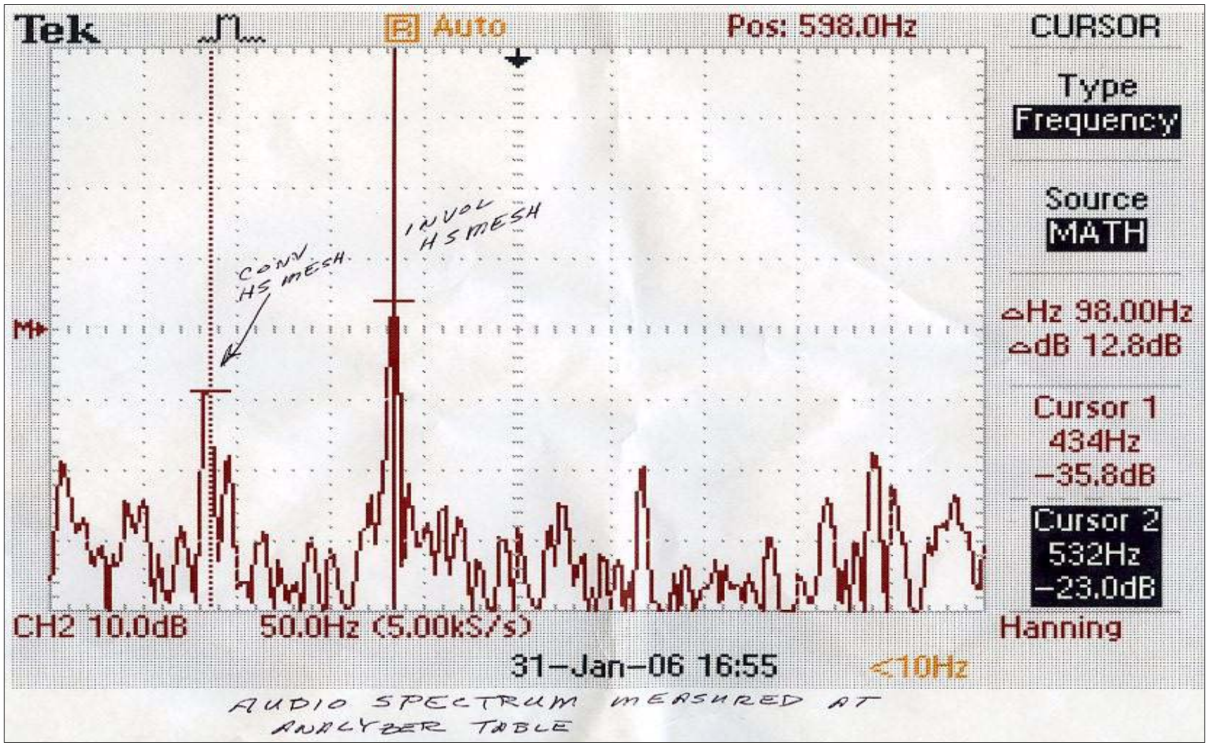


Figure P.32. Audio (sound) spectrum at a distance of 10 ft after approximately 300 hours of operation

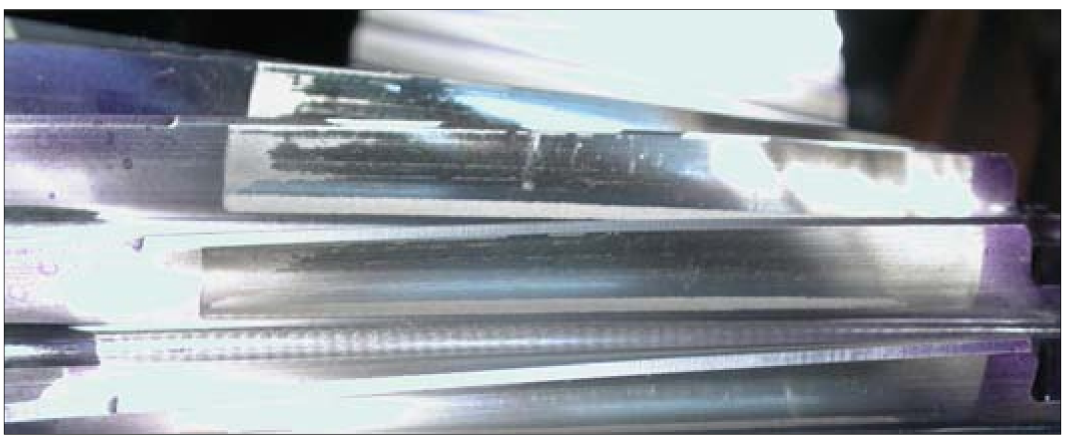


Figure P.33. Involute HS pinion showing progression of micropitting (235 hours at 200% load; 355 hours at 240% load)

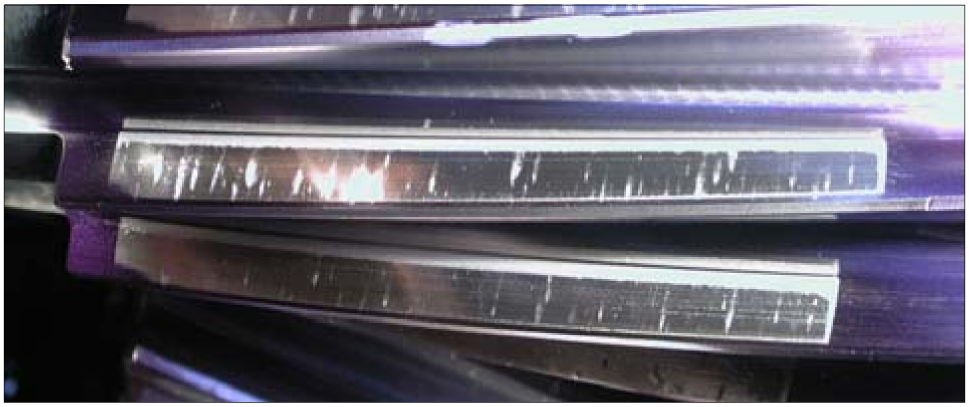


Figure P.34. Convoloid HS pinion showing minimal progression of micropitting (235 hours at 200% load; 355 hours at 240% load) vertical streaks are from aluminum particles (from the heat exchanger)

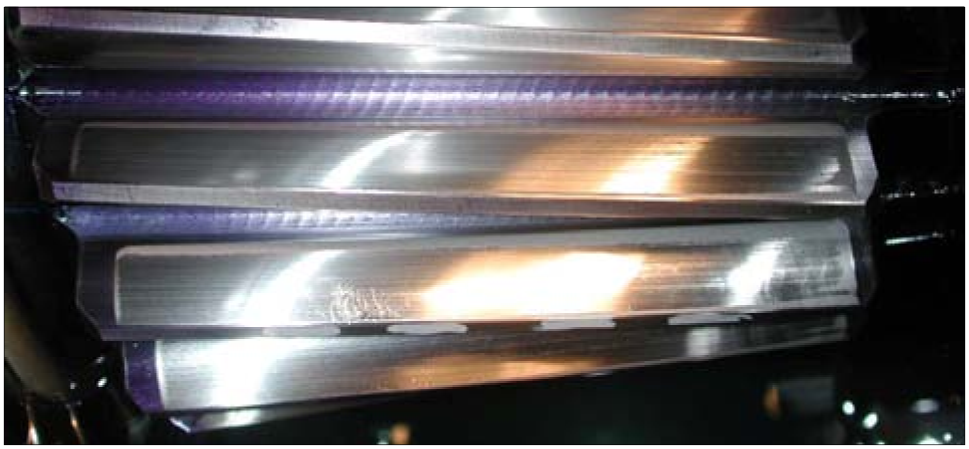


Figure P.35. Involute LS pinion showing substantial micropitting at the right edge and in the dedendum (235 hours at 200% load; 355 hours at 240% load)

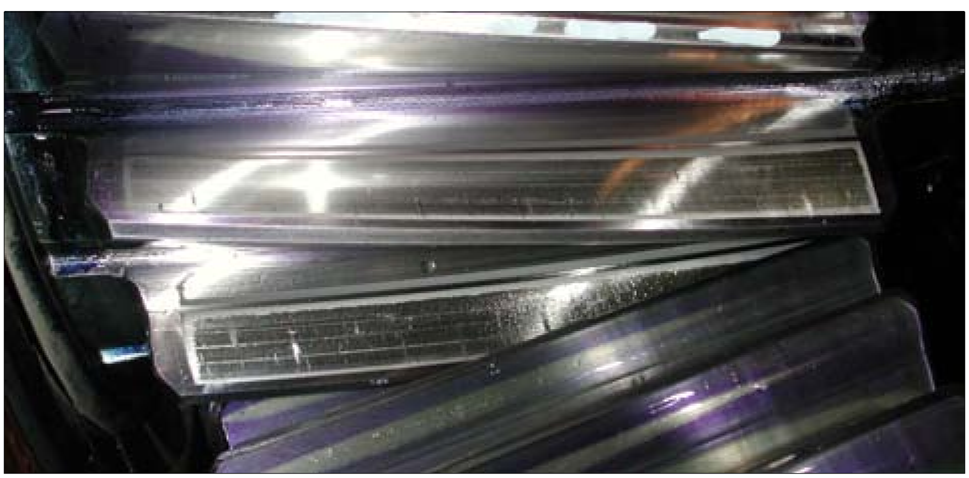


Figure P.36. Convoloid LS pinion also shows minimal progression of micropitting (235 hours at 200% load; 355 hours at 240% load)

