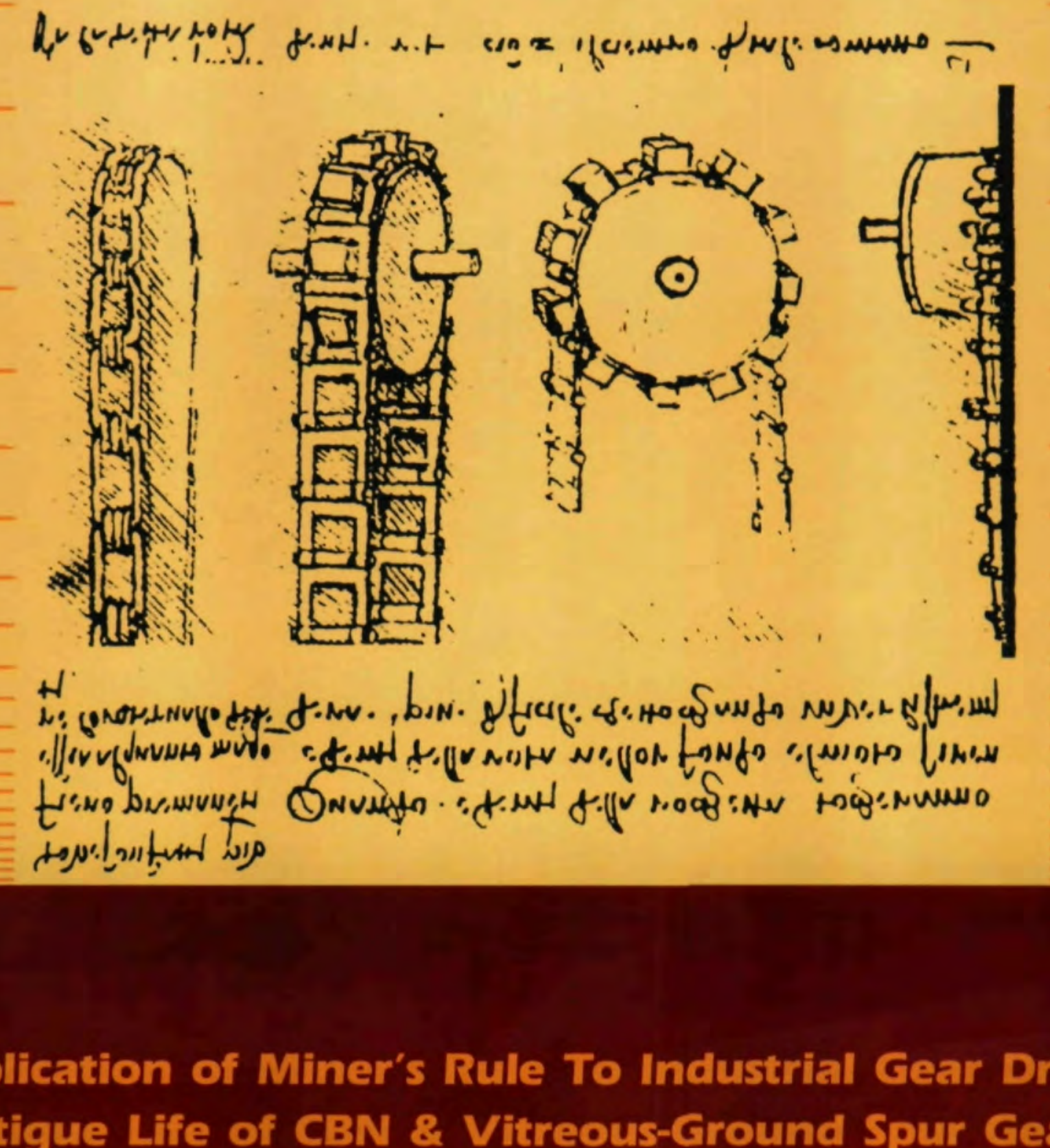
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

JANUARY/FEBRUARY 1990



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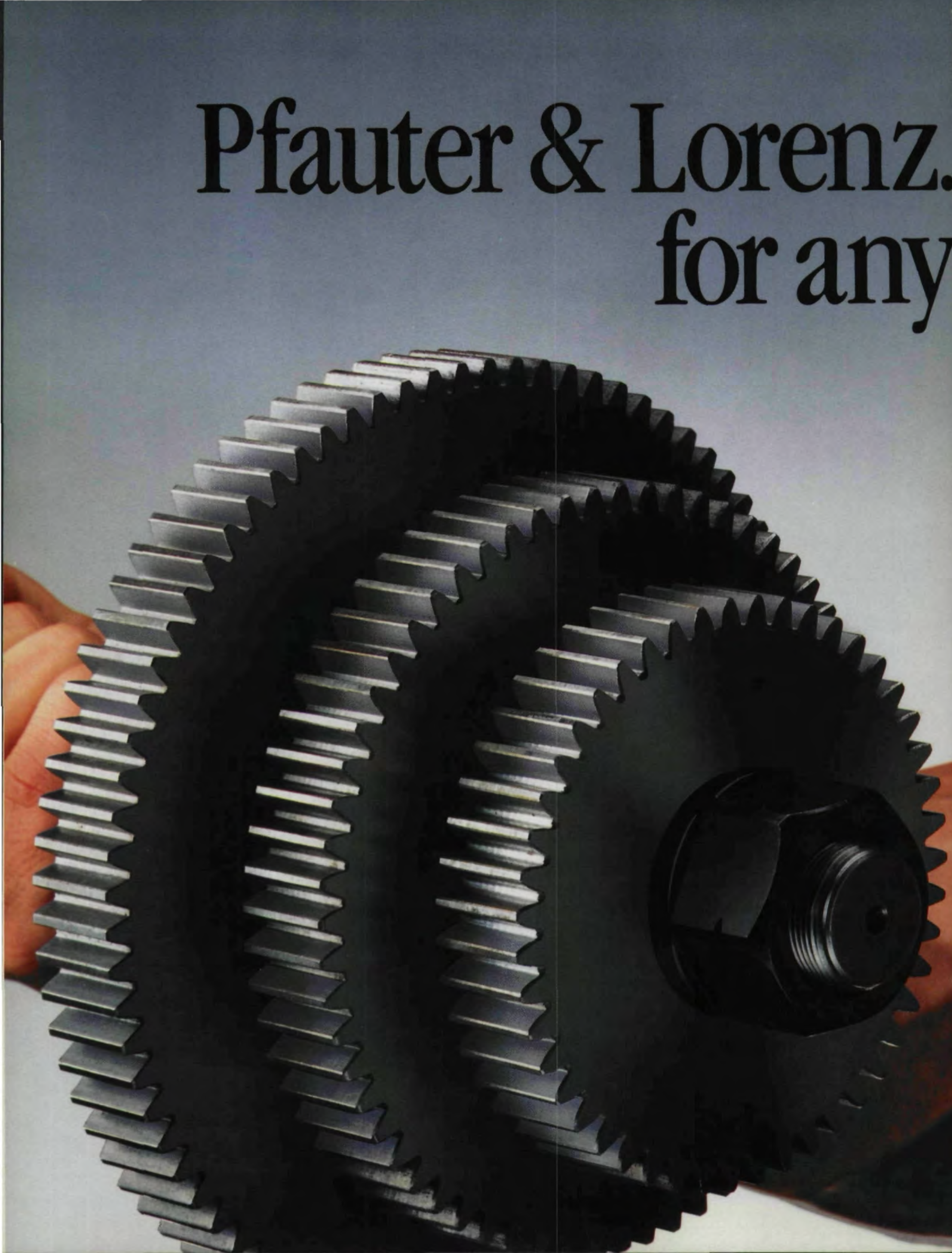
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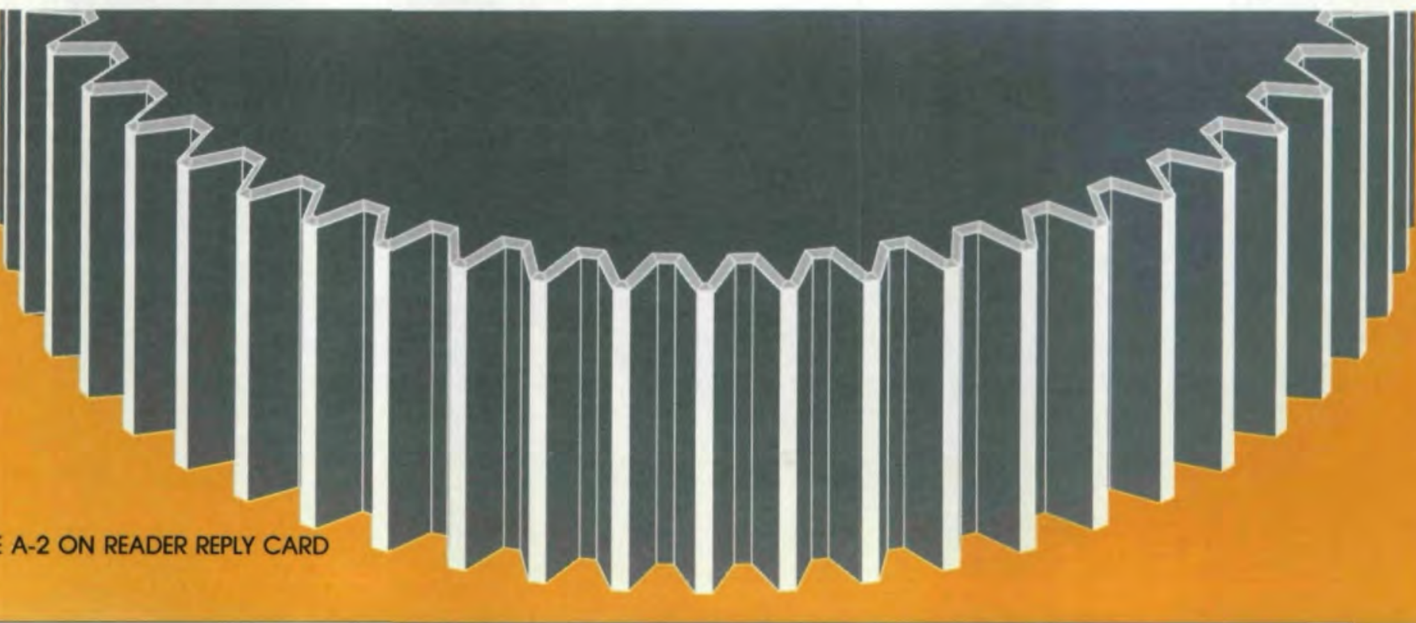

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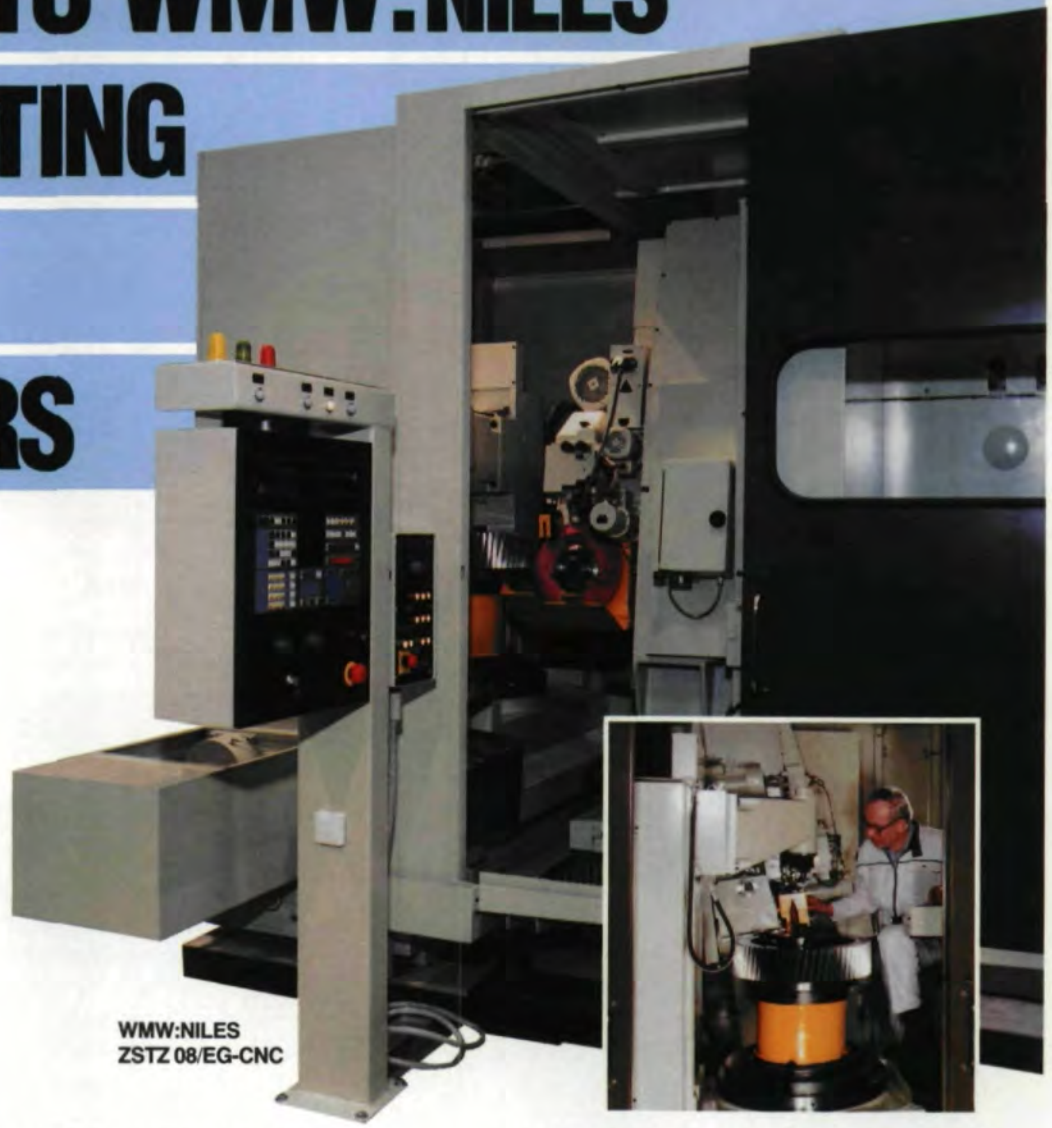

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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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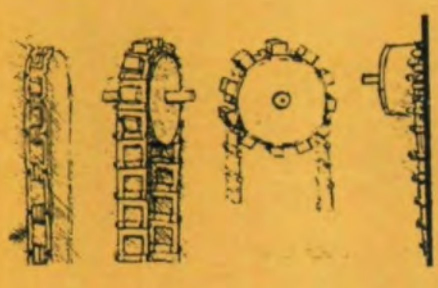

The Advanced Technology of Leonardo da Vinci 1452-1519

COVER

Drive chain for Leonardo's bicycle. Like many other of Leonardo's ideas, this one was never built until centuries after his death. Note the square teeth on this drive wheel, which make the design impractical. Later sketches show that Leonardo reconsidered and began thinking about the use of rounded teeth.