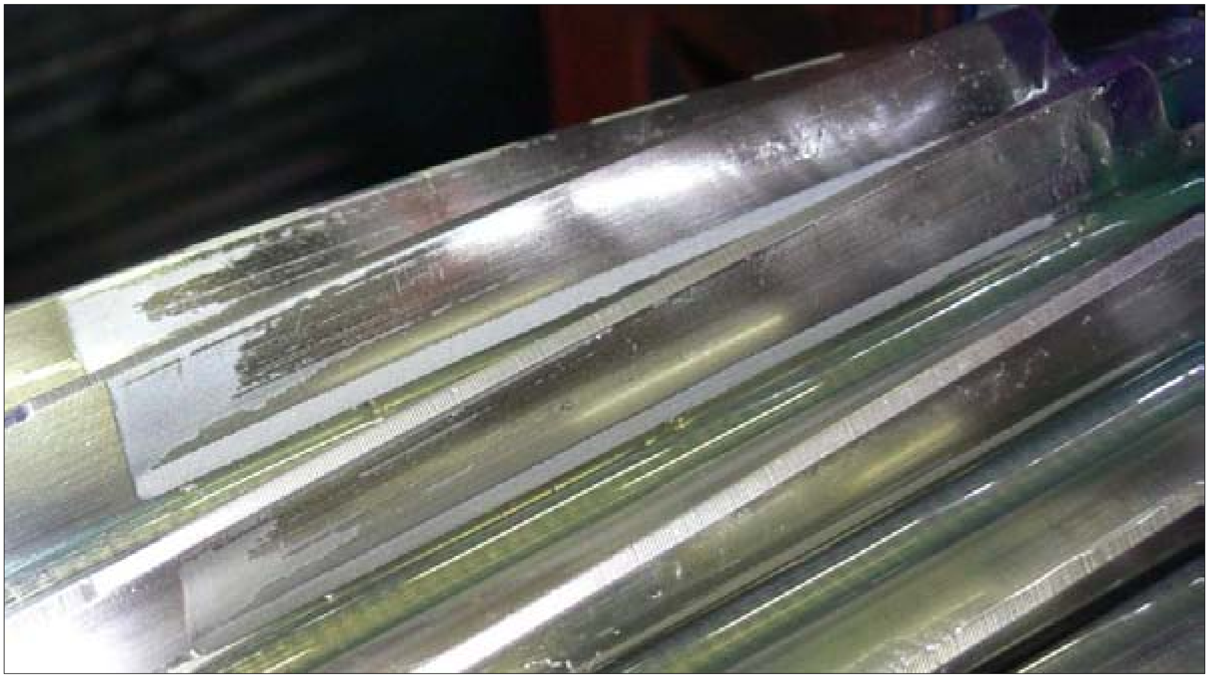


Figure P.37. Involute HS pinion at the end of test; wear due to micropitting is extensive along one side (235 hours at 200% load; 528 hours at 240% load)

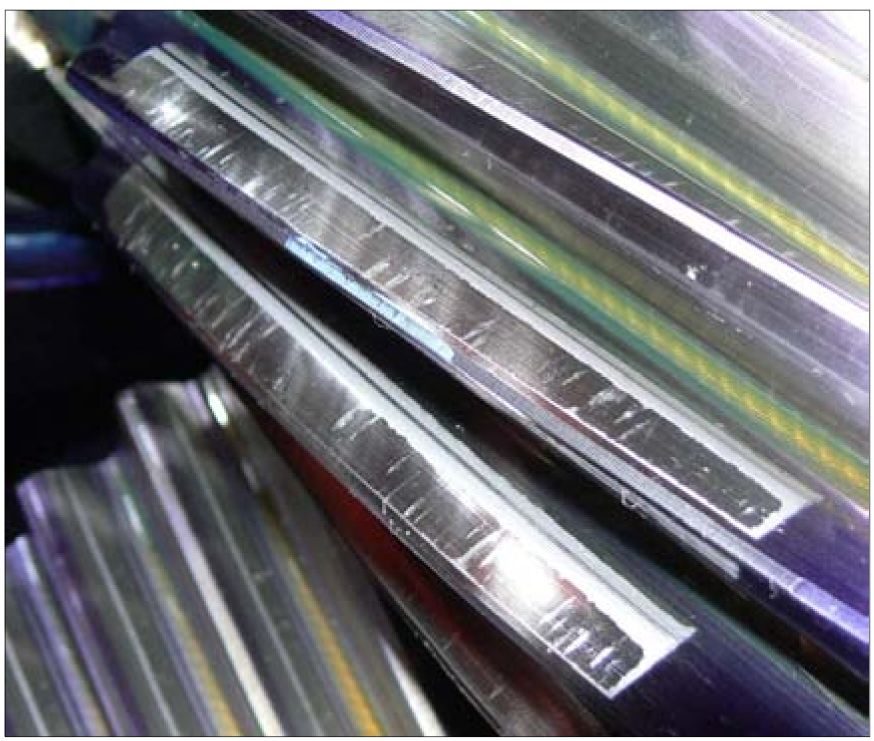


Figure P.38. Convoloid HS pinion at the end of test; wear due to micropitting is expanded slightly during the test, but no progression occurred on the edges (235 hours at 200% load; 528 hours at 240% load)

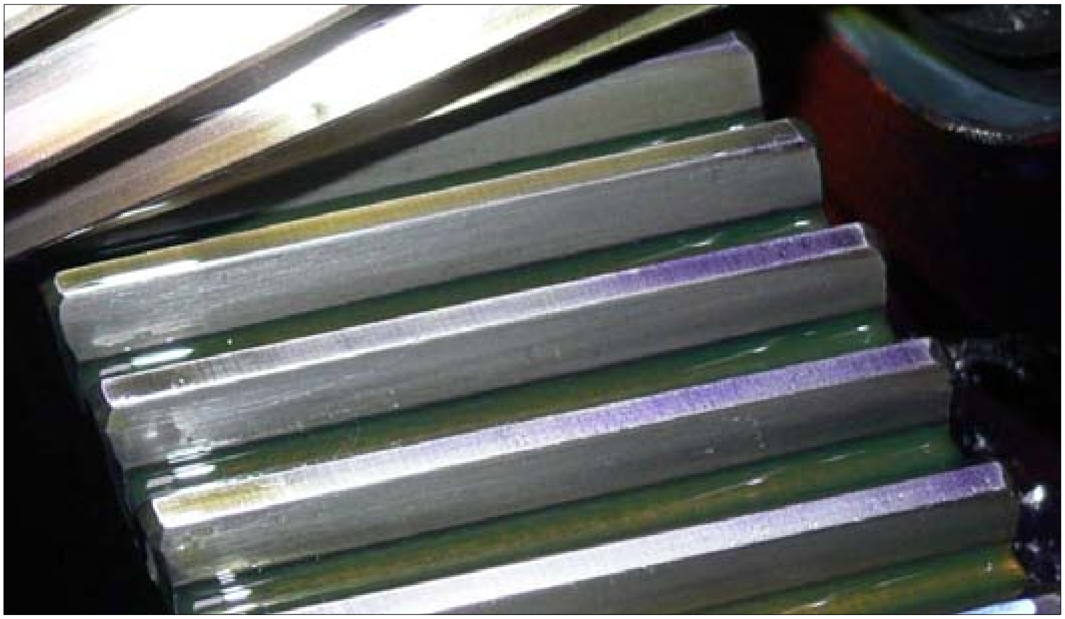


Figure P.39. Involute HS gear at the end of test; no wear is apparent (235 hours at 200% load; 528 hours at 240% load)

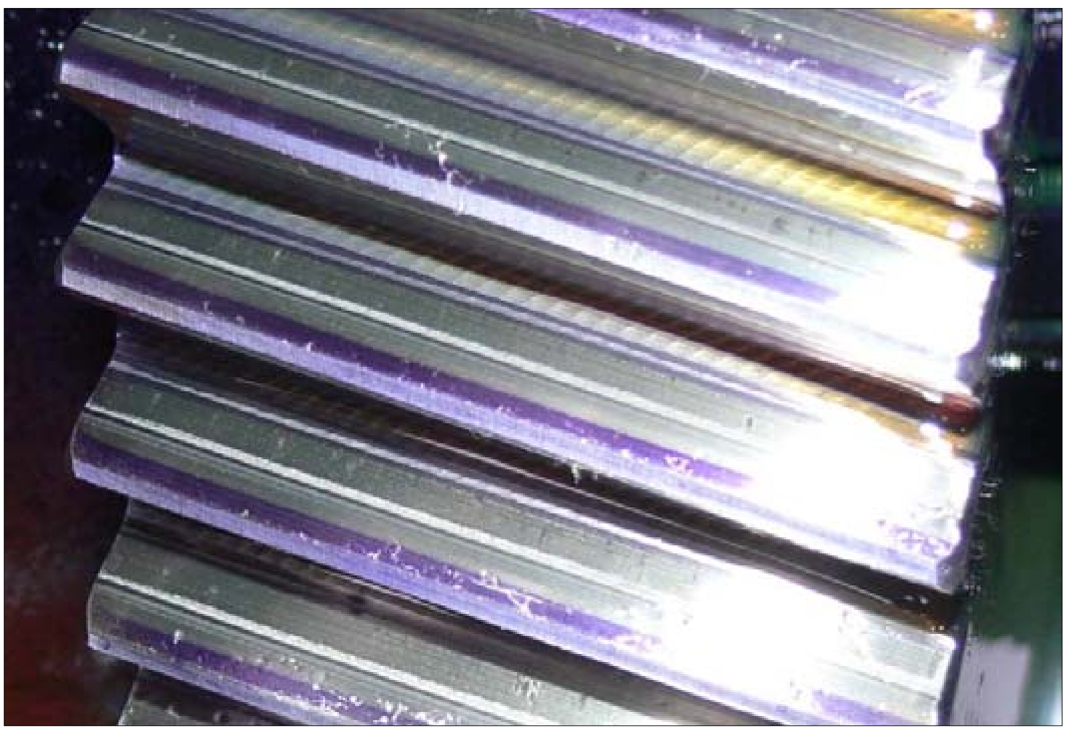


Figure P.40. Convoloid HS gear at the end of test; some wear due to micropitting is visible on the dedendum side of the transition zone; a large section of the addendum still shows marking dye indicating too much tip relief in the profile (235 hours at 200% load; 528 hours at 240% load)

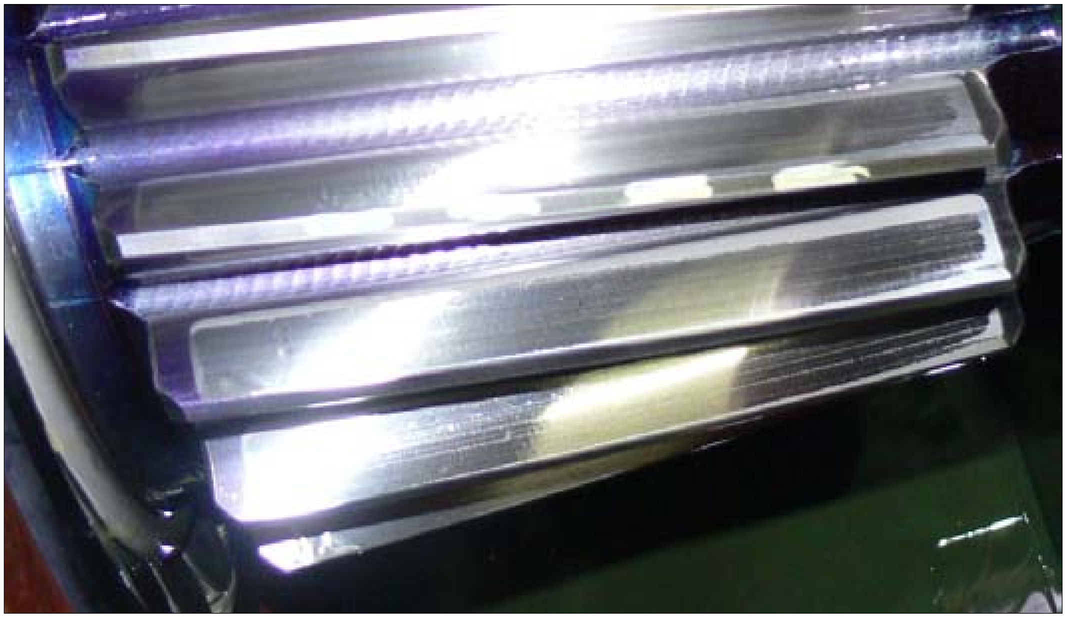


Figure P.41. Involute LS pinion at the end of test; similar to the HS pinion, micropitting wear is concentrated along one side and in the dedendum (235 hours at 200% load; 528 hours at 240% load)

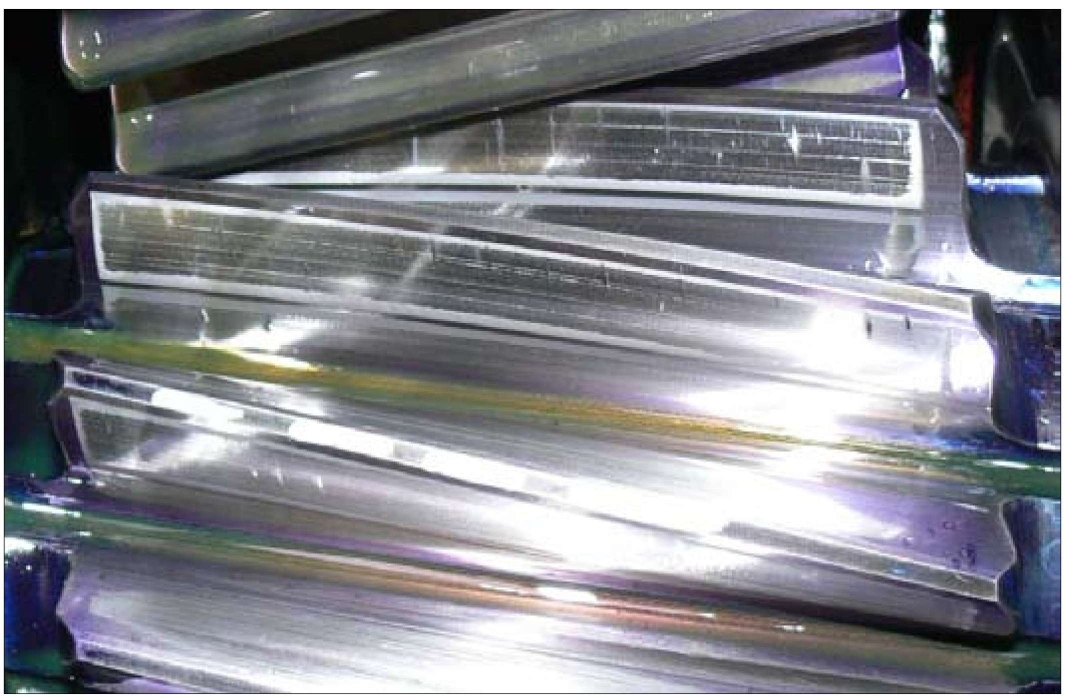


Figure P.42. Convoloid LS pinion at the end of test; wear due to micropitting is brackets the operating surface except for in the dedendum due to proper tip relief on the mating gear (235 hours at 200% load; 528 hours at 240% load)

Figure P.43. Involute LS gear at the end of test; no wear is visible (235 hours at 200% load; 528 hours at 240% load)

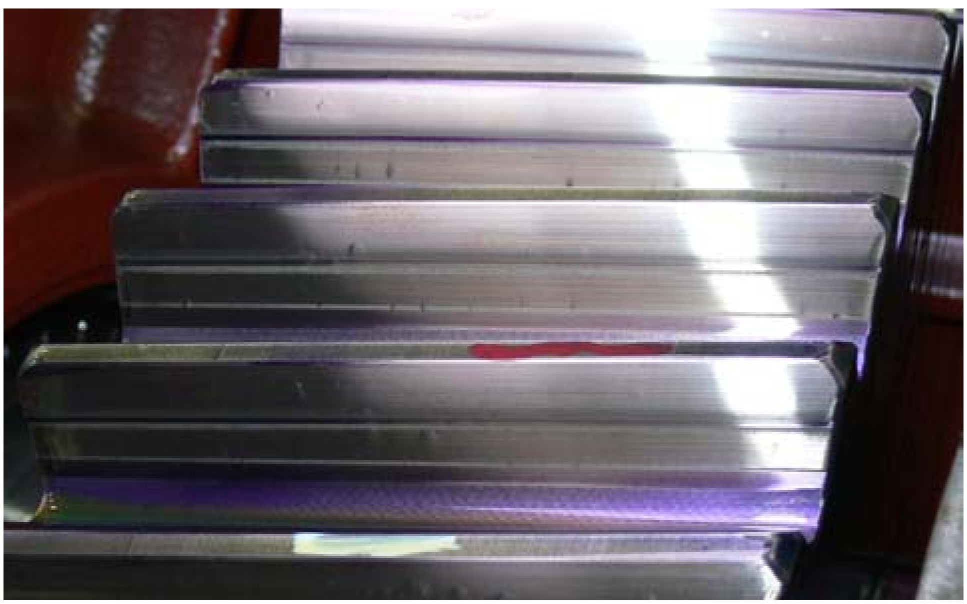


Figure P.44. Convoloid LS gear at the end of test; slight wear due to micropitting is visible in the dedendum (235 hours at 200% load; 528 hours at 240% load)

Appendix Q. Micon 108 Test Recommendations

Gear Engineers, Inc. P.O. Box 70327 Seattle, WA 98127 (206) 783-3919; gearengr@mcvittie.com

Memo Page 1 of 2 2004 September 22
Rev. A 2004 September 23
Genesis, LLP
Gear geometry proposal

To: Barney Berlinger

I've reviewed the recent correspondence and calculations for gear geometry of the test involute gears and offer the following recommendations.

Objectives of test:

- Compare performance of involute and Convoloid gear teeth in a typical small wind turbine application;
- Surface fatigue damage;
- Efficiency; and
- Avoid failure by tooth bending.

Rating calculations were made using the AGMA gear rating software, ver. 2.2, which provides comparative ratings according to AGMA 2001 and ISO 6336.

AGMA 6006, which governs wind turbine gear boxes, requires minimum ISO safety factors of 1.25 in pitting and 1.56 in bending at equivalent load (nominal rating times application factor) for this gearing.

The object of the 200% torque test is to run the gears at "endurance" load where the torque is equal to nominal load times application factor and safety factor (safety factor squared for pitting) to demonstrate that this required rated capacity has been achieved.

Any calculated ISO safety factor at 200% torque over 1.0 indicates that the gears are expected to survive the 200% torque test. Any calculated AGMA safety factor at 100% torque times application factor over 1.0 indicates that the gears are expected to survive the 200% torque 200 hour test, because AGMA ratings have a built-in 200% overload factor to allow a limited number of electric motor starts at 200% of nominal torque. The load distribution factor $K_{H\beta}$ was taken as 1.30 per LVR calculations. The actual dimensions of available hobs from TGW were used. It is essential that no hob substitutions be made during manufacture.

Grind stock allowances were adjusted to minimize the undercut at the root fillet due to hob protuberance. It is essential that this be closely monitored during manufacture because the hobs chosen are general-purpose tools and not the ideal tools for pinions with small numbers of teeth.

High speed gears:

The gear geometry has been optimized for pitting resistance as in a typical small wind turbine. This results in the following safety factors (allowable stress / calculated stress) at the test load of 492 HP (366 kW) at 1,830 rpm and 200-hr test duration.

- ISO pitting: 1.15
- AGMA pitting: 1.22
- ISO bending: 1.95 (limited by gear wheel)
- AGMA bending:1.34 (limited by gear wheel)

If the mesh geometry were optimized for bending, the AGMA rated lives in pitting would be slightly reduced and rated bending lives would be slightly increased. In my opinion this would jeopardize the acceptance of the test results, because it might be alleged that the design was biased to favor a pitting failure.

Typical wind turbine gear mesh geometry is optimized for pitting and micropitting resistance, because that is by far the most common failure mode in actual turbines and in turbine gearbox overload acceptance tests.

Low speed gears:

The gear geometry has been optimized for bending resistance to reduce the risk of an unwanted bending failure. The optimization was done by adjusting the profile shift factors of pinion and gear without changing the helix angle, pitch, or number of teeth. This results in the following safety factors (allowable stress / calculated stress) at the test load of 492 HP (366 kW) at 305 rpm and 200-hr test duration:

- ISO pitting: 1.15
- AGMA pitting: 1.12
- ISO bending: 1.80
- AGMA bending:1.12

If the mesh geometry were optimized for pitting, the AGMA rated lives in pitting would be slightly increased and rated bending lives would be slightly decreased. I believe that that the optimization for bending can be justified, because there are good reasons to avoid a bending failure.

The calculation details are attached as separate files.

Don McVittie

Appendix R. Micon 108 200% x 200-Hour Test Vibration Analysis

Gearbox noise is directly related to the various vibrations generated within. From the different gear meshes to individual bearing components, all vibrations are related to the shaft speed. Using a detailed analysis and a spectrum analyzer with filtering, it is possible to monitor a system for early wear. Each peak within the frequency band points to a different excitation, and thus a different component, within the gearbox. Proper analysis requires monitoring to establish baseline values. Subsequent readings can be compared to the baseline to determine which frequencies are changing.

The primary focus of the vibration data readings that were collected was to monitor the changes in vibration as the test progressed. Detailed analysis using a spectrum analyzer was conducted late in the test. The specific vibration values differ greatly from the Convoloid to the involute. However, this is not deemed significant in that there are too many variables to determine how potential natural frequencies might affect the vibrations. The locations of the sensors also have an affect on the magnitudes.