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GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published bimonthly by Randall Publishing Co., Inc., 1425 Lunt Avenue, P. O. Box 1426, Elk Grove Village, IL 60007. GEAR TECHNOLOGY, The Journal of Gear Manufacturing. Subscription rates are: $40.00 in the United States, $50.00 in Canada, $55.00 in all other foreign countries. Second-Class postage paid at Arlington Heights, IL and at additional mailing office.

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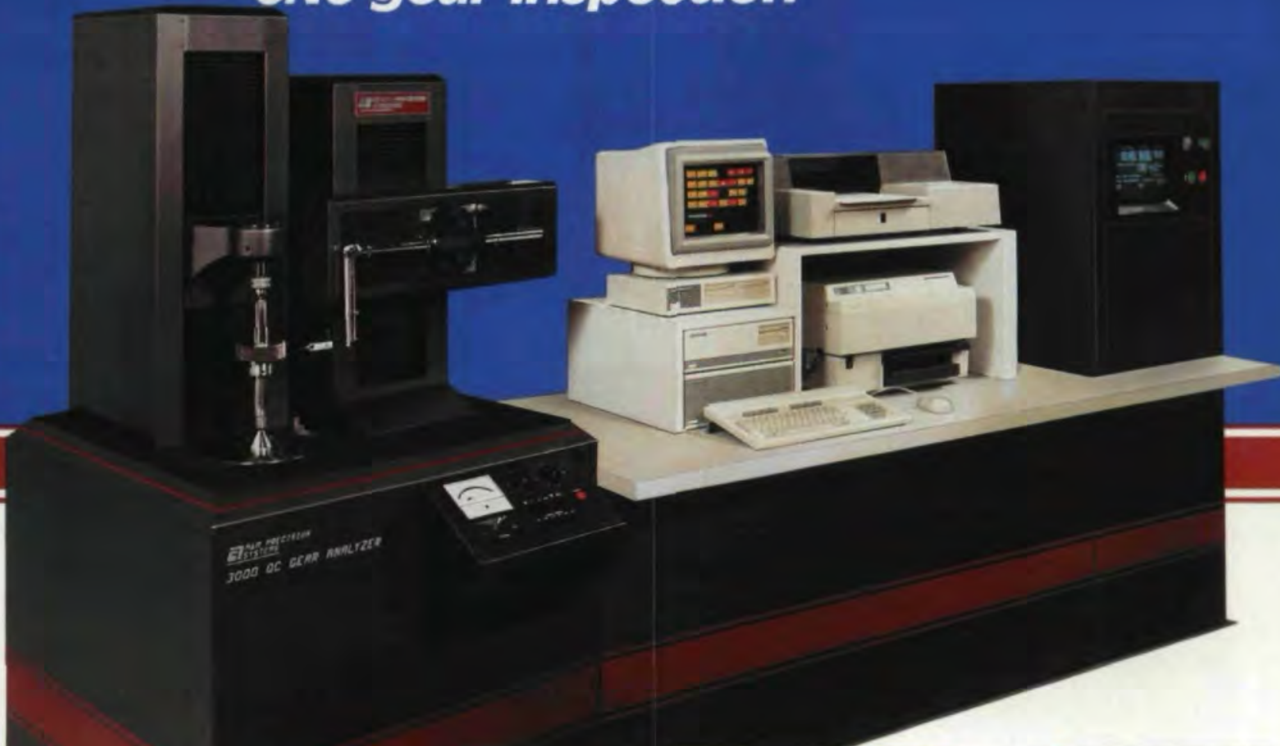

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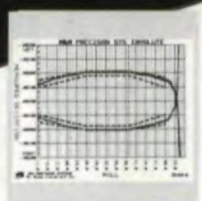

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January 23, 1990, Cincinnati, OH
Gear Failure Analysis
March 6, 1990, Cincinnati, OH
Rational Gear Design and Lubrication
May 8, 1990, Los Angeles, CA
Controlling the Carburizing Process
June 5-6, 1990, Alexandria, VA
Gear System Design for Noise Control

MAY 1-3, 1990. Houston Tool & Manufacturing Exposition, George R. Brown Convention Center, Houston, TX. Both national and local exhibitors of CAD/CAM equipment, robotics, communication and information systems, precision machine parts, and quality control systems. For more information, contact: Houston Chapter, NTMA, 515 Post Oak Blvd., Suite 320, Houston, TX 77027. (713) 439-5890.

SEPTEMBER 5-13, 1990. IMTS-90, McCormick Place, Chicago, IL. Largest trade show in the Western hemisphere with exhibitors from around the world. A new feature this year will be the 120,000 sq. ft. Forming and Fabricating Pavilion. For more information, contact: IMTS-90, 7901 Westpark Drive, McLean, VA 22102-4269. Ph:(703) 893-2900. Fax: (703) 898-1151.

Editorial

KUDOS TO AGMA FOR PITTSBURGH SHOW

AGMA's Gear Expo '89 was, by all accounts, a great success, proving again the wisdom of having a trade show devoted exclusively to gearing and gear-related products. Over 1500 people attended the show, and 86 different companies exhibited their goods and services.

Pittsburgh proved to be a truly beautiful town, surprising those of us who had visions of a grim "Steel City". Its modern, attractive downtown area, the spectacular vistas offered by the confluence of the three rivers, its restaurants and hotels, and the Lawrence Convention Center itself all provided good memories of this show.

Attendance at the '89 show was just about equal to that of two years ago. This may have been because of Pittsburgh's location, which required many more people to fly in and stay overnight to attend. Two years ago, because of the Sunday hours and the Cincinnati location, many people were able to come for an afternoon and return the same day. Keeping in mind the need to keep the show as accessible as possible to the majority of people from the gear engineering and manufacturing fields, AGMA plans to rotate future expos between Detroit, Indianapolis, and Cincinnati, cities a little closer to the heart of the gear industry.

But quantity of attendees is not the only criterion for success to consider. It was my impression that this year's show attracted more key decision makers – the kind of people who were able to influence buying choices or, in many cases, to make commitments to purchase right on the show floor. So I think the steady attendance and the increased sales can be counted as a trade-off.

Given the good reports from our own people who attended and from others, it seems unfair to suggest that anything at the show was less than 100%. I do, however, have a couple of "I-wish-they-had's" and some suggestions for future shows.

I wish more of the major manufacturers had committed to showing their machinery instead of merely bringing pictures and literature. I sympathize with concerns about cost and location, but at this point, the AGMA show has more than proven itself in terms of its capability to draw interested buyers. Detroit in 1991 is the time to take advantage of this audience in the context of a show devoted exclusively to gearing.

I wish manufacturers exhibiting at the show would do a better job of telling the marketplace what equipment they plan to show. This information would be a good drawing card. For example, this year Klingelnberg and M & M both

Rick Norment, (left) Executive Director, Michael Goldstein, and James Partridge, AGMA President, at the Gear Technology booth.

had good crowds around their gear checkers. How many more people would have been drawn to the show and to these booths had they known that there was an opportunity to do serious, hands-on comparison shopping for this type of machine?

I wish exhibitors were given a stronger voice in practical, basic matters like show length, days of the week, daily hours for the show, amount of time needed for setup, overlap with the Technical Conference, etc. While none of these matters were a serious hindrance to the show's success this year, I think improvement could be made in some areas. It's important to consult all exhibitors – both large and small – about these details. Concern for their convenience is important. They, after all, are the people paying the bills. Without the exhibitors, there is no show.

But these are all minor quibbles. Overall, Gear Expo '89 was a rousing success, and congratulations are in order to everyone at AGMA and at the exhibitors' companies who worked hard to make it that way.

I, for one, consider this year's success a good beginning for an even better show in 1991.

Michael Goldstein,
Publisher

Editors' Note: We have received a great number of favorable comments about our editorial in the Nov/Dec issue and requests for additional copies. Reprints of this editorial are available on request from the editorial office.

Surface Fatigue Life of CBN and Vitreous Ground Carburized and Hardened AISA 9310 Spur Gears

Dennis P. Townsend
NASA Lewis Research Center, Cleveland, OH
P.R. Patel
Bell Helicopter Textron, Forth Worth, TX

Abstract:

Spur gear surface endurance tests were conducted to investigate CBN ground AISI 9310 spur gears for use in aircraft applications, to determine their endurance characteristics and to compare the results with the endurance of standard vitreous ground AISI 9310 spur gears. Tests were conducted with VIM-VAR AISI 9310 carburized and hardened gears that were finish ground with either CBN or vitreous grinding methods. Test conditions were an inlet oil temperature of 320 K (116°F), an outlet oil temperature of 350 K (170°F), a maximum Hertz stress of 1.71 GPa (248 ksi), and a speed of 10,000 rpm. The CBN ground gears exhibited a surface fatigue life that was slightly better than the vitreous ground gears. The subsurface residual stress of the CBN ground gears was approximately the same as that for the standard vitreous ground gears for the CBN grinding method used.

Introduction

Grinding of carburized and hardened gear teeth for aircraft application has been standard practice for many years. Grinding is required to produce the required accuracy and surface finish necessary for improved life, reduced noise, and dynamic loads for aircraft gears. Until a few years ago, the method used for grinding hardened gears was the standard vitreous grinding wheel. The vitreous grinding method typically produces a very shallow compressive stress [<0.013 mm (0.0005 in.)] on the surface of the ground part, but has very little effect on the subsurface compressive residual stress.

A few years ago cubic boron nitride (CBN) grinding wheels were introduced for grinding gears and other parts.[1] The CBN grinding wheel allows a much greater rate of stock removal of hardened parts without producing the grinding burns that are prevalent with vitreous grinding. The CBN crystals have a high thermal conductivity compared to the vitreous material and conduct the heat away from instead of into the part. In addition, the CBN crystals are very sharp and very hard and produce a chip-like cutting action. When a hardened gear or other part is ground very hard with considerable force, a subsurface residual compressive stress is developed below the surface.[2] This subsurface residual compressive stress has been shown to improve the subsurface fatigue life of gears and bearings.[3,4] The CBN grinding of carburized and hardened AISI 9310 steel spur gears should, therefore, produce equivalent or improved surface fatigue life.

The objectives of the research reported herein were (1) to investigate CBN grinding as a method for finishing aircraft-type gears; (2) to determine the surface endurance characteristics of CBN ground carburized and hardened AISI 9310 steel spur gears; (3) to compare the results with standard virtreous ground carburized and hardened AISI 9310 steel spur gears. To accomplish these objectives, tests were conducted with two groups of gears manufactured from one lot of material. One group of spur gears from that lot were CBN ground. For comparison purposes, the other group of spur gears were manufactured by vitreous grinding. The gears had a gear pitch diameter of 8.89 cm (3.50 in.) and 3.2 module (8 diametrial pitch). Test conditions included an oil inlet temperature of 320 K (116°F) that resulted in an oil outlet temperature of 350 K (170°F), a maximum Hertz stress of 1.71 GPa (248 ksi), and a shaft speed of 10,000 rpm.

AUTHORS:

MR. D.P. TOWNSEND *is a gear consultant for NASA and numerous industrial companies. Townsend earned a BSME from the University of West Virginia. During his career at NASA he has authored over eighty papers in the gear and bearing research area. For the past several years, he has served in active committee roles for ASME. Presently he is a member of the ASME Design Engineering Executive Committee.*

MR. P.R. PATEL *is a principal engineer at Bell Helicopter Textron, Ft. Worth, TX. He has worked in the areas of materials and process control for aircraft drives systems. He holds an M.S. in Metallurgical Engineering from the University of Minnesota and an M.B.A. from the University of Dallas. Mr. Patel is a member of the American Society for Metals and the American Helicopter Society.*

Apparatus and Procedures

GEAR TEST APPARATUS – The gear fatigue tests were performed in the NASA Lewis Research Center's gear fatigue test apparatus (Fig. 1a). This test rig uses the four-square principle of applying the test gear load so that the input drive only needs to overcome the frictional losses in the system.

A schematic of the test rig is shown in Fig. 1(b). Oil pressure and leakage flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, torque is applied to the shaft. This torque is transmitted through the test gears back to the slave gear, where an equal but opposite torque is maintained by the oil pressure. This torque on the test gears, which depends on the hydraulic pressure applied to the load vanes, loads the gear teeth to the desired stress level. The two identical test gears

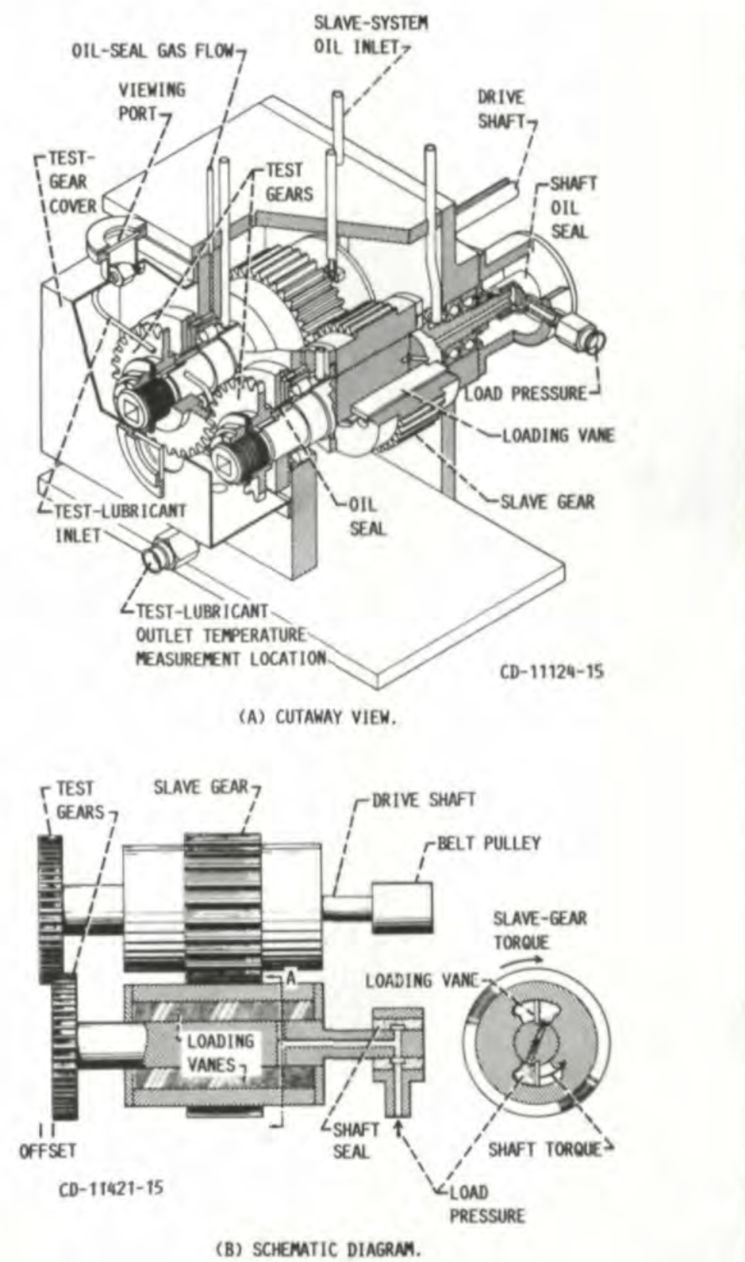


Fig. 1 – NASA Lewis Research Center's Gear Fatigue Test Apparatus.