What is interesting is that although the involute vibration remained steady, vibrations within the Convoloid gearbox reduced as the test progressed. Figure R.1 shows the vibration values for the first full day of testing at 200% load. Compare this with Figure R.2 which was collected the last full day of testing. The average Convoloid value fell from 1.685 to 1.520, a 9.8% decrease. The variations in the total vibration values are due to cycling of the lubricant cooling fans. Vibration readings dramatically increased when the fans ran (especially for the Convoloid).

Spectrum analysis of the two gearboxes shows distinct spikes at the high speed gear mesh frequencies (385 Hz for the Convoloid and 471 Hz for the involute) (*see* Figure R.3, Figure R.4). Additionally, the harmonics of these fundamental frequencies are very evident. In fact, the harmonics tend to show higher amplitudes than the fundamental. A summary of Mr. Meredith's notes which detail frequencies and amplitude are summarized below. Without having a baseline for comparison, no further analysis is possible.

- High speed shaft frequency (1768 rpm / 60) = 29.47 Hz
- Hydraulic dither frequency = 263.75 Hz

Convoloid: High speed gear mesh frequency (29.47 * 13 teeth) = 383 Hz.

Table R.1. Convoloid Frequency Spikes and Their Amplitudes

Vertical Deflection		Horizontal Deflection	
Frequency (Hz)	Amplitude (G.O-P)	Frequency (Hz)	Amplitude (G.O-P)
383	67×10^{-3}	117.5	410×10^{-3}
471	65×10^{-3}	765.75	500×10^{-3}
766	200×10^{-3}	1150	243×10^{-3}
1150	126×10^{-3}	1530	800×10^{-3}
1530	315×10^{-3}	1915	315×10^{-3}

No frequency spikes relate to the dither frequency, therefore vibrations from the loading mechanism are not significant.

Involute

High speed gear mesh frequency (29.47 * 16 teeth) = 471 Hz

Table R.2. Involute Frequency Spikes and Their Amplitudes (a frequency spike also occurred at 88.5 which is a harmonic of the high speed shaft speed)

Vertical Deflection		Horizontal Deflection	
Frequency (Hz)	Amplitude (G.O-P)	Frequency (Hz)	Amplitude (G.O-P)
470	364×10^{-3}	470	176×10^{-3}
940	389×10^{-3}	940	227×10^{-3}
1410	397×10^{-3}	1410	330×10^{-3}
		1880	225×10^{-3}

Motor

High speed shaft frequency = 29.47 Hz

Table R.3. Motor Frequency Spikes and Their Amplitudes (a frequency spike parallel to the shaft occurred at 1410-1414 Hz with an amplitude of 302×10^{-3} G.O-P)

Vertical Deflection		Horizontal Deflection	
Frequency (Hz)	Amplitude (G.O-P)	Frequency (Hz)	Amplitude (G.O-P)
29.5	104×10^{-3}	470	280×10^{-3}
78.5	124×10^{-3}		
88.5	46.1×10^{-3}		
118	56.5×10^{-3}		
157	52×10^{-3}		
470	101×10^{-3}		

Conclusions

At the primary frequencies of the high speed gear meshes (383 Hz for Convoloid, 471 Hz for the involute) the involute definitely shows higher amplitudes. The involute frequency is great enough to show in the Convoloid measurement. Subsequent spikes at various harmonics could be due to natural frequencies of the test structure and gearboxes. Due to cessation of testing, no additional analysis (natural frequency determinations) is available.

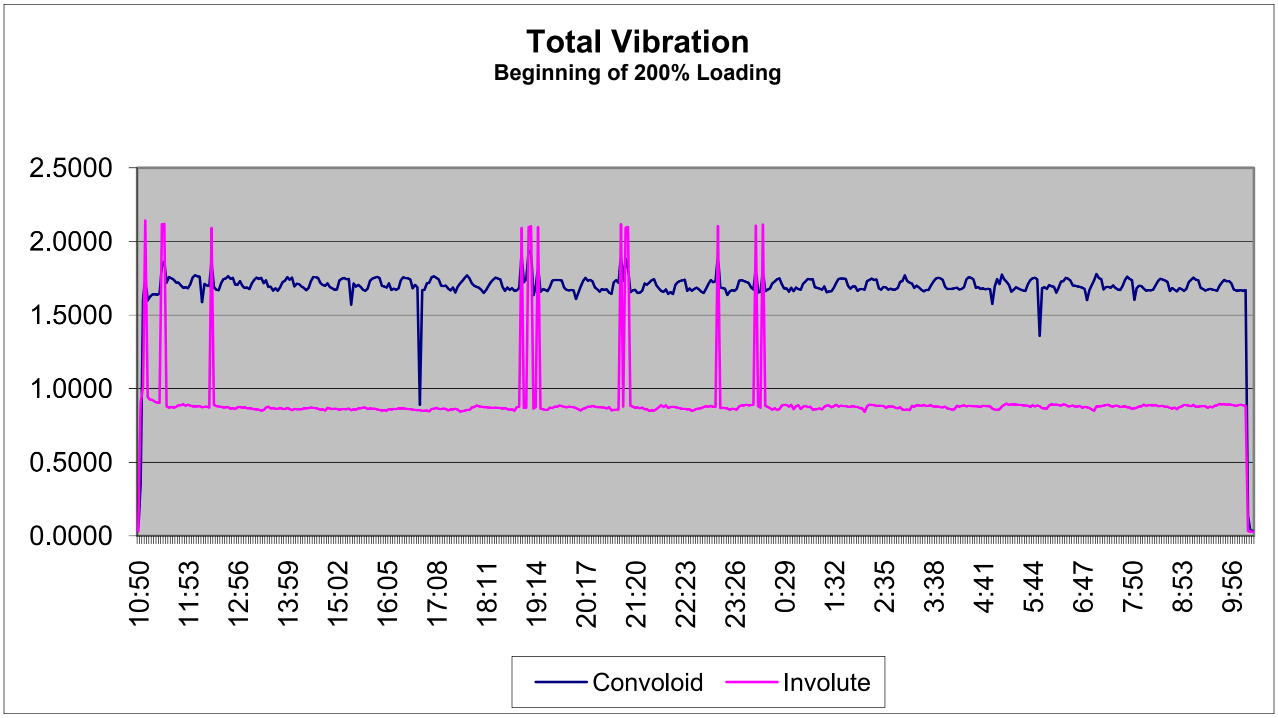


Figure R.1. Gearbox total vibration at the beginning of the test—200% load

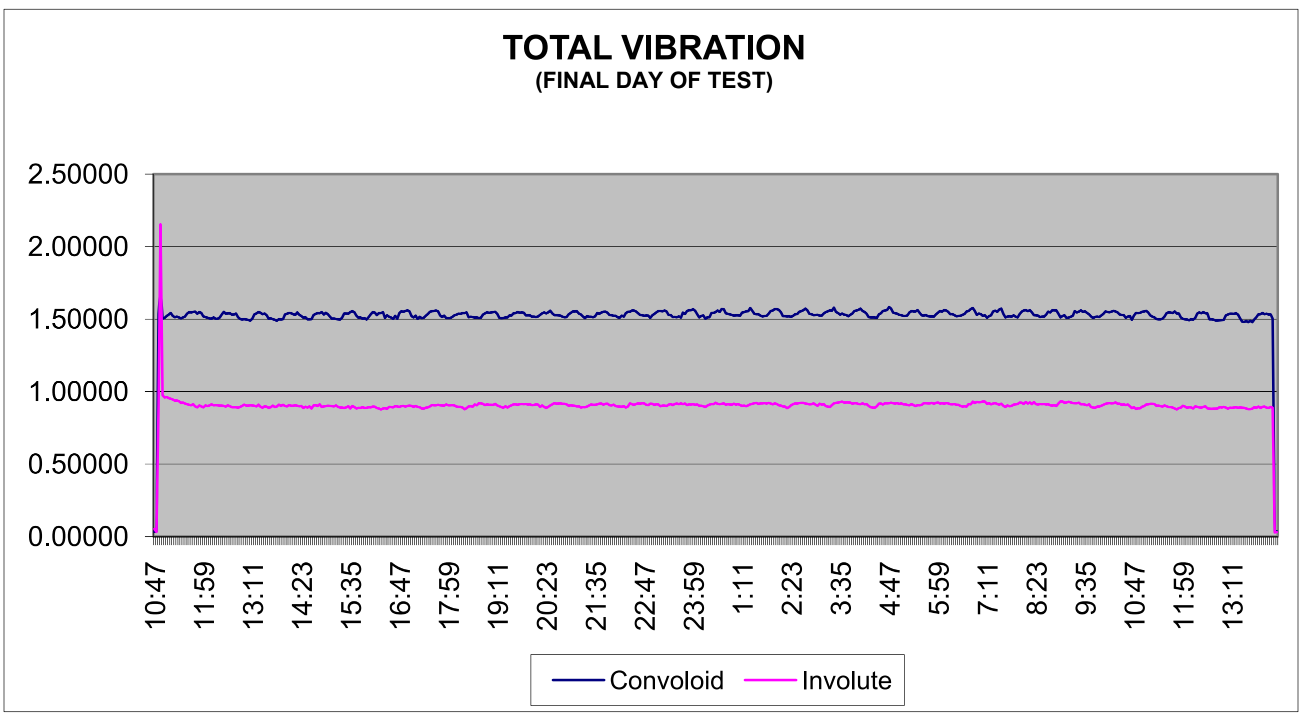


Figure R.2. Gearbox total vibration at the end of the test—200% load

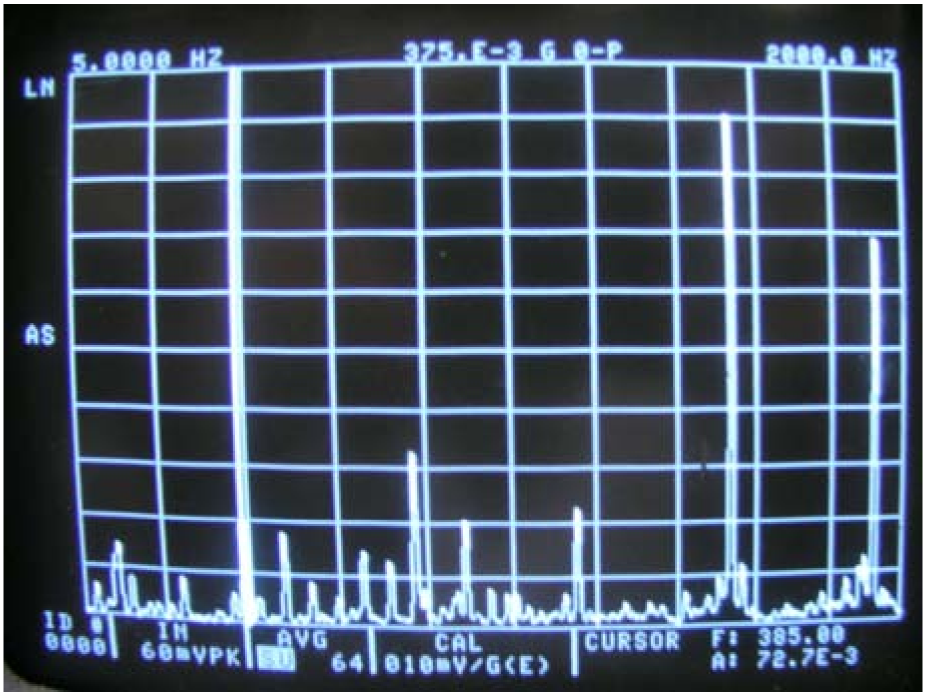


Figure R.3. Vibration spectrum of the Convoloid gearbox; cursor is located at the HS pinion fundamental frequency (385 Hz)

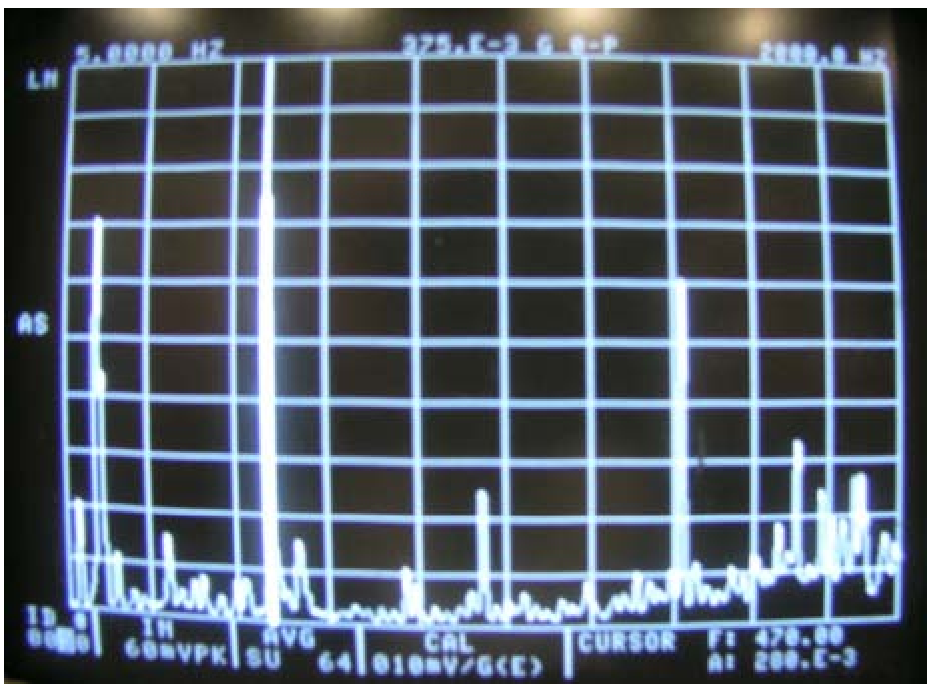


Figure R.4. Vibration spectrum of the involute gearbox; cursor is located at the HS pinion fundamental frequency (470 Hz)

Appendix S. Test Observations—Richard L. Meredith, D.C. Energy

Observations From Genesis
200% Rated Load 235-Hour Test Data

December 23, 2005

Richard L. Meredith, D C Energy, Bothell, WA

1. The hottest bearing in the Convoloid gearbox was the outside low speed which in the worst case was 1°F above its sump with a maximum of 167°F and 166°F respectively.
2. The hottest bearing in the involute gearbox was the inside intermediate bearing which never exceeded 163°F (which was also only 1°F above that sump temperature).
3. Both reservoirs were controlled to a maximum of 161°F and a minimum of 145°F.
4. Over the 235-hour course of the test, the vibration level of the involute gearbox tended to increase slightly. Less than 6% increase was noted.
5. Over the same period, the Convoloid gearbox vibration level tended to decrease slightly. Around a 9% change downward was noted.
6. As calculated above, the total losses for both gearboxes were between 5.26% and 5.52%. This includes the bearing losses and the oil churning losses.
7. Throughout the test, the ISO cleanliness level for 6u particles in the Convoloid gearbox was 7, according to the online particle counter.
8. Throughout the test, the ISO cleanliness level for 6u particles in the Involute gearbox was 11, according to the online particle counter.
9. The foregoing two points were confirmed by daily patch tests for each gearbox. Patches were made into slides and have been preserved.
10. There was never a “hot” bearing or any unusual noises from either gearbox during the 235-hour run.
11. The oil samples that were collected every day from each gearbox indicated that there was more entrained air in the oil from the involute box. No concrete reason for this could be determined.
12. Due to the way that the prime mover motor was mounted in relation to the involute gearbox, there was approximately 700 FPM air blowing on the outside of the involute gearbox; partially accounting for the fact that the heat exchanger fan on the involute box less frequently than that of the Convoloid box. Additionally, due to the design of the heat exchanger used on the Convoloid box, there was more cooling due to normal flow through the core even when the fan was not running. The temperature drop across this heat exchanger, with normal flow, was 3° to 5° F as compared to 1°F for the Convoloid box.
13. Daily physical inspections did not reveal any impending catastrophic failures in either gearbox.

At the completion of the 235.8-hour test run, both gearboxes were running normally in all observable respects with nothing noted that would preclude further test running.

Richard L. Meredith
D C Energy
Test Instrumentation Supervisor.

Appendix T. Derivation of Gearbox Efficiencies

Overall System Efficiency

In a four-square test setup, the only power required by the system is that required to rotate the shafts. The amount of power supplied by the motor is equal to the losses within the system. Therefore, by determining the motor power supplied to the system, the losses can be found. Two methods were used to ascertain the power loss: amount of variation in shaft speed of the motor—no-load versus load; and electrical power consumption. This discussion is taken from a fax sent by Mr. Meredith to Mr. Williston on December 21, 2005.

Input Power Determination

Motor power based upon motor speed:

- Test conditions: Motor speed at 200% gearbox load, measured with a calibrated strobe tachometer = 1,768 rpm
- No-load speed = 1,775 rpm (test system attached, yet no loading to the system)
- Open shaft motor speed = 1,778 rpm (no equipment attached to the motor shaft)

Conclusion

The output frequency of the variable frequency drive (VFD) for the motor is less than 60 Hz. Because 60 Hz would result in ~1,800 rpm shaft speed, the actual frequency provided by the VFD is as follows.

$$\frac{(1778 \cdot 60)}{1800} = 59.27 \quad \text{HZ}$$

The motor nameplate shows 1,775 rpm for the 60 HP delivered power. Therefore, there is a full load (FL) slip in motor speed of 25 rpm. The full load power factor is given as 0.90. The full load motor line current is given as 67A.

Measured slip at the operating frequency if 1,778 - 1,768 = 10 rpm.

$$\frac{10\text{rpm}}{25\text{rpm}} \cdot 60\text{HP} = 24 \quad \text{HP test load}$$

Or 24/60 = 40% of motor horsepower.

Motor power by electric input:

$$W = E \cdot I \cdot PF \cdot \sqrt{3}$$

Where: W = electrical power (watts);
E = voltage (rms) = 477 volts rms;
I = current (amps) = 35.5 A; and
PF = Power Factor (a typical power factor at 40% load) = 0.64.

$$W = (477)(35.5)(.64)(1.732) = 18.77kW$$

Converting to horsepower:

$$18.77kW * \left(\frac{1hp}{.746kW} \right) = 25.16hp$$

There is a fairly good agreement between these two methods of arriving at the input power (losses).

Circulating Power

$$HP = \left(\frac{torque * speed}{63025} \right) = \left(\frac{(439000lb - in) * 65.48rpm}{63025} \right) = 456.1hp$$

Efficiency

Efficiency % = (1—loss) * 100%

Overall losses for the system:

- Motor Speed method: (24 hp / 456.1 hp) = 0.0526
- Electrical Power method: (25.16 hp / 456.1 hp) = 0.0552

The resultant system efficiency is 94.74% and 94.48% respectively, yielding individual gearbox efficiencies of 97.4% and 97.2%.

Gearing Efficiency

The following discussion references AGMA ISO 14179-1: Gear Reducers—Thermal Capacity Based on ISO/TR 14179-1.

Every component within a gearbox generates a certain amount of heat during operation. This heat is converted from mechanical power due to inefficiencies in the component. An approximation of these losses is given in AGMA ISO 14179-1. By removing the losses due to components other than the gearing, an approximate gearing efficiency can be determined. The additional step of removing the gear churning losses and the load-based losses of the involute gear set could provide an approximate idea of the Convoloid load-based losses. Thus, the efficiency of the Convoloid gear set can be found.

Non-gearing losses consist of heat generated in the bearings and by contacting oil seals. Two types of loads must be calculated: Load-based in which the amount of heat generated is affected by the amount of load applied, and non-load based in which the generated heat is strictly due to friction and/or oil churning losses based on rotational speed. Load-based losses are calculated with 456.1 hp. Table T.1 shows the heat (in horsepower) generated in the involute gearbox.

Table T.1. Involute Non–Gear Inefficiency Losses

Component	Type of Load	Shaft Speed (rpm)	Power Loss (hp)

High Speed (2 seals)	Non-load	1768.0	0.072
Low Speed (2 seals)	Non-load	65.5	0 (non-contacting seals)
HS Bearings (2)	Non-load (churning loss)	1768.0	0.031
	Load-based		0.771
INT Bearings (2)	Non-load (churning loss)	294.7	0.029
	Load-based		0.691
LS Bearings (2)	Non-load (churning loss)	65.5	0.028
	Load-based		0.500
		TOTAL:	2.12 hp

Table T.2. Convoloid Non–Gear Inefficiency Losses

Component	Type of Load	Shaft Speed (rpm)	Power Loss (hp)
High Speed (1 seals)	Non-load	1768.0	0.036
Low Speed (2 seals)	Non-load	65.5	0.0 (non-contacting seals)
HS Bearings (2)	Non-load (churning loss)	1768.0	0.031
	Load-based		0.811
INT Bearings (2)	Non-load (churning loss)	294.7	0.029
	Load-based		0.889
LS Bearings (2)	Non-load (churning loss)	65.5	0.028
	Load-based		0.630
		TOTAL:	2.49 hp

Using the electrical power inefficiency, the total non-gearing component losses equate to 4.61 hp. Therefore the total losses of only the gearing in the test becomes: 25.16 hp - 4.61 hp = 20.55 hp; giving a gearing total inefficiency of: (20.55 hp / 456.1 hp) = 0.0450.