Table I. — Gear Data
[Gear tolerance per AGMA Class 12.]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3927)
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Chordal tooth thickness reference, cm (in.)	0.485 (0.191)
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Outside diameter, cm (in.)	9.525 (3.750)
Root diameter, cm (in.)	7.988 (3.145)
Root fillet, cm (in.)	0.10 to 0.15 (0.04 to 0.06)
Measurement over pins, cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash reference, cm (in.)	0.025 (0.010)
Tip relief, cm (in.)	0.0013 (0.0005)
Tooth width, cm (in.)	0.64 (0.25)

Table II. — Grinding Data
For Vitreous and CBN Ground Spur Gears

	Wheel speed, rpm	Grit size	Finish μm ($\mu in.$)	Table speed, sec/pass	Number of passes per tooth	Depth of cut per pass	Time to grind one gear
Vitreous	1600	60	0.36 (14)	2	36	0.018 mm (0.0007 in.)	15 hr
CBN	3400	70	0.30 (12)	6	5	0.13 mm (0.005 in.)	20 min

Table III. — Data for Gear Used for Residual Stress Measurements

Number of teeth	31
Diametral pitch	8.5
Pressure angle, deg	22
Pitch diameter, cm (in.)	9.264 (3.647)
Face width, cm (in.)	3.386 (1.333)

Table IV. — Chemical Composition of Test Materials by Percent Weight

Element	AISI 9310 gears
Carbon (core)	0.10
Manganese	.60
Phosphorus	.006
Sulfur	.005
Silicon	.24
Copper	.04
Chromium	1.35
Molybdenum	.16
Vanadium	.01
Nickel	3.37
Iron	Balance

Table V. — Heat Treat Procedure for Test Gears

Pre-carburize heat treatment	
Normalize	1725°F for 1 hr Air cool
Harden	1500°F for 1 hr Oil quench
Temper	1000°F for 4 hr
Carburize	1700°F for 6.5 hr 1.0 percent carbon potential
Post-carburize heat treatment	
Sub-critical anneal	1150°F for 2 hr Air cool
Harden	1500°F for 1 hr Oil quench
Sub-zero treat	−115°F for 4 hr
Temper	300°F for 4 hr Air cool

can be started under no load, and the load can be applied gradually without changing the running track on the gear teeth.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubricant systems are separated at the gearbox shafts by pressurized labyrinth seals. Nitrogen is the seal gas. The test gear lubricant is filtered through a 5 μm nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The skin temperature of the heater is controlled to prevent overheating the test lubricant.

A vibration transducer mounted on the gearbox is used to automatically shut off the test rig when a gear surface fatigue occurs. The gearbox is also automatically shut off if there is a loss of oil flow to either the main gearbox or the test gears; if the test gear oil overheats; or if there is a loss of seal gas pressurization.

The belt-driven test rig can be operated at several fixed speeds by changing pulleys. The operating speed for the tests reported herein was 10,000 rpm.

TEST GEARS – A photograph of the test gears is shown in Fig. 2. The dimensions of the gears are given in Table I. All gears had a nominal surface finish on the tooth face of 0.2 μm (8 μ in.) rms or better. Typical surface finish charts for both grinding methods are shown in Fig. 3. All gears have a standard 20° involute profile with tip relief. The tip relief was 0.0013 cm (0.0005 in.) starting at the highest point of single tooth contact. One group of gears was ground with a vitreous grinding wheel with speed, feed, and metal removal rate as shown in Table II. The second group of gears were ground with a CBN form grinder with speed, feed, and metal removal rate as shown in Table II.

Residual stress profiles were established using a gear configuration described in Table III, to determine the difference between the two grinding techniques. For baseline condition, one gear was tested in as-carburized condition. The stress measurements were made using x-ray diffraction technique at the approximate pitch diameter of the gears. The results of residual stress measurements are summarized in Fig. 4.

TEST MATERIAL – The gears were manufactured from vacuum induction melted, vacuum arc remelted (VIM VAR) AISI 9310 steel. The nominal chemical composition of the gears is given in Table IV. The heat treatment procedure for the test gears is given in Table V. The case and core properties of the test gears are given in Table VI. Photomicrographs of the case and core of a test gear are given in Figs. 5a and b.

TEST LUBRICANT – All the gears were lubricated with a single batch of synthetic paraffinic oil, which was the standard test lubricant for the gear tests.

Fig. 2 – Test Gear Configuration.

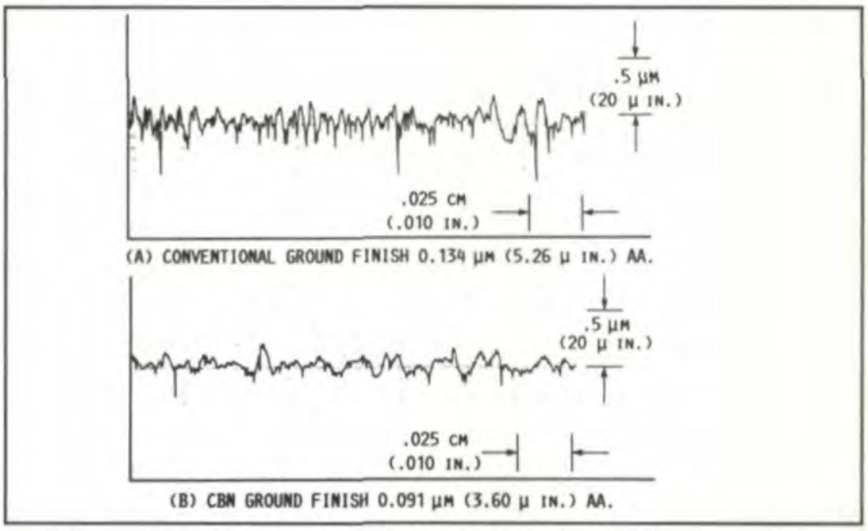


Fig. 3 – Surface Finish Measurement in Profile Direction With .010 Cutoff.

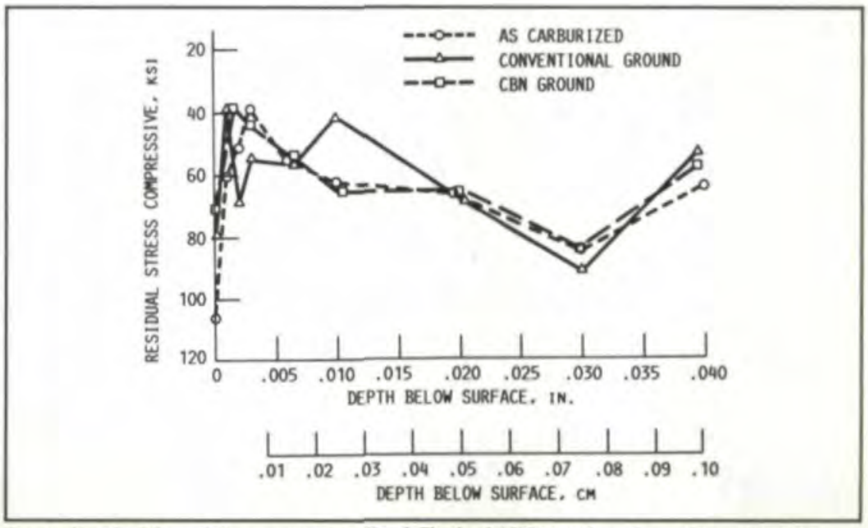
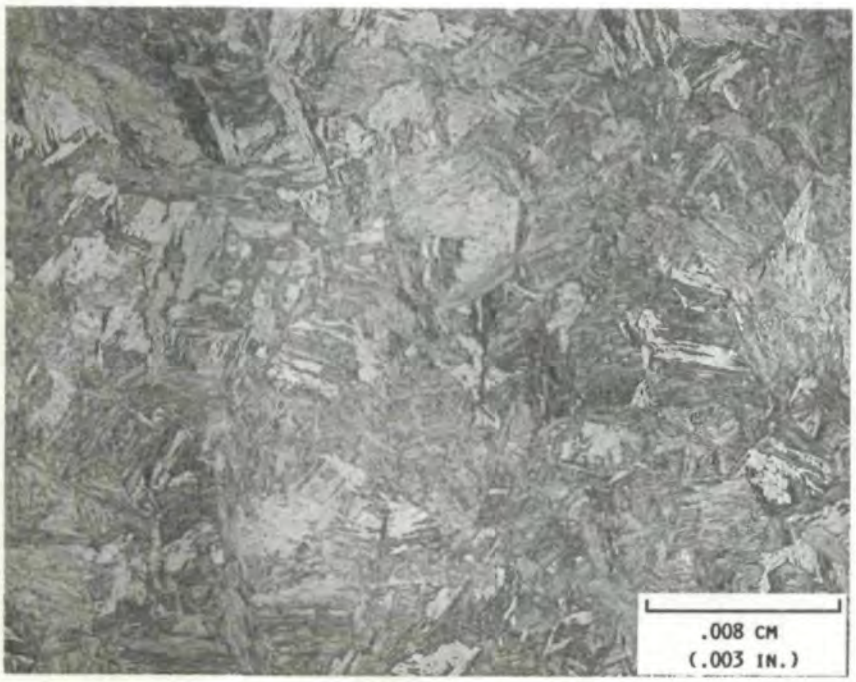


Fig. 4 – Residual Stress Measurements on Tooth Flank of AISI 9310 Spur Gears Ground by Vitreous and CBN Grinding Wheels.

Table VI. — Case and Core Properties of Test Gears

	Conv. Ground	CBN Ground
Surface hardness, HRC	61.5	63.0
HRC 60 depth, mm (in.)	0.45 (0.018)	0.53 (0.021)
HRC 50 depth, mm (in.)	0.99 (0.039)	0.97 (0.038)
Core hardness, HRC	38.0	38.0
Retained austenite, percent	6.0	6.6

(A) Case



(B) Core

Fig. 5—Photomicrographs of the Case and Core Material for Test Gears.

The physical properties of this lubricant are summarized in Table VII. Five percent of an extreme pressure additive, *designated Lubrizol 5002 (partial* chemical analysis given in Table VII) was added to the lubricant.

TEST PROCEDURE— After the test gears were cleaned to remove their protective coating, they were assembled on the test rig. The test gears were run in an offset condition with a 0.30 cm (0.120 in.) tooth-surface overlap to give a surface load width on the gear face of 0.28 cm (0.110 in.); thereby allowing for an edge radius on the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each set of gears. All tests were run in at a load per unit width of 1230 N/cm (700 lb/in.) for 1 hr. The load was then increased to 5800 N/cm (3300 lb/in.), which resulted in a 1.71 GPa (248 ksi) pitch line maximum Hertz stress. At the pitch line load the tooth bending stress was 0.21 GPa (30 ksi) if plain bending was assumed. However, because there was an offset load, there was an additional stress imposed on the tooth bending stress. Combining the bending and torsional moments gave a maximum stress of 0.26 GPa (37 ksi). This bending stress did not include the effects of tip relief, which would also increase the bending stress.

Since the offset test method may introduce edge loading effects, the method was originally checked with and without crowned gears. There was no difference between crowned and uncrowned gears. Also all fatigue spalls with uncrowned gears originate evenly along the tooth flank and never start at the edge location. This is proof that the offset test condition is an acceptable method for surface fatigue testing.

Operating the test gears at 10,000 rpm gave a pitch line velocity of 46.55 m/sec 9163 ft/min). Lubricant was supplied to the inlet mesh at 800 cm^3/min (49 $in.^3$/min) and 320±6 K (116±10°F). The lubricant outlet temperature was nearly constant at 350±3 K (170±5°F). The tests ran continuously (24 hr/day) until the rig was automatically shut down by the vibration detection transducer (located on the gearbox adjacent to the test gears) or until 500 hours of operation without failure were completed. The lubricant circulated through a 5 μm fiberglass filter to remove wear particles. For each test, 3.8 liters (1 gal.) of lubricant were used. At the end of each test, the lubricant and filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder.

The pitch line elastohydrodynamic (EHD) film thickness was calculated by the method of Reference 5. It was assumed for this film thickness calculation that the gear surface temperature at the pitch line was equal to the outlet oil temperature and that the inlet oil temperature to the contact zone was equal to the gear temperature, even though the oil jet inlet temperature was considerably lower. It is possible that the gear surface temperature was even higher than the oil outlet temperature, especially at the end points of sliding contact. The EHD film thickness for these conditions was computed to be 0.33 μm (13 $\mu in.$), which gave an initial ratio of film thickness to composite surface roughness (h/σ) of 1.15 at the 1.71 GPa (248 ksi) pitch line maximum Hertz stress.

Each pair of gears was considered as a system and, hence, a single test. Test results were evaluated using Weibull plots calculated by the method of Johnson.[6] (A Weibull plot is the number of stress cycles versus the statistical percent of gear system failed.)

Results and Discussion

One lot of VIM VAR AISI 9310 steel spur gears was divided into two groups and endurance tested. One group was ground by a vitreous grinding wheel, while the second group was ground by a CBN form grinding wheel. Test conditions consisted of a tangential tooth load of 5800 N/cm (3300 lb/in.), which produced a maximum Hertz stress of 1.7 GPa (248 ksi), and a speed of 10,000

Table VII. – Lubricant Properties

Property	Synthetic Paraffinic oil *plus additives**
Kinematic viscosity, cm^2/sec (c) at:	
244 K (−20°F)	2500×10^{-2} (2500)
311 K (100°F)	31.6×10^{-2} (31.6)
372 K (210°F)	5.5×10^{-2} (5.5)
477 K (400°F)	2.0×10^{-2} (2.0)
Flash point, K (°F)	508 (455)
Fire point, K (°F)	533 (500)
Pour point K (°F)	219 (−65)
Specific gravity	0.8285
Vapor pressure at 311 K (100°F), mm Hg (or torr)	0.1
Specific heat at 311 K (100°F), J/(kg) (K); Btu/(1b)(°F)	2190 (0.523)

*Additive Lubrizol 5002 (5 vol%): phosphorus 0.03 vol%; sulfur, 0.93 vol %.