The resultant efficiency for the gearing is 95.5 % for the system, or 97.8% for each gearbox yielding a per–gear mesh efficiency estimate of 98.9% for both the involute and Convoloid pairs.

Appendix U. Convoloid Gear Separating Force Qualification

Comparison of Convoloid Gear Performance To
AGMA Rating Formulae and Factors—Micon 108

February 6, 2007

A key aspect of designing a gear drive system is to provide sufficient support for the gears. This foundation takes the form of bearings and the housing supporting the bearings. Calculation of forces induced by the torsional loads applied to the gears is necessary to properly size the bearings and to confirm the strength of the housing. When torque is applied to helical gearing, reactive forces occur in all three planes. For Convoloid gearing, two of these forces, tangential and axial (or thrust), directly follow existing load theory. However, due to the difference in profile of the Convoloid tooth form to the involute, the third load, the separating force, must be calculated in minute increments with the resultant being the sum of the forces. Although there is much confidence in these calculations, it is prudent to qualify these forces.

During the operational test, two Micon 108 gearboxes (one with involute gearing and one with Convoloid gearing) were operated in a “four-square” or back to back type test. The housings of the gearboxes were instrumented with, among other things, strain gages to help understand how the housing deflected when loaded. Three different test loads were applied for a significant period of time during the test giving three distinct groups of data.

Strain gages were permanently mounted on both sides of the housing and located between the high speed (HS) and intermediate (INT) bores; and between the INT and low speed (LS) bores. These gages, arranged in a Full Poisson Bridge (Wheatstone Bridge), allow for a high degree of sensitivity (2.6 times the sensitivity of a quarter bridge) and compensate for temperature variations. To have a point of reference in which to reduce the copious amount of data from the operational test, values obtained from the strain gages must be compared to strain values obtained when the housing is subjected to known loads.

To develop a load / strain cross reference, a static load test was conducted on the housings. In addition to the Convoloid housing, the involute housing was also tested to qualify the method. If the resulting strains can be correlated to the loads for involute gearing, then the method should reasonably predict the loads for the Convoloid gearbox. These loads then could be compared to the calculated loads. The existing strain gages were able to be used, preserving the consistency with the operational test.

Static Load Test Equipment

In an attempt to duplicate how the loading of internal gear loads impact the housing (through the bearings), machined plugs were fitted into the bearing bores (Figure U.1). Steel tubing is fitted through these plugs and hydraulic cylinders are used to apply separating forces between the bores. The cylinders are placed within the housing in the approximate positions of the gear mesh they are duplicating (Figure U.2).

The amount of force produced by the hydraulic cylinders is the product of hydraulic pressure and cylinder surface area. The HS cylinder has an area of 3.14 in^2, and the area of the LS cylinder is 7.22 in^2. By applying a given amount of pressure (to produce a known load) to both cylinders, the resulting strain can be gathered. This data should give a good baseline that can be used for analysis of values obtained during testing.

To preserve the viability of the test, none of the strain gages have been removed or altered from the original test configuration. Comparison of the strain values gathered during the operational test and these static values allows for a direct determination of the actual loads that were imposed on the housing during the test. It is important to compare the actual loads with theoretical calculations in order to validate or refine those calculations. Also, comparison of the static and dynamic test values obtained from the involute gearbox should give insight into potential measurement error and unknown variations the test could have.

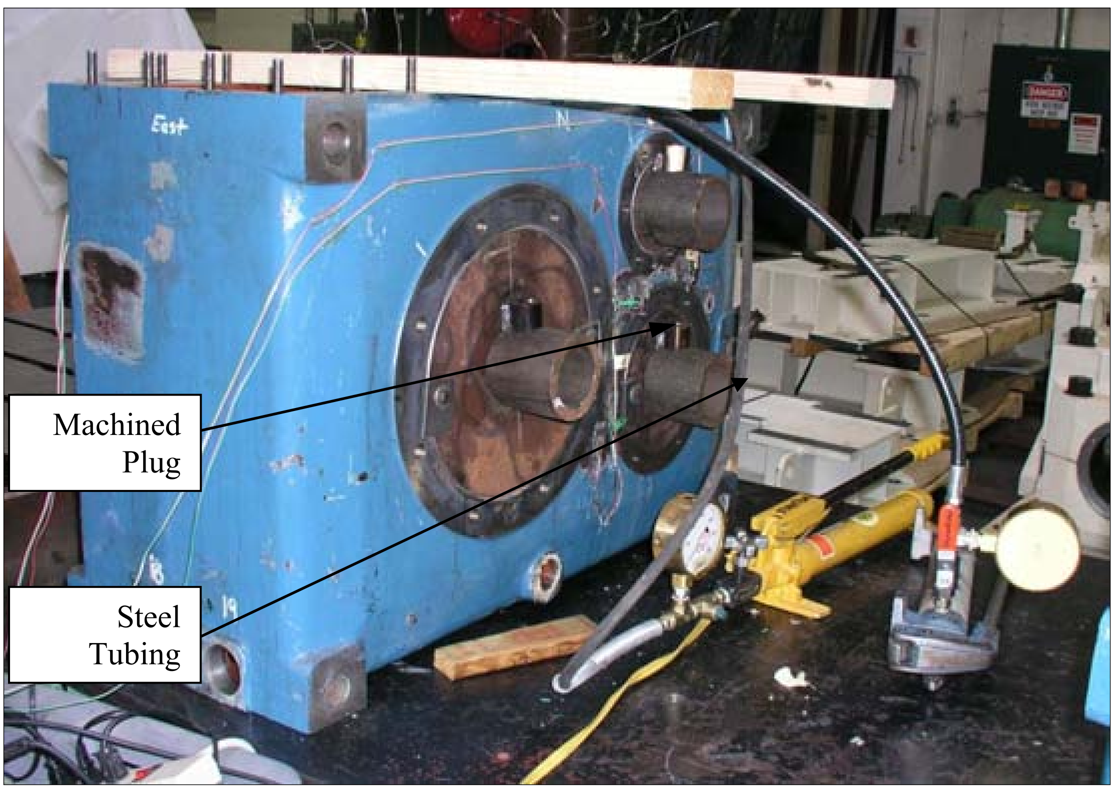


Figure U.1. Static load test equipment set-up

As can be seen from the photos, access through the bearing bores to the cylinders was not possible. Therefore, plumbing for the hydraulic cylinders required that the coverplate not be installed. Early in the project, the housings were found to be unacceptably flexible when subjected to thrust loads (induced when setting the tapered roller bearings). The faces of the housing literally "bowed" out, preventing the bearings from having the proper setting. To counter this deflection, the coverplate was pinned in four places with removable taper pins.

Subsequent measurements showed that the deflection at the high speed bearing bore had been reduced from 0.009 in (at a load equal to that produced by the 200% test load) to 0.0015in. As this test was focused on qualifying the separating loads (the forces that try to push the two gears in mesh away from each other) and not thrust, it was felt that any support of the cover would be minimal and could be neglected.

Test Results

The amount of loading applied is related to the forces produced by the torques of the operational test. At the end of the test, a torque equal to 240% of the rated load was applied. This equates to an approximate separating force of 5,200 lb between the HS and INT bores, and a force of 16,800 lb between the INT and LS bores. The range of forces to be applied by the hydraulic cylinders began with no loading (to establish the strain gage offset) and increased (in 500-psi increments) to 3,000 psi (~9,400 lb) on the HS cylinder and 2,500 psi (~18,050 lb) on the LS cylinder. The amount of pressure on the LS cylinder was limited to prevent an excessive amount of load on the housing.

Hydraulic pressure was initially applied separately to the cylinders to qualify the operation of the strain gages. As expected, strain appeared to follow increases in the applied pressure (and thus load) in a linear manner. Values from the Convoloid housing were higher than with the involute housing (Figure U.3, Figure U.4).

Figure U.2. Close-up of a hydraulic cylinder in position to apply a separating load

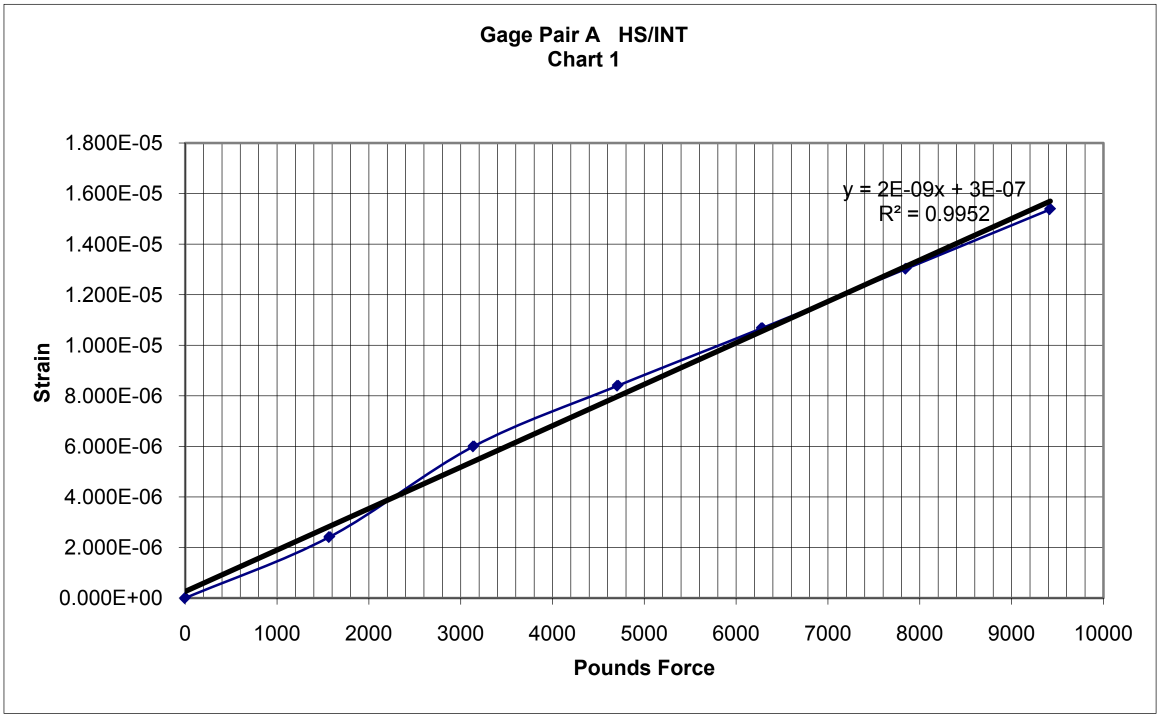


Figure U.3. Representative strain/force diagram for the involute housing—HS outside position

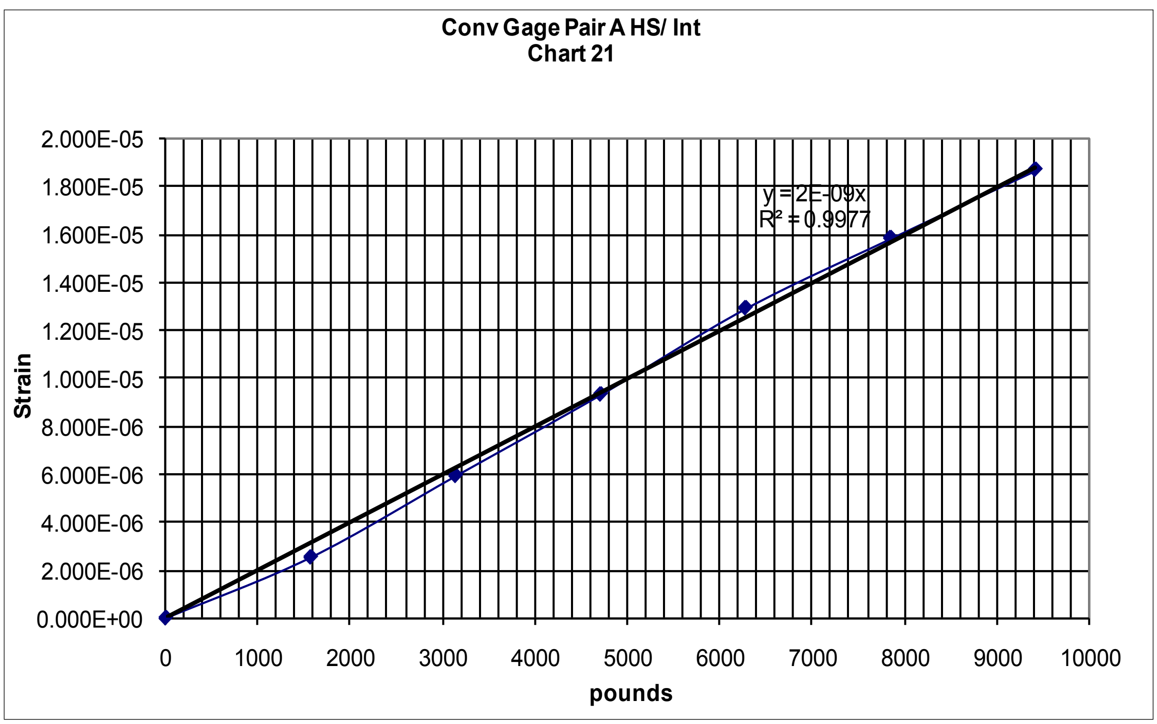


Figure U.4. Representative strain/force diagram for the Convoloid housing—HS inside position

When pressure (load) was applied to both cylinders, strain readings were inconsistent with the amount of applied load. Figure U.5 shows the same strain gage as Figure U.3 being loaded with both cylinders. Strain is predictable until the HS cylinder is increased without increasing the LS.

For the strain gages between the LS and INT bores, values vary drastically. In Figure U.6, as pressures were increased, strain readings for one of the LS strain gages show a compressive load under low load conditions. The strains then reverse and show the expected tensile relationship until the relationship between the two hydraulic cylinders varies (when the HS pressure increases to 3,000 psi and the LS pressure remains constant at 2,500 psi).

Multiple trials were conducted to ensure that the data variations were not due to damage to the housings and/or the strain gages. The tests were repeatable, however, including the pressure/strain relationship of one of the Convoloid LS gages (Figure U.7) where the strain went to zero at the maximum loaded condition.

Conclusions

Although the forces applied to the housings were intended to span the range of loads developed by the operational test, there was some concern that the housings might have been overloaded during the static test. Review of the strain readings from both tests reveals that the amounts of strain realized from the static test were much lower (by nearly an order of magnitude in some cases) than those seen in the operational test. Therefore it is assumed that the housings were not deformed excessively with the static test.

Because the loads applied by the hydraulic cylinders were in the range of the calculated separating loads, the additional strain appearing in the operational test can only be explained by other gear forces (tangential load and axial, or thrust, load). An example of the disparity between the static test and the operational test can be seen by comparing strain values in the range of the final test load (240% of rated torque) which can be approximated by the hydraulic loads at 2,000 psi.

During the operational test, strain between the HS and INT bearings (outside) was 20.2 micro-strain (E-06) and during the static test a strain of 17.2 was realized—well within the expected range. For the other side of the housing, the operational strain was 77.3 compared to a static test value of 5.9. This disparity can only be explained by the presence of substantial thrust induced by the gearing of the operational test.

Similar comparisons occur between the INT and LS bores where the outside strain values for the operational test were 136 compared to 4.4 on the static test. The strain values for the inside position were much closer with 35 on the operational test and 27 on the static test.

It appears that the amount of housing strain must be greatly affected by the thrust component of the gear loads. This is a force that was not modeled nor induced by the hydraulic cylinders. Additional forces (e.g., tangential loads and bearing setting forces) also can affect strain readings.

Due to the great disparity in strain readings, an appropriate comparison with values from the operational test cannot be made, and the nonlinear relationship of the curves precludes any extrapolation. To be truly meaningful, the calibration should have been carried out with a good bearing set and an operational set of gears in their respective housings with torque arms attached to the input and output shafts. This method would have provided the thrust component to the strain gages.

This test and the findings thereof have emphasized to Genesis Partners personnel and their associated subcontractors and consultants the need for more thorough analysis and planning prior to conducting similar tests.