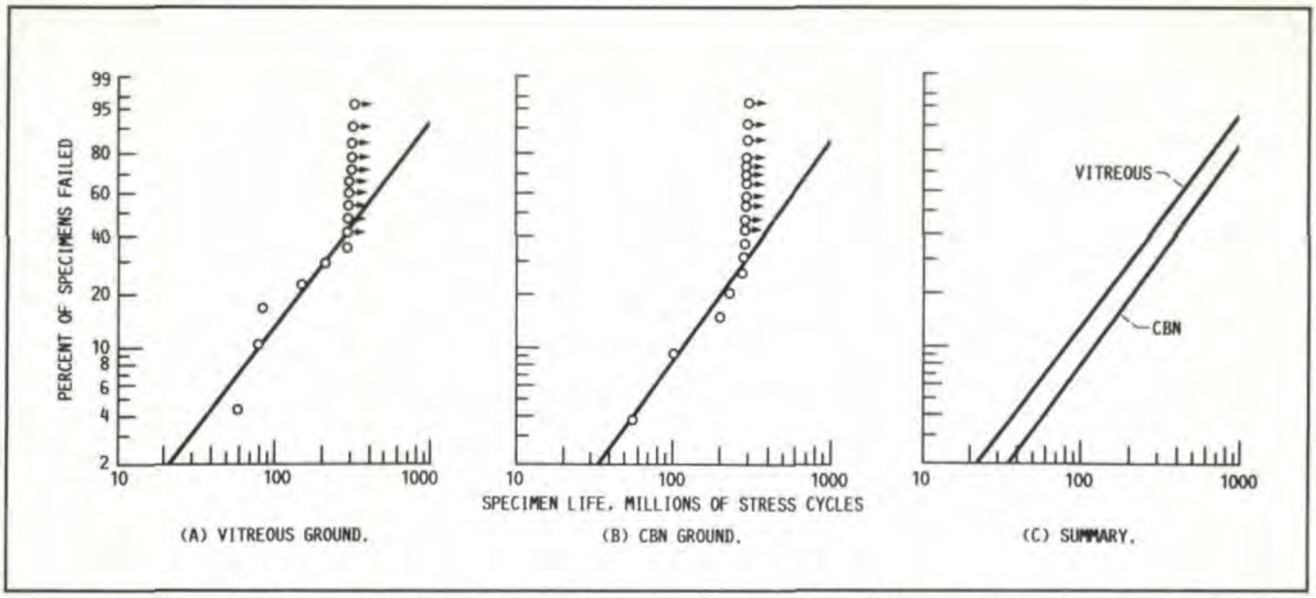


Fig. 6 – Surface Fatigue Life of Carburized, Hardened, Ground and Shot Peened Test Gears. Speed 10,000 rpm; Maximum Hertz Stress, 1. 71 GPa (248 ksi); Temperature, 350 K (170° F); Lubricant, Synthetic Paraffinic With 5% E P Additive.

Fig. 7 – Typical Fatigue Spall for AISI 9310 Gears.

rpm. The gears failed by classical subsurface pitting fatigue. The pitting fatigue life results of these tests are shown in the Weibull plots of Fig. 6 and are summarized in Table VIII.

Pitting fatigue life results for the gears that were ground by the vitreous grinding wheel are shown in Fig. 6a. The 10 and 50% lives were 82.5×10^6 and 371×10^6 stress cycles (137 and 618 hr), respectively. The Weibull slope was 1.25. The failure index (i.e., the number of fatigue failures out of the number of sets tested) was 6 out of 16. A typical fatigue spall that occurs near the pitch line is shown in Fig. 7. This is a typical fatigue spall similar to those observed in rolling element fatigue tests. The pitch line pitting is the result of a high subsurface shearing stress which develops subsurface cracks. These subsurface cracks propagate into a crack network which results in a fatigue spall that is slightly below the pitch line, where the sliding condition is more severe.

Pitting fatigue life results for the gear systems that were ground by a CBN form grinder are shown in Fig. 6b. The 10 and 50% surface fatigue lives were 122.7×10^6 and 502×10^6 stress cycles (205 and 837 hr), respectively. The Weibull slope was 1.34. The failure index was 7 out of 18. The 10% surface fatigue life of the CBN ground gears was ∿1-½ times that of the standard vitreous ground gears. The confidence number was 60%, which indicates that there are 600 chances out of 1000 tests that the 10% life of the CBN ground gears will be superior to the 10% life of the vitreous ground gears. This indicates that there is not a lot of statistical significance to the life difference between the two groups of gears. However, it does indicate that the CBN gears are at least equivalent in life to the vitreous ground gears or slightly better. The equivalent residual stress profile of the two methods of grinding would also indicate that the fatigue life should be approximately the same. A more vigorous CBN grinding could induce some additional compressive residual stress;

Table VIII. — Spur Gear Fatigue Life Results

[Pitch diameter, 8.89 cm (3.50 in.); maximum Hertz stress, 1.71 GPa (248 ksi); speed, 10,000 rpm; lubricant, synthetic paraffinic oil; gear temperature, 350 K (170°F).]

Material	Gear system life, revolutions		Weibull slope	Failure index*	Confidence number at 10% level**
	10% life	50% life			
Standard ground VIM-VAR AISI 9310	82.5×10^6	371×10^6	1.25	6 out of 16	–
CBN ground VIM-VAR AISI 9310	122.7×10^6	502×1^6	1.34	7 out of 18	60

*Number of surface fatigue failures out of number of gears tested.

**Percentage of time that 10% life obtained with AISI 9310 gears will have the same relation to the 10% life obtained with Ex-53 gears or CBS 1000 M.

thereby, improving the surface fatigue life.[2-3] A summary of the fatigue lives of the two groups of ground gears are given in Fig. 6c.

Summary of Results

Spur gear endurance tests were conducted to investigate CBN ground AISI 9310 spur gears for use in aircraft gear applications, to determine their endurance characteristics and to compare the results with the endurance of standard vitreous ground AISI 9310 spur gears. Tests were conducted with VIM-VAR AISI 9310 carburized and hardened gears that were finished ground with either CBN or vitreous grinding methods. Test conditions were an inlet oil temperature of 320 K (116°F), an outlet oil temperature of 350 K (170°F), a maximum Hertz stress of 1.71 GPa (248 ksi), and a speed of 10,000 rpm. The following results were obtained:

1. The CBN ground gears exhibited a surface fatigue life that was slightly better than the vitreous ground gears.
2. The subsurface residual stress of the CBN ground gears was approximately the same as that for the standard vitreous ground gears for the CBN grinding method used.

References

1. DeVRIES, R.C. "Cubic Boron Nitride: Handbook of Properties," REPT-72CRD178, General Electric Co., Schenectady, NY, June 1972. (Avail. NTIS, AD-907330L).
2. KIMMET, G.J. "CBN Finish Grinding of Hardened Spiral Bevel and Hypoid Gears," AGMA Paper 84FTM6, American Gear Manufacturers Association, Alexandria, VA, Oct. 1984.
3. JOHNSON, G.A. & RATTERMAN, E. "Enhanced Product Performance Through CBN Grinding." *Gear Technology*, Vol. 5, No. 5, Sept/Oct 1988.
4. COE, H.H. & ZARETSKY, E.V. "Effect of Interference Fits on Roller Bearing Fatigue Life," *ASLE Trans.* Vol. 30, No. 2, Apr. 1987, pp. 131-140.
5. DOWSON, D. & HIGGINSON, G.R. *Elastohydrodynamic Lubrication, The Fundamental of Roller and Gear Lubrication.* New York: Pergamon Press, 1966.
6. JOHNSON, L.G. *The Statistical Treatment of Fatigue Experiments.* New York: Elsevier Publication Co., 1964.

Acknowledgement: *Originally published as NASA Technical Memorandum 100960 and AVSCOM Technical Report 88-C-019. Reprinted with permission.*

Application of Miner's Rule to Industrial Gear Drives

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Introduction

We need a method to analyze cumulative fatigue damage to specify and to design gear drives which will operate under varying load. Since load is seldom constant, most applications need this analysis.

Service and application factors have been used to approximate the effect of variable load, but they can give poor results when we extrapolate experience with one design, such as a through-hardened parallel shaft reducer, to a replacement design of different configuration or material, such as a carburized planetary reducer to drive the same machine. They can also be unreliable in estimating the size of gear reducers required for a new application, as in the following wind turbine example.

One of the reasons for this weakness is that the slope of the S-N curve affects the fatigue life and the amount of damage done at each stress level. When we change steels, we should change service factors.

When existing similar drives are satisfactory and no change in design concept is contemplated, service factors can be an adequate method of sizing industrial gear units. When we make changes from the design or operating conditions which generated the original service factors, we need to be very conservative.

When operating conditions or material properties are better known, Miner's rule provides a superior method of estimating gear size and performance.

Miner's Rule

Although Fuchs and Stevens (1980) called the concept of cumulative fatigue damage a "useful fiction", experience has shown that components subjected to varying loads do, in fact, fail in a manner which is consistent with cumulative fatigue damage. The linear-cumulative-fatigue-damage rule was first proposed by Palmgren (1924) for predicting ball bearing life and independently by Miner (1945) for predicting the fatigue life of aircraft components. They introduced the simple idea that if a component is cyclically loaded at a stress level that would cause fatigue failure in 10^5 cycles, then each cycle consumes one part in 10^5 of the life of the component. If the loading is changed to a stress level that causes failure in 10^4 cycles, each of these cycles consumes one part in 10^4 of the life, and so on. When the sum of the individual damages equals 1.0, fatigue failure is predicted. In equation form, Miner's Rule is

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \cdots + \frac{n_i}{N_i} = 1 \qquad (1)$$

where:

n_i = number of cycles at the ith stress.
N_i = number of cycles to failure at the ith stress.

$\frac{n_i}{N_i}$ = damage ratio at the ith stress.

If the fraction of cycles at each stress is known rather than the actual number of cycles, the cycles are given by

$$n_i = \alpha_i{}^*N \qquad (2)$$

where

α_1 = cycle ratio (fraction of cycles at the ith stress).
N = resultant fatigue life (total cycles).

Miner's Rule may be rewritten as

$$\frac{\alpha_1{}^*N}{N_1} + \frac{\alpha_2{}^*N}{N_2} + \cdots + \frac{\alpha_i{}^*N}{N_i} = 1 \qquad (3)$$

which may be solved for the resultant life:

$$N = \frac{1}{\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \cdots + \frac{\alpha_i}{N_i}} \qquad (4)$$

The cycle ratio may be obtained from the load spectrum by

$$\alpha_i = \frac{n_i}{\Sigma n_i} \qquad (5)$$

where

n_i = number of cycles at the ith load in the load spectrum.
Σn_i = total number of cycles in the load spectrum.

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The number of cycles at each load is calculated from

$$n_i = 60^* w_i^* t_i \tag{6}$$

where

w_i = speed at the ith load (rpm).
t_i = time at the ith load (hour).

The equivalent (baseline) speed is given by

$$w_b = \frac{1}{\frac{\alpha_1}{w_1} + \frac{\alpha_2}{w_2} + \cdots + \frac{\alpha_i}{w_i}} \tag{7}$$

The resultant life in hours is

$$L = \frac{N}{60^* w_b} \tag{8}$$

The use of Miner's rule for gears was described by Hapeman (1971). Appendices to AGMA 170.01-1976, "Design Guide for Vehicle Spur and Helical Gears," and AGMA 218.01-1982, "Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth," also describe its use.

Method

The application of Miner's rule to gear drives requires knowledge of the load, usually a cyclic, repetitive pattern which can be closely analyzed; actual gear geometry from a trial design or the final design; gear material S-N curve.

The repetitive pattern of the load data allows it to be divided arbitrarily into sections, summing the loads and cycle counts into a load spectrum. Fig 1. shows the results graphically. It is assumed that the pattern is repeated throughout the life of the gear set. The load spectrum is shown in form suitable for computer input in Table 1.

Table 1
Load spectrum arranged for computer input.

Load Spectra at 1400 AMP Limit

Load Segment In KIPS	Spectrum 1 Time, Sec	Spectrum 1 %	Spectrum 5 Time	Spectrum 5 %	Spectrum 12 Time	Spectrum 12 %	Average %	Cycles per Index
120>100	3.0	8.11	2.0	5.41	1.2	3.24	5.59	19.33
100>80	9.0	24.32	6.5	17.57	5.0	13.51	18.47	63.90
80>60	11.8	31.89	15.3	41.35	12.8	34.59	35.95	124.37
60>40	6.0	16.22	6.8	16.22	9.0	24.32	18.92	65.46
40>0	7.2	19.46	7.2	19.46	9.0	24.32	21.08	72.94
Total	37.0	100.00	37.8	100.00	37.8	100.00	100.00	346.00

It is important to note that as the loads are grouped, the individual loads are all assumed to be the same value as the maximum for that group. In the interest of accuracy, the subdivisions of groups should be narrow for higher loads where most of the fatigue damage is done. It is also important to include occasional peak loads, since they can be very damaging.

Various cycle-counting techniques such as the Range-Pair, Rainflow and Racetrack methods are described by Nelson (1978) and Fuchs (1980) to convert complicated load spectrums into simplified histograms. Most of these methods were developed for analysis of structural members where stress does not return to zero at each application of the load. For gear teeth it is usually sufficiently accurate to count each load application as a cycle.

In most transmissions it is possible for the same tooth to see the peak load at each repetition of the load spectrum. In some low speed gears, such as the final drive gear of the microwave antenna in Example 4, the peak load may not be applied to the same tooth at each repetition.

Each gear in the machine is checked to find which has the shortest life. The authors know no shortcut way to do this. A computer is indispensable to handle the voluminous calculations of bending stress, pitting stress, resultant lives at those stresses and the summation of those lives for each loading condition and each gear in the transmission.

Example 1: Wind Turbine Speed Increaser

A wind turbine, Fig. 2, must turn at a constant speed to maintain the correct frequency of the electrical power that it generates. The wind speed is far from constant and many gusts exceed 50 miles per hour. The inertia of the wind turbine rotor smooths small wind gusts, but larger variations in wind speed are usually accommodated by pitching the blades of the rotor. Many wind turbines have a computer to control the generator speed to less than 1% variation.

A gearbox is used to increase the rotor speed (typically less

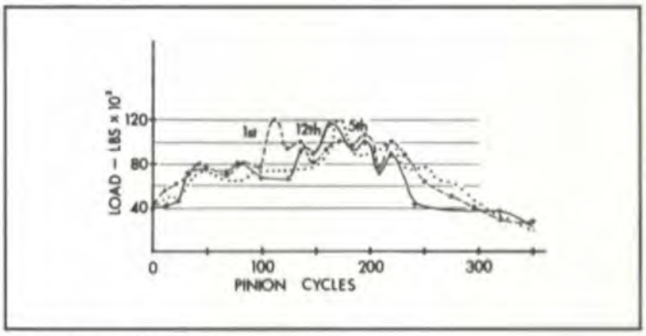

Fig. 1 – Typical load spectrum.


Fig. 2 – Wind turbine generators.

than 100 rpm) to the speed of the generator (usually 1800 rpm). The gearbox loads are non-uniform due to wind gusts and aerodynamic turbulence of the rotor, causing the entire system of rotor, drive train, generator and tower to vibrate. Each time a rotor blade passes the "shadow" of the tower, the gearbox experiences a torque pulsation. Because the vibration is so severe, standard industrial practice cannot be used for a wind turbine gearbox.

At one wind farm, several thousand gearboxes of two different designs were installed side by side. One of the designs survived, but the other failed prematurely. Inspection of the failed low-speed gears has shown that they were manufactured with excessive lengthwise crowning, which reduced the effective face width and increased the load on the central portion of the teeth. As part of the failure analysis, the low-speed gear set was rated per AGMA 218.01 using actual measured loads.

Field measurements of the load on a wind turbine were made over a four month period. The reaction torque was measured by applying strain gages to the torque arm of the shaft-mounted gearbox. Data was collected on a self-contained, microprocessor-based recorder. The transducer was calibrated by statically loading the rotor with known loads. Data were collected by storing the number of peaks occurring in fifteen discrete bins of equal increments of torque. The strain signal from the torque arm transducer was converted to shaft torque by multiplying by the calibration constant.

The load histogram is included in Appendix 1. The load ratio was calculated by dividing the torque at each of the sampling bins by the torque corresponding to 100 kw generator output power. The cycle ratio was calculated by dividing the number of counts in each bin by the total number of counts.

Fig. 3 – Container crane.