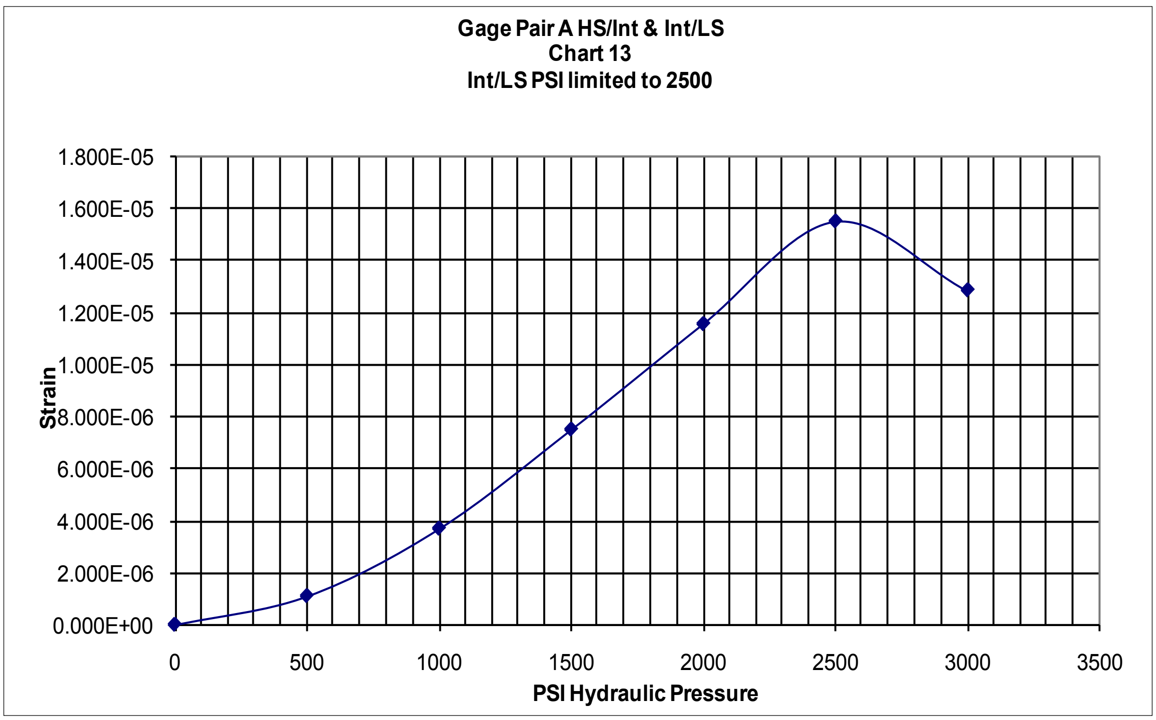


Figure U.5. HS (outside) strain gage reading under combined load—involute

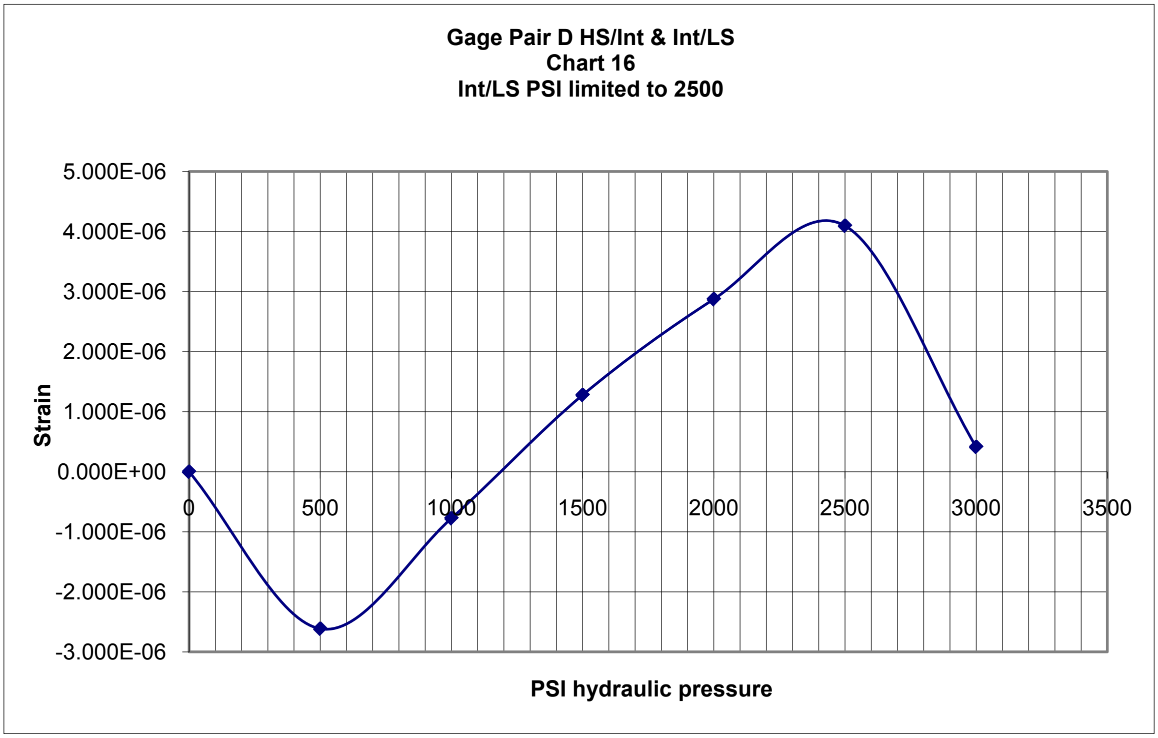


Figure U.6. LS (inside) strain gage reading under combined loading—involute

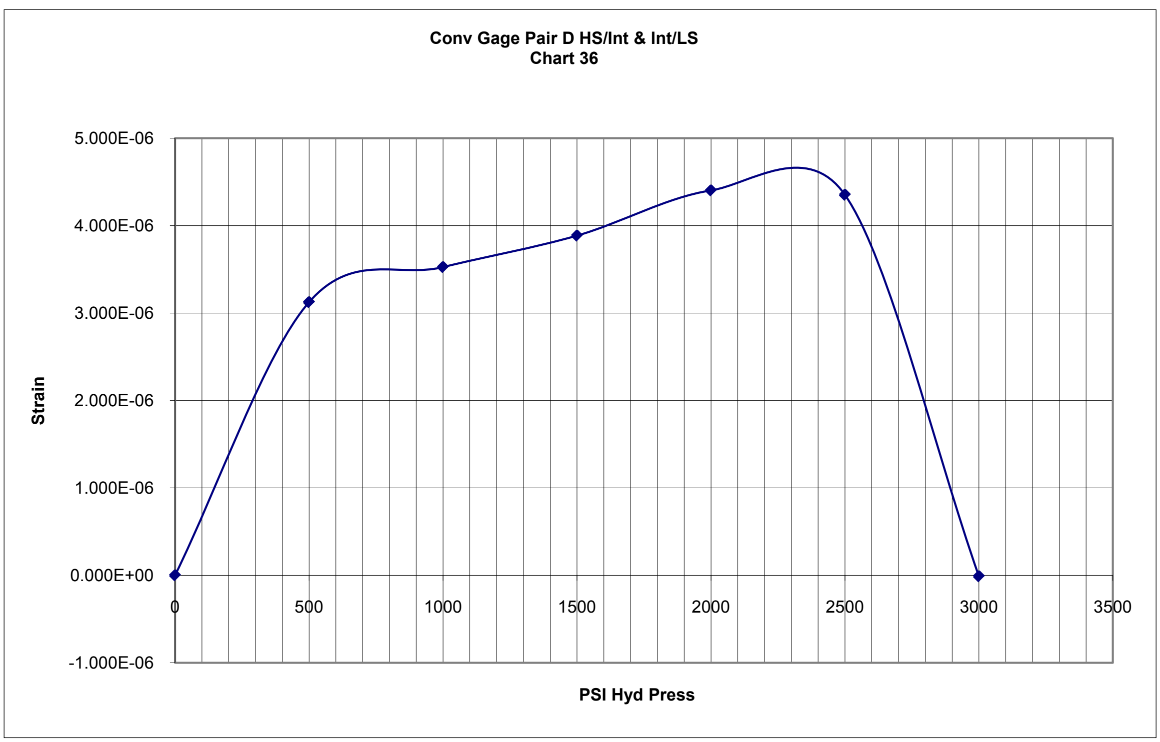


Figure U.7. LS (outside) strain gage reading under combined load—Convoloid

Appendix V. Tutorial on Convoloid Gearing Technology

Description

Convoloid represents a gear technology intellectual portfolio (IP) encompassing the design, stress analysis, manufacture, and inspection of gear teeth of a totally new design. The involute curve, first proposed for use in gearing in 1754, has been used almost exclusively for the production of geared power transmissions. Continued analysis and field experience has refined this tooth form to a high degree. The relatively recent proliferation of computers has made possible the rapid optimization of most factors involved to markedly increase the power density of gear drive systems.

Trademark

The trademark Convoloid is owned by Genesis Partners LP.

Tooth Form

The Convoloid tooth form is computer generated and has the following primary characteristics.

- Convex/concave contact of meshing tooth surfaces
- Conjugacy
- Optimized relative curvatures
- Carefully selected curvature values

An example of the Convoloid tooth form is given in Figure V.1.

Architectures

Convoloid gear pairs primarily are intended for parallel axis helical gear architectures. Included in these architectures are straightforward helical-parallel axis pairs either speed increasing or speed decreasing, planetaries including internals and other epicyclic arrangements. The latter arrangements are especially well suited to the more compact gear system architectures of wind turbine gearboxes.

Advantages

Advantages over involutes include:

- Considerably lower surface durability stresses,
- Considerably lower tooth bending stresses,
- Enhanced entraining velocities and other tribological characteristics, and
- Manufacturing and inspection utilize the existing capital asset infrastructure.

Patents

Basic patents have been filed and issued in the United States and many other countries around the world. Additional related patent applications have been submitted and others will be submitted on a continuing basis.

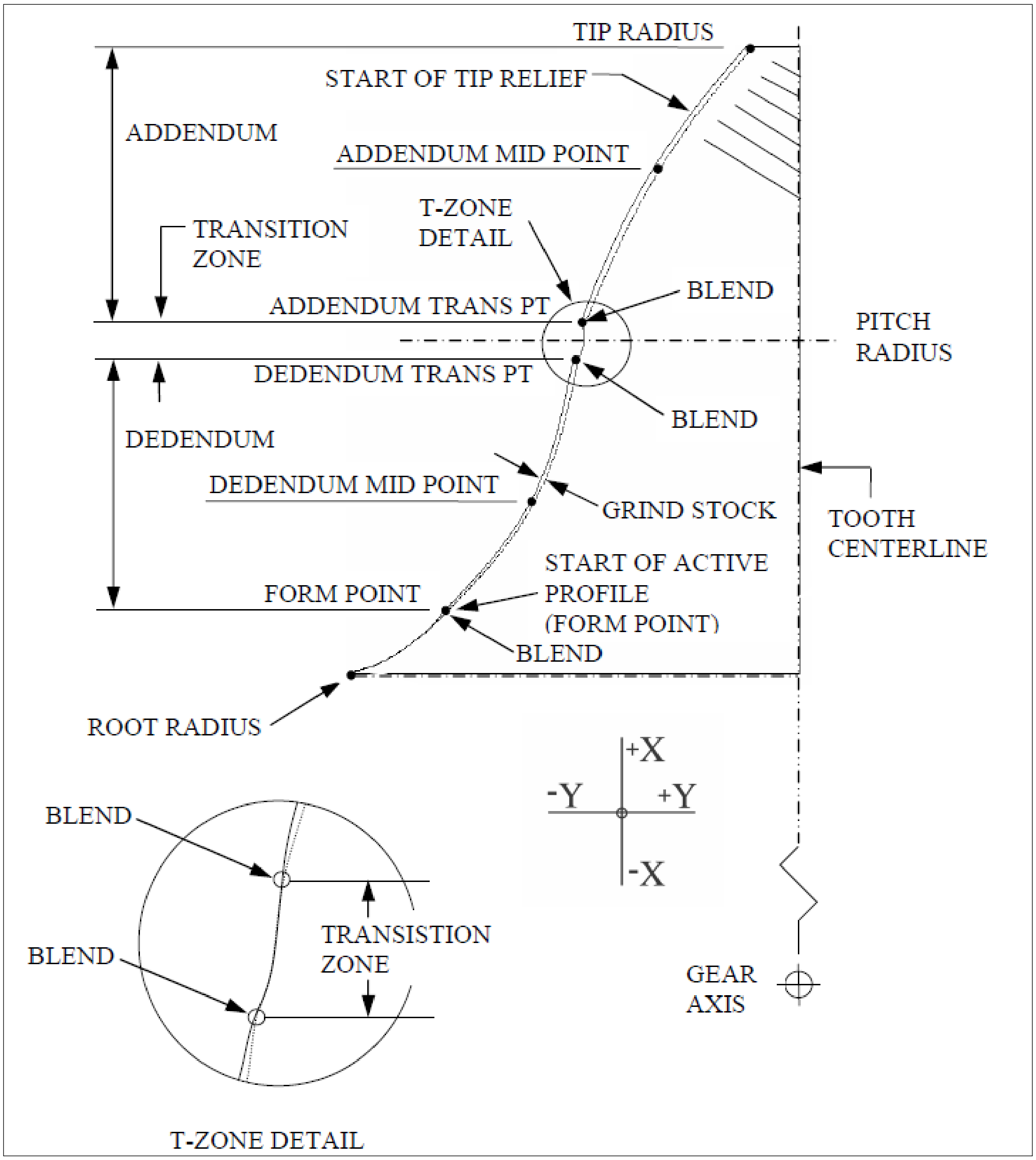


Figure V.1. Graphical display of the Convoloid tooth form

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This report presents the results of a study conducted by Genesis Partners LP as part of the United States Department of Energy Wind Energy Research Program to develop wind technology that will enable wind systems to compete in regions having low wind speeds. The purpose of the program is to reduce the cost of electricity from large wind systems in areas having Class 4 winds to 3 cents per kWh for onshore systems or 5 cents per kWh for offshore systems. This work builds upon previous activities under the WindPACT project, the Next Generation Turbine project, and Phase I of the Low Wind Speed Turbine (LWST) project. This project is concerned with the development of more cost-effective gearing for speed increasers for wind turbines.

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gear teeth; low wind speed turbine; 108-kW wind turbine gearboxes; low wind speed; cost-effective gearing for speed increasers for wind turbines

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