The expected life of the drive is 50,000 hours. The Miner's Rule rating of the low-speed gear indicates that its pitting and bending fatigue life should be more than adequate if its helix is properly modified. However, with excessive crown the load distribution factor increases from Cm = 1.3 to as high as Cm = 2.6, and both pitting and bending fatigue lives drop to approximately 100 hours. These calculated results correlate with field experience where gears with proper crowning survive for years of operation, while those with excessive crown fail in a few hundred to several thousand hours.

Example 2: Container Crane Main Hoist

The gearing for the main hoist of a container crane, Fig. 3, has a spectrum of loads because some of the time it must lift only the spreader (the device which attaches to the top of the container), and at other times it must lift both the spreader and a container which ranges from 10 to 40 long tons, depending on its size. Some main hoist systems consist of dual cable-winding drums with twin drive trains. In these cases, the load on one of the gear trains is increased if the loads in the container are off center. The duty cycle also influences the loads on the gearing; sometimes the container crane will only be used to either unload or load a ship, while at other times it will both unload and load. In the first case, the gearing is only fully loaded for one half the time, while in the second case it is loaded all the while the trolley travels from the ship to the dock and back again.

The Federation Européene de la Manutention "Rules for the Design of Hoisting Appliances" gives the load spectrum shown in Fig. 4. It considers hoisting motions with and without useful loads. In the figure, δ represents the useful load of container and its contents, and γ represents the weight of the spreader, head block, sheaves and portions of the lifting ropes. Fig. 4 is based on a typical application where

$$\delta = 90{,}000 \text{ lb (40 T container)}$$
$$\gamma = 30{,}000 \text{ lb (spreader, head block, etc.)}$$

$$\delta + \gamma = 120{,}000 \text{ lb}$$
$$2/3^*\delta + \gamma = 90{,}000 \text{ lb}$$
$$1/3^*\delta + \gamma = 60{,}000 \text{ lb}$$

Fig. 4 also shows an actual load spectrum determined from records of container weights for a particular crane at the Port of Oakland obtained over a one-year period. It shows that the F.E.M. spectrum is conservative for this example because fully loaded, maximum size containers were rarely encountered.

The following example demonstrates a load spectrum for a main hoist where the motor speed varies with the lifted load. (See Table 2.) It is based on the percent times given in the F.E.M. specification, and it shows that percent time is not the same as percent cycles when the speed varies.

Table 2
Main Hoist Load Spectrum

Load No.	Power P (kW)	Speed w_i (rpm)	Time t (hr)	Torque T_i (lb-in)	Cycles n_i (10^8)	Load Ratio β_i (T_i/T_{max})	Cycle Ratio α_i $n_i/\Sigma n_i$
1	560	650	3750	72720	1.4625	1.0000	0.0831
2	560	850	3750	55610	1.9125	0.7647	0.1087
3	560	1240	5000	38120	3.7200	0.5242	0.2114
4	340	1400	12500	20260	10.5000	0.2786	0.5968
		$\Sigma t_i = 25000$			$\Sigma n_i = 1.7595 \times 10^9$		1.0000

The main hoist cable-winding drum is driven by a DC electric motor through a parallel shaft, single helical, three stage speed reducer. The overall ratio is 23/1.

The load histogram (See Appendix 2.) was calculated based on the F.E.M. specification. Required life is 25,000 hours.

Equivalent (baseline) speed:

$$w_b = \frac{1}{\frac{\alpha_1}{w_1} + \frac{\alpha_2}{w_2} + \frac{\alpha_3}{w_3} + \frac{\alpha_4}{w_4}} \tag{9}$$

$$= \frac{1}{\frac{.0831}{650} + \frac{.1087}{850} + \frac{.2114}{1240} + \frac{.5968}{1400}}$$

$$= 1173 \text{ rpm}$$

Baseline power:

$$P_b = \frac{(T_b)(w_b)}{63025} \tag{10}$$

$$= \frac{(72720)(1173)}{63025} = 1354 \text{ hp}$$

The Miner's Rule rating shows that % time is not the same as % cycles; i.e.,

Load No.	% time $t_i/\Sigma t_i$	% cycles $n_i/\Sigma n_i$
1	0.15	0.0831
2	0.15	0.1087
3	0.20	0.2114
4	0.50	0.5968

Miner's rule shows that the cubic mean load cannot be used for gearing; i.e.,

$$P_{eff} = P_b\,[\Sigma\beta_i^{e*}\alpha_i]1/e \tag{11}$$

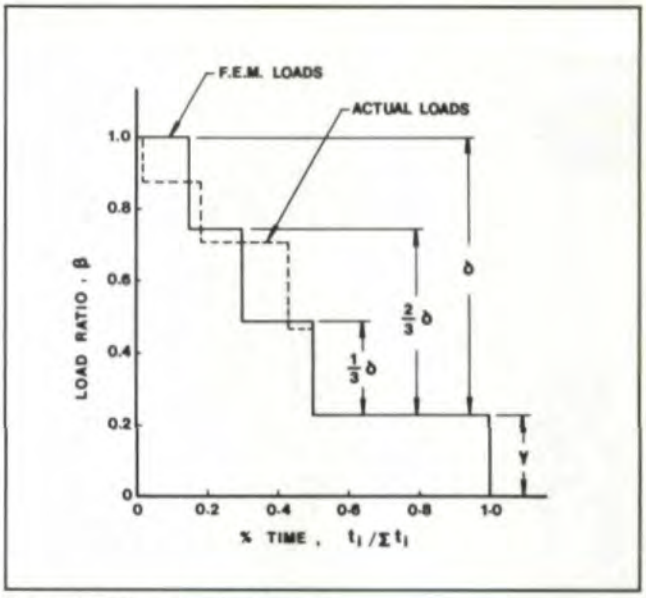

Fig. 4 – Container crane load spectra.

Using e = 3 (cubic mean) gives

$$P_{eff} = 1354\,[(1)^3(.0831) + (.7647)^3(.1087) + (.5242)^3(.2114) + (.2786)^3(.5968)]^{1/3}$$
$$= 757 \text{ hp}$$

Using e = 1/2(.056) = 8.93 (AGMA 218.01 Fig. 20, lower curve) gives

$$P_{eff} = 1354[(1)^{8.93}(.0831) + (.7647)^{8.93}(.1087) + (.5242)^{8.93}(.2114) + (.2786)^{8.93}(.5968)]^{1/8.93}$$
$$= 1038 \text{ hp}$$

Hence, using cubic mean load underestimates the effective load by a factor of 1.37.

Example 3: Train Positioner

Unit trains of about 100 cars, carrying 10,000 metric tons of coal and powered by five locomotives, Fig. 5, are used to haul coal to power stations and to the ports. The trains are more than 7000 feet long. The coal is dumped by rotating the cars, one or two at a time, around their couplings. The train is automatically positioned by a winch for each dumping sequence. A direct current mill motor drives the cable drum through a 68/1 ratio parallel shaft, single helical three-stage gear reducer. Four years after it was installed, the high speed pinion failed.

Fig. 5 – Unit coal train.

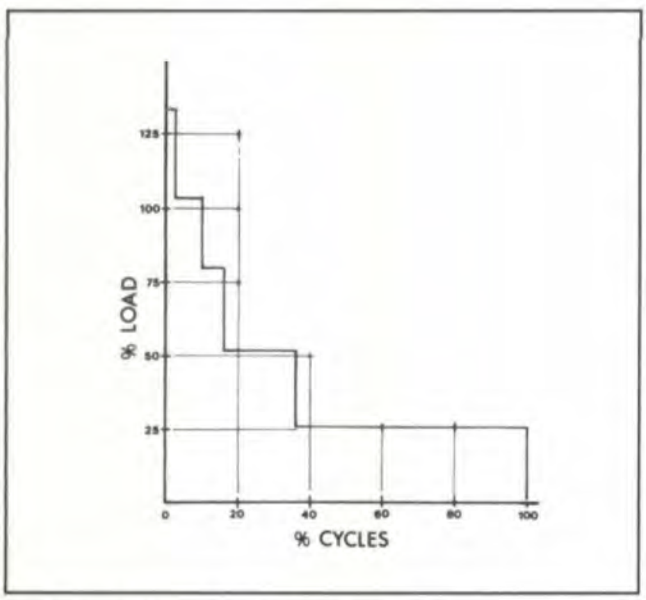

Fig. 6 – Load histogram for train positioner.

Fig. 7 – Positioner pinion after 5×10^7 cycles

A load histogram was abstracted from field measurement of load for a 106 car train. (See Appendix 3.) Motor current was measured with a recording ammeter which was calibrated against actual cable tension by a load cell in the cable anchor. Three sections, each representing one "car" of the complete ammeter recording, were analyzed. The graph was divided into zones representing 20% load bands. The time at which the measured load was in each band was measured from the charts and the three sets of data were averaged. Fig. 1 shows a similar load spectrum. The load histogram is shown in Fig. 6.

The required life was 10,000 trains = 1.06×10^6 cars = 3.6×10^8 pinion cycles under load.

Using Miner's rule, the calculated lives and modes of failure are:

Gear	Calculated Life		Mode
	Cycles	Hours	
1st Pinion	7.37×10^7	1580	Pitting
1st Gear	3.02×10^7	3050	Pitting

Only the input mesh is included in this example. The first input pinion failed by tooth fracture with heavy pitting after moving approximately 1400 trains or 5.6×10^7 pinion cycles. The calculated life of 7.4×10^7 cycles agrees reasonably well, indicating that this was an overload failure.

*The first pinion in a second drive was removed from serv*ice a year after the first pinion failed. It had moved approximately the same number of trains and was heavily pitted. (See Fig. 7.)

The designer of the positioner had made a cubic-mean-load analysis of the expected load spectrum and had sized the electric motor and the gear drive on the resulting load, with a service factor of 1.6. The electric motor has been maintenance free in this application, probably because it is thermally limited and has enough time to cool off between torque peaks. The pinions, which easily meet the 1.6 service factor rating, just weren't big enough to handle the load. The gear rating had to be increased by 50% to survive in this service.

The original through-hardened pinions have been replaced with carburized and ground parts, and the load has been reduced 30% by limiting the motor torque. Miner's rule predicts that with these changes the drive will give satisfactory service.

Example 4. Microwave Antenna.

Large microwave antennas, Fig. 8, whether they are used for satellite communication or for radar, are subjected to variable loads. Load spectra for these antennas come from historic weather data, combined with occasional high acceleration requirements to reach the stowage position and to pick up new satellites. Tracking antennas and radars are subjected to varying inertia loads as well. The forces required to achieve the required accelerations are established by measurement (strain gage or motor current) on the same or similar machines. The accelertion requirements, severity and frequency are usually established by a performance specification, based on the intended use of the machine.

The following example is typical of many antenna drives which see the heaviest loads on just a few teeth. It is an

Fig. 8 – Microwave antenna wth az-el mount.

azimuth-elevation mount, with a yoke which rotates on a vertical axis (azimuth motion) supporting the antenna on a horizontal axis (elevation motion). Separate ring gear sectors for each motion are driven by pairs of opposing gear drives to eliminate backlash. Direct current servomotors are controlled by a pointing system to sweep back and forth through a 105° sector of the sky.

In order to investigate the feasibility of converting a surplus antenna mount for this application, a Miner's rule study of the proposed gear train was undertaken. The load spectrum was estimated from the friction and inertia portions of a similar existing antenna's load spectrum. It is shown as Fig. 9. Both antennas are in enclosures, so no aerodynamic loads are encountered.

In this antenna, a right angle enclosed special gear reducer drives an exposed pinion which meshes with an external spur gear cut integral with a large roller bearing. The overall ratio is 300/1.

The required life is 3800 "scan cycles" of 56 tooth azimuth gear travel in each direction per day for 1000 days or approximately 14,000 loaded hours.

A graph of load vs. position (Az. gear tooth number) was calculated from operating test results on the identical antenna mount and adjusted mathematically for the higher accelerations required for this service. The graph was divided into zones representing acceleration and velocity steps. (See Fig. 9.) The pinion loads are different by the amount of torque bias required to control backlash.

A separate load spectrum was developed for the gear teeth because one gear tooth would only see the maximum load every "scan cycle" if the antenna were always trained in one direction. For this analysis, the antenna is assumed to be

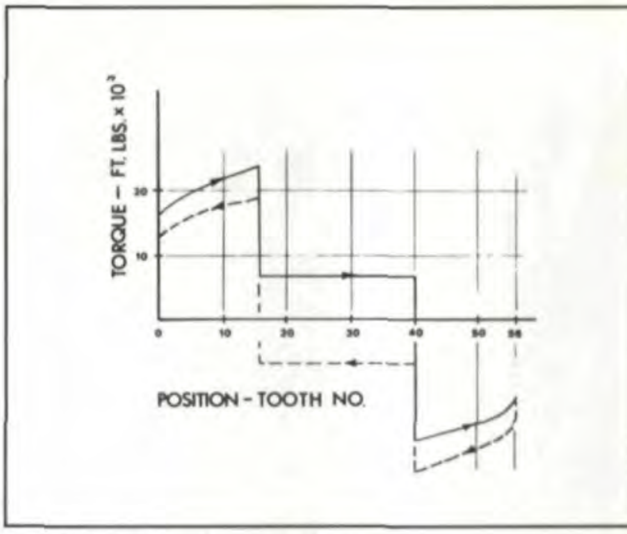

Fig. 9 – Load spectrum for radar antenna.

trained in random directions, averaging the load over the gear teeth. This is accomplished by the large "unload" block in the gear load spectrum.

In addition to the operating cycle, a maintenance cycle is included in the load spectrum. The loads are lighter than the operating cycle, so it does little damage to the gear teeth.

The load histogram is included in Appendix 4.

Only the output mesh is included in this example. The through-hardened output pinion had a calculated pitting life of less than 1000 hours under the predicted load spectrum, so the substitution of a carburized pinion was investigated. The carburized pinion has a satisfactory projected life, but the through- hardened azimuth gear limits the expected life of the drive to 6400 hours.

Significance of Peak Loads

The damage ratios shown in the examples, (Appendices 1-4) show that peak loads are very damaging, even if they operate for short times. They also show that peak loads are relatively more damaging to the bending fatigue life than to the pitting fatigue life. For this reason, gear tests that are accelerated by increasing the load are likely to accentuate bending fatigue.

Conclusions

- Miner's rule can be successfully applied to industrial gear drives.
- Peak loads cannot be ignored in gear life calculations because they frequently do the most damage even if they operate for short times.
- Peak loads are much more damaging to the bending fatigue life than the pitting fatigue life. For this reason, gear tests that are accelerated by increasing the load are likely to accentuate bending fatigue.
- If the operating speed varies, percent time does not equal percent cycles.
- The "cubic mean load" applies to ball bearings, but not to gears because their S-N curves have different shapes.

(continued on page 26)

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CIRCLE A-12 ON READER REPLY CARD

(continued from page 23)

Appendix 1
Data for Example 1 — Wind Turbine

Part A — Input Data Summary

Gear Geometry Data		Pinion	Gear
Tooth Number	NP, NG =	21.	104.
Net Face Width (In.)	F1, F2 =	4.7500	4.7500
Outside Diameter (In.)	do, Do =	4.3180	19.9490
Internal Gear I.D. (In.)	Di =		0.0000
Normal Diametral Pitch	Pnd =	5.5000	
Normal Pressure Angle (Deg.)	PHI(c) =	25.0000	
Standard Helix Angle (Deg.)	PSI(s) =	15.0996	
Operating Center Distance (In.)	C =	11.7700	
Gear Geometry Data For Pnd = 1.0			
Addendum Modification Coefficient	X1, X2 =	0.0000	0.0000
Thinning For Backlash Delta (sn1), Delta (sn2)	=	0.0240	0.0240
Stock Allow. Per Side For Finishing	Us1, Us2 =	0.0086	0.0086
Tool Geometry Data For Pnd = 1.0			
Tool Normal Tooth Thickness	tce1, tce2 =	1.5536	1.5536
Tool Addendum	hao1,hao2 =	1.3570	1.3570
Tool Tip Radius	rTel, rTe2 =	0.2670	0.2670
Tool Protuberance	Delta(o1), Delta(o2) =	0.0110	0.0110
Materials/Heat Treatment Data			
Modulus of Elasticity (PSI)	EP,EG =	30,000,000.	30,000,000.
Poisson's Ratio	MU(P), MU(G) =	0.3000	0.3000
Brinell Hardness	HBP, HBG =	654	543
Material (Code)	=	Steel (1)	Steel (1)
Material Grade	=	2	1
Heat Treatment (Code)	=	Carburized (4)	Ind. Hard (3)
Induction Hardening Pattern	=	N/A	A (1)
Load Data			
Transmitted Power (HP)	P =	134.0000	
Pinion Speed (rpm)	n(P) =	362.0000	
Gear Blank Temperature (Deg. F)	Tb =	200.	
Reliability	R =	0.9900	
Number of Contacts per Revolution	=	1	1
Reversed Bending?	=	N	N
Derating Factors			
Application Factor For Pitting Resistance	Ca =	1.0000	
Size Factor For Pitting Resistance	Cs =	1.0000	
Surface Condition Factor	Cf =	1.0000	
Load Dist. Factor For Pitting Resistance	Cm =	2.6000	
Dynamic Factor For Pitting Resistance	Cv =	0.9000	
Runtime Options			
Option Chosen For Calculating mN	=	Accurate	
Type of Analysis Chosen	=	Miner's Rule	
Curve Chosen	=	Lower	

Case Ident: Example 1 Wind Turbine
Program AGMA218 v.1.06B
Analysis Option: Miner's Rule

Part B – Hertzian Life – Pinion

Example Wind Loads

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
2.15	2.67E-06	295254.	7.81D+004	1.35D−003
2.01	3.3E-07	285479.	1.42D+005	9.17D−005
1.87	.000007	275358.	2.71D+005	1.02D−003
1.72	.00015	264083.	5.72D+005	1.04D−002
1.58	.00279	253108.	1.22D+006	9.03D−002
1.44	.0184	241634.	2.80D+006	2.60D−001
1.29	.0653	228703.	7.47D+006	3.46D−001
1.15	.1079	215936.	2.08D+007	2.05D−001
1.01	.1161	202366.	6.64D+007	6.92D−002
.86	.0944	186735.	2.79D+008	1.34D−002
.72	.0978	170861.	1.36D+009	2.84D−003
.57	.1146	152025.	1.10D+010	4.13D−004
.43	.1402	132042.	1.36D+011	4.08D−005
.29	.1416	108437.	4.58D+012	1.22D−006
.14	.10075	75343.	3.05D+015	1.31D−009
	1.0000			1.0000

Baseline Hertzian Stress Sc = 2.01D+005 • Resultant Hertzian Life Nc = 3.96D+007 Cycles • Resultant Hertzian Life Nc = 1.82D+003 Hours

Part C – Bending Life – Pinion

Example Wind Loads

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
2.15	2.67E-06	138092.	1.39D+004	1.13D−003
2.01	3.3E-07	129100.	2.45D+004	7.96D−005
1.87	.000007	120108.	4.49D+004	9.21D−004
1.72	.00015	110473.	9.05D+004	9.79D−003
1.58	.00279	101481.	1.84D+005	8.93D−002
1.44	.0184	92489.	4.02D+005	2.70D−001
1.29	.0653	82855.	1.01D+006	3.81D−001
1.15	.1079	73863.	2.65D+006	2.40D−001
1.01	.1161	64871.	1.06D+008	6.49D−003
.86	.0944	55237.	1.53D+010	3.64D−005
.72	.0978	46245.	3.75D+012	1.54D−007
.57	.1146	36610.	5.19D+015	1.30D−010
.43	.1402	27618.	3.20D+019	2.59D−014
.29	.1416	18626.	6.33D+024	1.32D−019
.14	.10075	8992.	3.92D+034	1.52D−029
	1.0000			1.0000

Baseline Bending Stress St = 6.42D+004 • Resultant Bending Life Nt = 5.90D+006 Cycles • Resultant Bending Life Nt = 2.72D+002 Hours

Part D — Hertizan Life — Gear

Example Wind Loads

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
2.15	2.67E-06	295254.	1.00D+004	1.19D−004
2.01	3.3E-07	285479.	1.00D+004	1.48D−005
1.87	.000007	275358.	1.00D+004	3.13D−004
1.72	.00015	264083.	1.00D+004	6.71D−003
1.58	.00279	253108.	1.37D+004	9.09D−002
1.44	.0184	241634.	3.15D+004	2.62D−001
1.29	.0653	228703.	8.40D+004	3.48D−001
1.15	.1079	215936.	2.34D+005	2.06D−001
1.01	.1161	202366.	7.47D+005	6.96D−002
.86	.0944	186735.	3.14D+006	1.35D−002
.72	.0978	170861.	1.53D+007	2.85D−003
.57	.1146	152025.	1.23D+008	4.15D−004
.43	.1402	132042.	1.53D+009	4.10D−005
.29	.1416	108437.	5.15D+010	1.23D−006
.14	.10075	75343.	3.43D+013	1.31D−009
	1.0000			1.0000

Baseline Hertzian Stress Sc = 2.01D+005 • Resultant Hertzian Life Nc = 4.47D+005 Cycles • Resultant Hertzian Life Nc = 1.02D+002 Hours

Part E — Bending Life — Gear

Example Wind Loads

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
2.15	2.67E-06	116571.	1.42D+003	1.03D−003
2.01	3.3E-07	108981.	2.49D+003	7.25D−005
1.87	.000007	101390.	4.56D+003	8.39D−004
1.72	.00015	93257.	9.20D+003	8.92D−003
1.58	.00279	85666.	1.88D+004	8.14D−002
1.44	.0184	78076.	4.09D+004	2.46D−001
1.29	.0653	69943.	1.03D+005	3.47D−001
1.15	.1079	62352.	2.70D+005	2.19D−001
1.01	.1161	54761.	8.01D+005	7.93D−002
.86	.0944	46629.	3.33D+006	1.55D−002
.72	.0978	39038.	8.15D+008	6.56D−005
.57	.1146	30905.	1.13D+012	5.56D−008
.43	.1402	23314.	6.95D+015	1.10D−011
.29	.1416	15724.	1.38D+021	5.63D−017
.14	.10075	7591.	8.51D+030	6.48D−027
	1.0000			1.0000

Baseline Bending Stess St = 5.42D+004 • Resultant Bending Life Nt = 5.47D+005 Cycles • Resultant Bending Life Nt = 1.25D+002 Hours

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Appendix 2
Example 2 Main Hoist

Part A — Input Data Summary

		Pinion	Gear
Gear Geometry Data			
Tooth Number	NP, NG =	24.	54.
Net Face Width (In.)	F1, F2 =	4.1700	4.1700
Outside Diameter (In.)	do,Do =	7.5880	15.5630
Internal Gear I.D. (In.)Di =			0.0000
Normal Diametral Pitch	Pnd =	3.6286	
Normal Pressure Angle (Deg.)	PHI(c) =	20.0000	
Standard Helix Angle (Deg.)	PSI(s) =	12.0000	
Operating Center Distance (In.)	C =	11.0236	
Gear Geometry Data For Pnd = 1.0			
Addendum Modification Coefficient	X1, X2 =	0.5000	−0.3680
Thinning For Backlash Delta (sn1), Delta (sn2)	=	0.0240	0.0240
Stock Allow. Per Side For Finishing Us1, Us2 =		0.0310	0.0310
Tool Geometry Data For Pnd = 1.0			
Tool Normal Tooth Thickness	tce1, tce2 =	1.5088	1.5088
Tool Addendum	hao1,hao2 =	1.3000	1.3000
Tool Tip Radius	rTel, rTe2 =	0.4500	0.4500
Tool Protuberance	Delta(o1), Delta(o2) =	0.0410	0.0410
Materials/Heat Treatment Data			
Modulus of Elasticity (PSI)	EP,EG =	30,000,000.	30,000,000.
Poisson's Ratio	MU(P), MU(G) =	0.3000	0.3000
Brinell Hardness	HBP, HBG =	654	654
Material (Code)	=	Steel (1)	Steel (1)
Material Grade	=	2	2
Heat-Treatment (Code)	=	Carburized (4)	Carburized (4)
Load Data			
Transmitted Power (HP)	P =	1,354.0000	
Pinion Speed (rpm)	n(P) =	1,173.0000	
Gear Blank Temperature (Deg. F)	Tb =	180.	
Reliability	R =	0.9900	
Number of Contacts per Revolution	=	1	1
Reversed Bending?	=	N	N
Derating Factors			
Application Factor For Pitting Resist.	Ca =	1.0000	
Size Factor For Pitting Resistance	Cs =	1.0000	
Surface Condition Factor	Cf =	1.0000	
Load Dist. Factor For Pitting Resist.	Cm =	1.4000	
Dynamic Factor For Pitting Resistance	Cv =	0.9160	
Runtime Options			
Option Chosen For Calculating mN	=	Accurate	
Type of Analysis Chosen	=	Miner's Rule	
Curve Chosen	=	Lower	

Part B — Hertzian Life — Pinion

Case Ident: Example 2 Main Hoist
Program AGMA218 v.1. 06B
Analysis Option: Miner's Rule

Main Hoist Loads

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
1	.0831	173902.	9.95D+008	8.87D−001
.7647	.1087	152072.	1.09D+010	1.06D−001
.5242	.2114	125908.	3.18D+011	7.06D−003
.2786	.5968	91790.	8.98D+013	7.06D−005
	1.0000			1.0000

Baseline Hertzian Stress Sc = 1.74D+005 • Resultant Hertzian Life Nc = 1.06D+010 Cycles • Resultant Hertzian Life Nc = 1.51D+005 Hours

Part C — Bending Life — Pinion

Main Hoist Loads

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
1	.0831	44495.	1.24D+013	1.00D+000
.7647	.1087	34026.	5.01D+016	3.23D−004
.5242	.2114	23325.	5.98D+021	5.26D−009
.2786	.5968	12396.	1.89D+030	4.71D−017
	1.0000			1.0000

Baseline Bending Stress St = 4.45D+004 • Resultant Bending Life Nt = 1.49D+014 Cycles • Resultant Bending Life Nt = 2.12D+009 Hours

Part D — Hertizan Life — Gear

Main Hoist Loads

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
1	.0831	173902.	9.95D+008	8.87D−001
.7647	.1087	152072.	1.09D+010	1.06D−001
.5242	.2114	125908.	3.18D+011	7.06D−003
.2786	.5968	91790.	8.98D+013	7.06D-005
	1.0000			1.0000

Baseline Hertzian Stress Sc = 1.74D+005 • Resultant Hertzian Life Nc = 1.06D+010 Cycles • Resultant Hertzian Life Nc = 3.39D+005 Hours

Part E — Bending Life — Gear

Main Hoist Loads

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
1	.0831	55431.	1.37D+010	1.00D+000
.7647	.1087	42388.	5.56D+013	3.23D−004
.5242	.2114	29057.	6.64D+018	5.26D−009
.2786	.5968	15443.	2.10D+027	4.71D−017
	1.0000			1.0000

Baseline Bending Stess St = 5.54D+004 • Resultant Bending Life Nt = 1.65D+011 Cycles • Resultant Bending Life Nt = 5.28D+006 Hours

Appendix 3
Example 3 Positioner

Part A — Input Data Summary

Gear Geometry Data		Pinion	Gear
Tooth Number	NP, NG =	22.	104.
Net Face Width (In.)	F1, F2 =	10.0000	10.0000
Outside Diameter (In.)	do,Do =	6.8541	30.2479
Internal Gear I.D. (In.)	Di =		0.0000
Normal Diametral Pitch	Pnd =	3.6286	
Normal Pressure Angle (Deg.)	PHI(c) =	20.0000	
Standard Helix Angle (Deg.)	PSI(s) =	14.9619	
Operating Center Distance (In.)	C =	18.0000	
Gear Geometry Data For Pnd = 1.0			
Addendum Modification Coefficient	X1, X2 =	0.0500	0.0546
Thinning For Backlash Delta (sn1), Delta (sn2)	=	0.0240	0.0240
Stock Allow. Per Side For Finishing Us1, Us2 =		0.0000	0.0000
Tool Geometry Data For Pnd = 1.0			
Tool Normal Tooth Thickness	tce1, tce2 =	1.5708	1.5708
Tool Addendum	hao1,hao2 =	1.3500	1.3500
Tool Tip Radius	rTe1, rTe2 =	0.3500	0.3500
Tool Protuberance	Delta(o1), Delta(o2) =	0.0000	0.0000
Materials/Heat Treatment Data			
Modulus of Elasticity (PSI)	EP,EG =	30,000,000.	30,000,000.
Poisson's Ratio	MU(P), MU(G) =	0.3000	0.3000
Brinell Hardness	HBP, HBG =	352	331
Material (Code)	=	Steel (1)	Steel (1)
Material Grade	=	1	1
Heat-Treatment (Code)	=	Thru Hard (1)	Thru Hard (1)
Load Data			
Transmitted Power (HP)	P =	960.0000	
Pinion Speed (rpm)	n(P) =	780.0000	
Gear Blank Temperature (Deg. F)	Tb =	180.	
Reliability	R =	0.9900	
Number of Contacts per Revolution	=	1	1
Reversed Bending?	=	N	N
Derating Factors			
Application Factor For Pitting Resist.	Ca =	1.0000	
Size Factor For Pitting Resistance	Cs =	1.0000	
Surface Condition Factor	Cf =	1.0000	
Load Dist. Factor For Pitting Resist.	Cm =	1.6000	
Dynamic Factor For Pitting Resistance	Cv =	0.8200	
Runtime Options			
Option Chosen For Calculating mN	=	Accurate	
Type of Analysis Chosen	=	Miner's Rule	
Curve Chosen	=	Lower	

Part B — Hertzian Life — Pinion

Case Ident: Example 3 — Positioner
Program AGMA218 v. 1.06A
Analysis Option: Miner's Rule

Car Puller 2000 Amps

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
.27	.641	68343.	4.19D+012	1.13D−005
.53	.2	95752.	1.02D+010	1.45D−003
.8	.059	117640.	2.57D+008	1.69D−002
1.07	.074	136051.	1.92D+007	2.85D−001
1.33	.026	151683.	2.75D+006	6.97D−001
	1.0000			1.0000

Baseline Hertzian Stress Sc = 1.32D+005 • Resultant Hertzian Life Nc = 7.37D+007 Cycles • Resultant Hertzian Life Nc = 1.58D+003 Hours

Part C — Bending Life — Pinion

Car Puller 2000 Amps

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
.27	.641	8489.	5.41D+027	1.23D−020
.53	.2	16664.	4.62D+018	4.49D−012
.8	.059	25153.	1.35D+013	4.56D−007
1.07	.074	33642.	1.65D+009	4.64D−003
1.33	.026	41816.	2.71D+006	9.95D−001
	1.0000			1.0000

Baseline Bending Stress St = 3.14D+004 • Resultant Bending Life Nt = 1.04D+008 Cycles • Resultant Bending Life Nt = 2.22D+003 Hours

Part D — Hertizan Life — Gear

Car Puller 2000 Amps

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
.27	.641	68343.	1.72D+012	1.13D−005
.53	.2	95752.	4.17D+009	1.45D−003
.8	.059	117640.	1.06D+008	1.69D−002
1.07	.074	136051.	7.87D+006	2.85D−001
1.33	.026	151683.	1.13D+006	6.97D−001
	1.0000			1.0000

Baseline Hertzian Stress Sc = 1.32D+005 • Resultant Hertzian Life Nc = 3.02D+007 Cycles • Resultant Hertzian Life Nc = 3.05D+003 Hours

Part E — Bending Life — Gear

Car Puller 2000 Amps

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
.27	.641	7396.	1.35D+029	8.94D−021
.53	.2	14518.	1.15D+020	3.26D−012
.8	.059	21913.	3.35D+014	3.31D−007
1.07	.074	29309.	4.12D+010	3.37D−003
1.33	.026	36431.	4.90D+007	9.97D−001
	1.0000			1.0000

Baseline Bending Stess St = 2.74D+004 • Resultant Bending Life Nt = 1.88D+009 Cycles • Resultant Bending Life Nt = 1.90D+005 Hours

Appendix 4
Example 4 Antenna Azimuth

Part A — Input Data Summary

Gear Geometry Data		Pinion	Gear
Tooth Number	NP, NG =	17.	192.
Net Face Width (In.)	F1, F2 =	4.6880	4.6880
Outside Diameter (In.)	do,Do =	6.3330	64.6660
Internal Gear I.D. (In.)	Di =		0.0000
Normal Diametral Pitch	Pnd =	3.0000	
Normal Pressure Angle (Deg.)	PHI(c) =	25.0000	
Standard Helix Angle (Deg.)	PSI(s) =	0.0000	
Operating Center Distance (In.)	C =	34.8330	
Gear Geometry Data For Pnd = 1.0			
Addendum Modification Coefficient	X1, X2 =	0.0000	0.0000
Thinning For Backlash. Delta (sn1), Delta (sn2)	=	0.0120	0.0120
Stock Allow. Per Side For Finishing Us1, Us2	=	0.0000	0.0000
Tool Geometry Data For Pnd = 1.0			
Tool Normal Tooth Thickness	tce1, tce2 =	1.5708	1.5708
Tool Addendum	hao1,hao2 =	1.3500	1.3500
Tool Tip Radius	rTel, rTe2 =	0.3500	0.3500
Tool Protuberance	Delta(o1), Delta(o2) =	0.0000	0.0000
Materials/Heat Treatment Data			
Modulus of Elasticity (PSI)	EP,EG =	30,000,000.	30,000,000.
Poisson's Ratio	MU(P), MU(G) =	0.3000	0.3000
Brinell Hardness	HBP, HBG =	341	285
Material (Code)	=	Steel (1)	Steel (1)
Material Grade	=	1	1
Heat-Treatment (Code)	=	Thru Hard (1)	Thru Hard (1)
Load Data			
Transmitted Power (HP)	P =	39.7900	
Pinion Speed (rpm)	n(P) =	56.4700	
Gear Blank Temperature (Deg. F)	Tb =	180.	
Reliability	R =	0.9900	
Number of Contacts per Revolution	=	1	2
Reversed Bending?	=	N	N
Spur Gear Loading Type	=	HPSTC (1)	
Derating Factors			
Application Factor For Pitting Resist.	Ca =	1.0000	
Size Factor For Pitting Resistance	Cs =	1.0000	
Surface Condition Factor	Cf =	1.0000	
Load Dist. Factor For Pitting Resist.	Cm =	2.0000	
Dynamic Factor For Pitting Resistance	Cv =	0.9260	
Runtime Options			
Type of Analysis Chosen	=	Miner's Rule	
Curve Chosen	=	Lower	

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Back To Basics

Achievable Carburizing Specifications

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Abstract:
A widespread weakness of gear drawings is the requirements called out for carburize heat treating operations. The use of heat treating specifications is a recommended solution to this problem. First of all, these specifications guide the designer to a proper callout. Secondly, they insure that certain metallurgical characteristics, and even to some extent processing, will be obtained to provide the required qualities in the hardened gear. A suggested structure of carburizing specifications is given.

In spite of widespread understaffing in engineering departments of gear manufacturers, gear drawings are reasonably well prepared insofar as design is concerned. However, in the very important matter of gear materials and their heat treatment, the situation is very different, especially for gears calling for case-hardening heat treatments.

The most obvious shortfall is either the quality of or the total absence of suitable heat treating specifications, the purpose of which are to facilitate obtaining the desired mechanical and metallurgical qualities in the metal. This is

Fig. 1 – Surface microstructure of a failed tooth from a 4 DP low and reverse pinion.

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understandable because few engineering departments in the USA have the budget to carry personnel knowledgeable in metallurgy. The result is the common practice by many design groups of reducing design stresses (overdesigning) so as to get by with questionable material and heat treatment engineering.

Gear designers should be aware of this practice in regard to the heat treatment of gears: It is relatively easy to produce a high quality gear when the requirements are known, as in a specification. It is nearly impossible to produce a so-called medium quality gear. When heat treating quality is reduced, it does not come down uniformly, but in a highly erratic manner. This usually results in a gear wherein some teeth may show high metallurgical quality, some borderline quality, and some very poor quality. This latter type often fails prematurely. Without suitable heat treating specifications, factors such as microstructure can go out of control undetected, resulting in an entire gear being seriously defective. (See Figs. 1 and 2.)

Here is what happened to a Fortune 500 company when design stresses were reduced to 200,000 psi in contact and 65,000 psi in bending to accomodate poor metallurgical quality. This firm was losing market share, and top management finally asked the sales department: "Why?" The answer received was: "Too many field failures." Research revealed that in a period of 25 years there were 1048 instances of major premature failure. For each failure both engineering and metallurgical investigations were made. The fault study revealed the following:

	Fault
Engineering	70.0%
Defective Material	9.6
Defective Heat Treating	15.2
Defective Manufacturing	5.2

The engineering department selected materials and specified heat treatments for which it had inadequate in-house specifications. The heat treating specification for carburizing of gears was particularly lacking, as shown below:

1) Carburize at 1650° to 1700°F
2) Cool to 1500° to 1550°F in the carburizing furnace.
3) Quench in oil

Obviously, merely having specifications was no assurance of getting a quality product.

Fig. 2 – Microstucture of a 2 DP final drive pinion which failed after 900 hours because of pitting.

Figs. 1 and 2 show microstructures of two of the company's gear failures. Fig. 1 is the surface microstructure at 500X with a 2% Nital etch of a failed tooth from a 4 DP low and reverse pinion. This failure by tooth breakage occurred after only 148 hours of operation. The reason was the lack of strength and toughness brought about by the carbide network. Fig. 2 is the microstructure at 500X of a 2 DP final drive pinion where failure by pitting occurred in approximately 900 hours. The reason for this failure was the large amount of dark etching quenching pearlite (often referred to as bainite).

The materials laboratory in this firm was used only for inspection of incoming material, technical control of heat treating, and failure analysis. This is quite typical. About 60% of the failures were carburized gears. Most of the gear failures were material and heat treatment selection errors due to incomplete specifications.

When proper heat treating specifications are available, they serve at least five important functions:

1) They insure, insofar as possible, that the important qualities counted on by the designer are provided by the heat treater.

2) They make it clear to the heat treater what is required from him.

3) They assist the designer in making the correct callout.

4) They permit heat treating changes to be made on large numbers of drawings with a minimum of effort.

5) They reduce drawing clutter.

The proposed specification format contains some of what would normally be considered material and processing standards. These might be considered out of place, however, the author believes that they should be included because 1) Details of heat treating processing can significantly affect engineering properties, including uniformity of quality in its broadest sense, and 2) Most firms do not have materials and processing standards, so a properly prepared heat treating specification can, at least in part, serve this purpose.

A complete carburizing specification should, as minimum, contain the following 15 articles:

I.	Scope
II.	Application
III.	Premachining Heat Treatment
IV.	Stress Relieving
V.	Carburizing
VI.	Hardening
VII.	Tempering
VIII.	Magnetic Particle Inspection
IX.	Cleaning
X.	Straightening
XI.	Deep Chilling
XII.	Metallurgical Requirements
XIII.	Rework
XIV.	Records & Reports
XV.	Drawing Callout

The purpose of the scope article is to give a broad description of the type of heat treatment for which it is intended; e.g., carburizing. A second function is its use in calling out certain corollary specifications, such as one for acceptable and unacceptable microstructures. Here is a suggested scope article for a carburizing specification:

I. Scope:

> This specification covers the requirements for a carburize and harden heat treatment for parts made from 9310 steel and is further qualified by AGMA-XXX (Microstructure Control).

The author believes that carburizing specifications can be written that are suitable for more than one grade of steel; e.g., 8620, 8720, and even 8822. The heat treating characteristics of 9310, however, are so different that a separate specification is preferred. Also, by combining many steels into one specification, the advantage of easily changing the requirements for one grade, shown on many drawings, is lost.

II. Application:

> This specification is intended to be used for parts such as gears and shafts made from 9310 steel. For a life of 10^8 cycles in rolling contact fatigue, a maximum design stress of 265,000 psi shall be used. A maximum bending stress for the same life of 85,000 psi is permissible. A part made per this specification provides maximum toughness. With the 9310H grade of steel applied, this heat treatment will provide a core hardness in the centerline of gear teeth at the whole depth location of 28 Rockwell C minimum. This is assuming a quench vigor of at least H = .35.

Unpredictable distortion in heat treatment causes many problems with parts such as gears. These are rework, scrap, excessive noise, and, of course, premature failure. There are two processing steps that can be taken to minimize this risk. First is a suitable premachining heat treatment. This insures that the microstructure is of maximum uniformity from one lot to the next, with accordant minimum distortion scatter. This treatment also removes stresses, from cold straightening of the raw material. Finally, it can be used to optimize machinability. Here is a suggested article:

III. Premachining Heat Treatment:

Before any machining except sawing of bars to length, all material heat treated to this specification shall have been normalized from 1740° to 1760°F and then tempered for four hours at 1140° to 1160°F. After cooling to room temperature, clean by sandblasting or a chemical means.

A second source of unpredictable distortion is the stresses developed in the material from cold working the surface in operations such as heavy turning, boring, and even rough hobbing. These stresses can be removed by a stress relieve before finish hobbing. A suggested stress relieve article is as follows:

IV. Stress Relieve:

(a) For parts requiring maximum distortion control, a stress relieve after rough machining is required. When this is the case a note will appear on the drawing as follows: STRESS RELIEVE AFTER ROUGH MACHINING.

(b) Stress relieving shall be done by heating the parts to 1000° to 1050°F and air cooling (no soak required).

(c) Cleaning, if necessary, after stress relieving shall be done by sandblasting or chemical means.

The actual carburizing operation is of major importance in the heat treatment of gears because the carbon content and its distribution in the carburized case affects these engineering qualities: strength (static and dynamic), toughness, pitting resistance, case crushing strength, wear resistance, sensitivity to grinding burn and cracks, and operating noise.

The author regrets to report that even with an operation of this importance, case carbon control has slipped in the past several years. This has been in a large part due to the widespread use of two devices: the oxygen probe atmosphere controller and direct reading spectrographs for case carbon analysis.

The problem with the oxygen probe is really threefold. First, it is a very delicate device, subject to damage and deterioration. Its readings are really in millivolts (0.001 volt). One millivolt is approximately 0.01% carbon in the carburized case on 9310 steel at 1700°F. Second, most oxygen probe auxiliaries are calibrated for a 20% carbon monoxide atmosphere (enriched endothermic gas). Often the atmosphere is changed to nitrogen and methanol without recalibration. Third, oxygen probes are not very reliable with case carbon levels below 0.80% or temperatures below 1400° F.

The problem with the spectrograph for carbon determinations is the lack of accuracy which at best is ± 0.05%. The preferred analytical procedure for carbon is combustion analysis of chips turned from a sample of the same steel as the parts being carburized.

Beyond these problems, many heat treaters have forgotten the fact that the oxygen probe reads carbon potential, but steels carburize to different levels, as shown in Table 1 for a 0.8% carbon atmosphere for 18 hours at 1700°F.[1] Because of these problems, the carburizing article in a specification must call for strong measures to insure proper case carbon control.

Insufficient case carbon content usually results in deficient case microstructure and/or low case hardness, which often results in pitting and an increased tendency to score. Excessive

Table 1

Type Steel	Case Carbon Content	
	At .002 Depth	At .007 Depth
1018	0.80%	0.75%
8115	0.80	0.74
8620	0.77	0.71
4718	0.80	0.74
4620	0.72	0.66
4820	0.67	0.63
9310	0.73	0.68

case carbon tends, first of all, to form a continuous network as shown in Fig. 1. This can make a gear tooth brittle and weaker by as much as 30%. Excessive case carbon can also result in excessive retained austenite, which adversely affects pitting life. Insufficient case depth invites case crushing, depending, of course, on the core hardness. Wear resistance increases with carbon content. A good rule to follow on case carbon is to specify no more than is necessary to achieve the required hardness. With most gear steels this content is from 0.60 to 1.00%

Following is a suggested article for the carburizing operation.

V. Carburizing:

(a) Carburizing shall be done in a furnace that is tight enough to maintain a prescribed carburizing atmosphere. The furnace shall also be equipped with au-

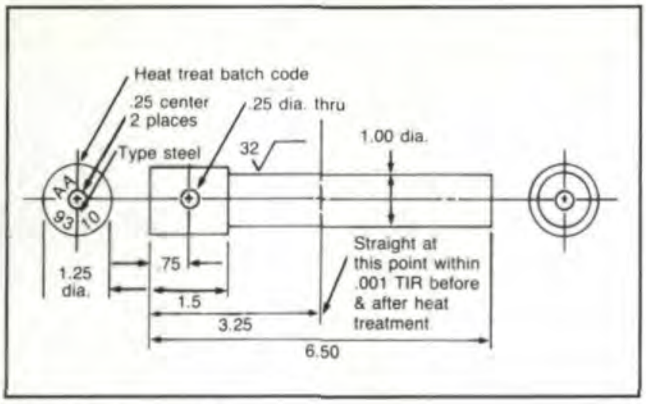

Fig. 3 – Step-turn sample.

tomatic temperature control and fans for circulating the atmosphere.

(b) The atmosphere shall consist of a mixture of endothermic and natural gasses automatically controlled by a suitable carbon potential device. When AGMA-XXX Grade A is called out on the drawing, there shall be at least one backup arrangement to insure that the desired carbon content is obtained. For example, an oxygen probe plus a dew point check, plus carbon steel progress specimens to be examined microscopically, and a step-turn sample as shown in Fig. 3.

(c) At least one step-turn sample as shown in Fig. 3 shall be charged with each furnace load, and conformance determined by combustion tests on chips turned from such a sample that has been both carburized and hardened with the parts.

(d) The hardened sample shall be tempered in a neutral material such as lead, bismuth, argon, or vacuum for two hours at 1200° to 1250°F to provide for the proper machinability to make the required chips.

(e) After the sample has been checked for straightness, the first cut shall be 0.0025" deep on a side. Additional cuts shall then be taken 0.005" deep on a side, until at least the minimum case depth specified has been reached. Chips from each cut shall be kept separate in properly marked envelopes.

(f) A carburizing medium prepared from nitrogen and methanol may be used so long as the oxygen probe control is calibrated for its use.

(g) The carburizing temperature shall be 1700°, ± 20°F unless otherwise specified on the part drawing. For case depths over 0.030 inch the carburize diffuse procedure is preferred. Total penetration is $0.025\sqrt{T}$ where T is the time in hours at 1700°F.[2]

(h) The maximum case carbon shall be at the surface of the parts and the sample, and shall be from 0.75% to 0.85%. For AGMA-XXX Grade B gears, spectrographic carbon results from the surface of a suitable sample of 9310 steel are acceptable.

(i) A mutually agreeable sampling plan shall be worked out for parts run in a continuous carburizer.

(j) The duration of the carburizing cycle shall be such that the specified case depth is retained on the parts after finish grinding, leaving at least 0.70% minimum carbon on the surface.

(k) The minimum case depth, unless experience has indicated otherwise, shall be determined via case crushing calculation per AGMA-218 Section 14.

(l) The tolerances on case depth for 9310 steel are shown in Table 2.

Table 2

Minimum Case Depth (Inch)	Plus or Minus Tolerance (Inch)
Up to 0.030	0.005
Over 0.030-050	0.0075
Over 0.050-0.070	0.010
Over 0.070-0.100	0.015
Over 0.100	0.020

(m) At the conclusion of the carburizing cycle, the parts shall be cooled to black in a protective environment.

After the carburizing operation, the next step in heat treating is the hardening. It will be noted in the following article (VI) that the author calls for a carburize-reheat harden type of treatment and has a preference for it over direct quenching. This type of treatment give maximum assurance for freedom from micro-cracks with attendant loss in bending fatigue qualities, as well as a reduction in the amount of retained austenite, and as a result, higher case hardness and resistance to pitting. The lower amount of retained austenite results in the best size stability in final manufacturing operations and field usage. These are both serious problems when direct quenching 9310 steel. Also, some manufacturers have found that the cost of a carburize-reheat heat treatment is no more than direct quenching. However, suitable furnace equipment must be available.

VI. Hardening:

(a) Parts shall be heated to a temperature of 1520° to 1540°F and then oil quenched.

(b) Reheating shall be done in an environment such that the surface carbon content of the parts is maintained within that specified for carburizing.

(c) The carburizing sample shown in Fig. 3 shall accompany the gears through both the carburizing and hardening cycles for AGMA-XXX Grade A and combustion analysis used. For AGMA-XXX Grade B the sample shall similarly pass through both the carburize and harden operations; however, spectrographic analysis may be used.

(d) Direct quenching from the carburizing furnace is not permitted.

(e) The preferred quenching oil should have a viscosity of 80 to 120 SUS at 100°F and be maintained at a temperature of 90° to 120°F.

After quenching it is customary to wash and draw carburized gears. The wash operation is usually done with a hot alkaline or solvent emulsion solution to remove the residual quench oil and some of the other debris from the heat treating operations. The draw is thought to reduce some undesirable stresses and transform some of the retained austenite to improve grinding qualities. The author is not aware of any work to substantiate this thinking, but it is known that a low temperature draw, e.g., 350°F, reduces the residual compressive stresses in a carburized case by several thousand psi. However, it is probably well to include a temper operation in a carburizing specification as follows:

VII. Tempering:

Unless otherwise specified on the part drawing, wash free of quench oil and other heat treating debris and temper for two hours at 325° to 350°F and air cool. Note: If magnetic particle or dye penetrant inspection is required for the part, it shall be done immediately after this operation or after finish grinding.

Large non-metallic inclusions on the flanks (faces) of carburized gear teeth can lead to premature failure. The most common mode of failure is one or more teeth breaking out at the root fillet. This failure occurred in only 725 hours; it was caused by a large alumina inclusion in the surface of the root fillet.

Inclusions in the tooth flanks can also be sites for pitting failures to commence.

To avoid having to make a drawing callout, an article as follows is suggested for magnetic particle inspection:

VIII. Magnetic Particle Inspection:

AGMA-XXX Grade A gears shall be 100% inspected using a wet flourescent process as set forth in AGMA-XXX. Grade B gears shall be similarly inspected, but on a formal sampling plan.

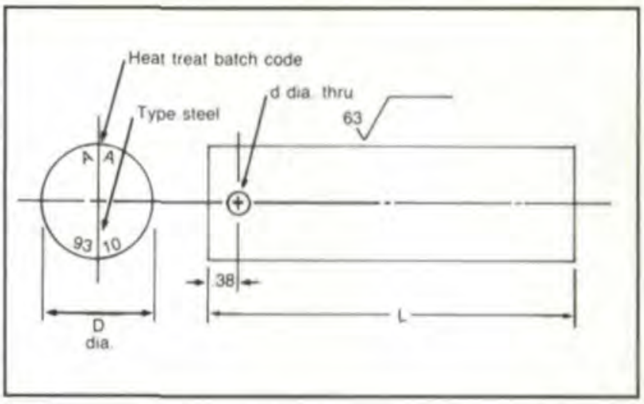

Fig. 4

After washing and tempering, many carburized gears still do not have an acceptable appearance, so it is customary to blast clean them with sand, shot, or other abrasive material. This operation also tends to finish the deburring operation. If soft shot of 45 to 50 Rockwell C hardness is used, the blasting will slightly reduce the residual compressive stresses in the carburized case. Here is a suggested cleaning article:

IX. Cleaning:

After cooling to room temperature, clean the parts by shot or sand blasting. Shot size shall be S-330 maximum. Grit blasting is not permitted.

At some time after hardening, especially if a quench press is not available, it is sometimes necessary to straighten carburized parts. This operation can be damaging to the parts' usefulness for the following reasons: 1) The hard case might be cracked, which could lead to premature failure from this defect; 2) The desirable residual compressive stresses in yielded areas of the part are eliminated; hence, long life bending and/or torsional fatigue qualities are reduced; 3) The part can be in an unstable condition, likely to return at least partially, to its unstraightened shape when put in service.

Because straightening is such a potentially damaging and expensive operation, everything practical should be done to eliminate the need for it. When it is still necessary, a reasonably satisfactory solution to the problem is to call for all straightening to be done hot, followed by 100% magnetic particle inspection for cracks. A suggested straightening article is as follows:

X. Straightening:

Parts heat treated to this specification shall only be straightened hot; i.e., at 325° to 350°F followed by air cooling to room temperature. All parts shall then be magnetic particle or dye penetrant inspected for cracks.

A practice that is usually the result of loss of control of the carburizing process with excessive case carbon is the necessity of deep chilling to obtain the specified case hardness. The reason for this is the retention of excessive amounts of austenite. Deep chilling transforms a large portion of this austenite into martensite. However, as reported by deBarbadillo, et al.,[3] this results in about a 25% loss in bending fatigue strength. So the following is recommended:

XI. Deep Chilling:

Unless permitted on the part drawing, deep chilling of parts heat treated to this specification is not permitted.

In a carburized part there are a number of metallurgical characteristics that provide evidence of proper heat treating. These should be part of a carburizing specification as shown below:

XII. Metallurgical Requirements:

(a) The surface hardness of parts after proper surface preparation shall be 59 to 63 Rockwell C measured at the test location shown on the drawing. Note: When the specified case depth is less than 0.030 inch, the surface hardness shall be 90 to 92 Rockwell 15-N.

(b) The tips and flanks of gear teeth shall be file hard to a medium mill bastard file (Nicholson or equal).

(c) The case depth shall be determined for each heat treating lot, and on gears is that distance measured normal to the surface at the LPSTC inward to where the equivalent of 50 Rockwell C occurs.

(d) If it is impractical to cut a gear, samples machined per Fig. 5 from 9310 steel and run with the parts may be used to measure case depth and evaluate microstructures. This work shall be done on a 0.25 inch thick transverse slice from the center of the specimen as shown in Fig. 4. The dimensions of the test specimens are shown in Table 3.

Table 3

(In inches)

Gear Pitch	Diameter (D)	Diameter (d)	Length (L)
1 and Coarser	3.00	0.25	6.00
Finer than 1 to 3	1.50	0.25	5.00
Finer than 3 to 8	1.00	0.25	4.00
Finer than 8	0.50	0.13	2.00

(e) The microstructure at the surface of a gear shall be examined at 400X to 500X at the LPSTC location midway between the ends of the teeth looking for microcracks, network carbide, and quenching pearlite (often referred to as bainite). No micro-cracks or subsurface quenching pearlite are permitted. Also carbide network is not permitted. If a gear cannot be cut, the specimen from Paragraph XII. D can be used for microstructure evaluation.

(f) The core microstructure shall indicate that the part had been properly austenitized for hardening with no blocky ferrite visible at 400X to 500X.

(g) The etchant for microscopic examination shall be 2 to 3% Nital. The etching time to detect micro-cracks and quenching pearlite is very short, usually only 2 to 4 seconds. In order to bring out network carbide and blocky ferrite in the core, the time will usually be from 5 to 7 seconds.

One of the most serious situations that can develop, which adversely affects the quality of carburized gears, takes place when parts do not meet drawing requirements and it is decided that they are salvageable by re-heat treating (rework). This is a potentially serious problem because a number of things can go wrong. For example: 1) Every time a gear is heated, it becomes more distorted; 2) If a hardened gear is charged into a hot furnace, it might crack; 3) If the carburized case depth is shallow, carburizing a second time just about doubles the case carbon content, because of the "super carburizing" effect. This can result of excessive retained austenite or a carbide network as shown in Fig. 1.

Accordingly, the following article is recommended:

XIII. Rework:

(a) All heat treating rework shall be approved by the design control.

(b) A written procedure shall be prepared for all rework.

(c) All reworked parts shall be suitably marked so that retrieval, if necessary, is possible.

The heat treater of high quality carburized gears should be in a position to verify, by examination of a part or samples, that specification requirements have been met on each batch processed. This should include tests for case carbon content, case depth, and case and core hardness. Test results should be suitably recorded and reports made as suggested below:

XIV. Records and Reports:

The heat treater shall perform, or have performed, tests to show compliance with this specification. He shall maintain records of these test results traceable to part number, order number, and heat treat batch code.

For AGMA-XXX Grade A gears, for each batch code, the heat treater shall provide the purchaser of the heat treating service, or the design control of the parts with a report of tests run including photomicrographs at 400X of the case surface and the core.

No matter how well heat treating specifications are prepared, if they are not properly called out on the drawing, confusion, as a minimum, is the result. Sometimes rework, scrap, and even premature field failures occur for this reason. Accordingly, it is suggested that a drawing callout article as shown below be included in a carburizing specification:

XV. Drawing Callout (For design use only):

(a) Heat Treatment: AGMA-XXX Grade A
Case Depth: .XXX-.XXX

or (b) Heat Treatment: AGMA-XXX Grade B
Case Depth: .XXX-.XXX

or (c) Heat Treatment: AGMA-XXX Grade A
Case Depth: .XXX-.XXX

STRESS RELIEVE AFTER ROUGH MACHINING

There are two additional factors that are important in obtaining heat treating of the prescribed quality. They are 1) The heat treater must have suitable basic equipment and systems in place for both production and quality control, along with personnel dedicated to doing the specified work. A procurement policy for heat treating that favors only price usually results in much job movement, and is discouraging to suitable capital investment in new facilities; and 2) There must be a harmonious relationship between the organization that designs the parts, the one that machines them, and the one that does the heat treating.

To properly carburize irregular parts such as gears, the carburizing gas must be vigorously circulated with hot fans. Trays, baskets, and fixtures must be available to hold and position parts so that a uniform flow of carburizing gas can take place in and around parts. Proper fixturing also minimizes distortion due to sagging at temperature. Gear teeth must not touch each other, nor should they touch a basket or fixture. A high percentage of quench installations lack vigor and/or uniformity and many loads that are quenched are too massive and tightly packed. It is suggested that heat treaters test their quenches for H value as set forth on page 43 of the March, 1985, issue of *Heat Treating* magazine. An H value of 0.50 indicates a well agitated oil quench.

A gear heat treater must be properly equipped with well maintained quality control equipment. This includes not only that for temperature and atmosphere composition but also for case carbon content, case depth, hardness, and microstructure. He should be in a position to submit a report of his tests showing compliance with the specification requirements.

The first step in making a good gear is having a complete and accurate drawing in terms not only of dimensions, tolerances, and finishes, but metallurgical qualities as well. Suitable heat treating specifications play an important role in making drawings complete. To prevent a major duplication of effort and a flood of new specifications to heat treaters, it is suggested that AGMA consider publishing heat treating specifications.

References

1. *Carburized Nickel Alloy Steels*. International Nickel Company. New York, New York, 1966, p. 8.
2. "Case Hardening of Steel", *Metals Handbook*, Vol. 4, 1981, p. 342, American Society for Metals, Metals Park, Ohio 44073.
3. J.J. DEBARBADILLO, "The Effect of Impact Prestressing on the High Cycle Fatigue Resistance of Carburized Gear Steels," SAE International Automotive Engineering Congress, Detroit, Michigan, January 8-12, 1973.

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This article also appeared in the March & April, 1989, issues of HEAT TREATING *magazine.*

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VIEWPOINT

I was excited to see Mr. Lefkowitz's photo of the Cone Drive double-enveloping gearset on the cover of your November/December 1989 issue. Mr. Lefkowitz borrowed that gearset from me almost three years ago, and I never expected to see the photo show up in publication. It looks great!

Sincerely,

Duane Gilbert
Marketing Supervisor
Cone Drive Operations, Inc.

(continued from page 35)

Part B – Hertzian Life – Pinion

Case Ident: Example 4 – Antenna Azimuth
Program *AGMA218* v. *1.06A*
Analysis Option: Miner's Rule

Load Ratio	Cycle Ratio	Antenna AZ Box Hertzian Stress	Cycles To Failure	Damage Ratio
.0001	0	2220.	9.98D+038	0.00D+000
.4282	.0199	145274.	3.75D+006	1.74D−002
.4593	.0199	150457.	2.00D+006	3.25D−002
.4904	.0398	155467.	1.12D+006	1.17D−001
.5502	.0796	164673.	4.00D+005	6.52D−001
.1459	.2389	84799.	5.61D+010	1.39D−005
.0861	.0002	65143.	6.22D+012	1.05D−010
.4498	.0796	148893.	2.42D+006	1.08D−001
.3947	.0398	139475.	7.76D+006	1.68D−002
.3684	.0199	134748.	1.44D+007	4.54D−003
.3421	.0199	129849.	2.78D+007	2.34D−003
.4282	.0158	145274.	3.75D+006	1.38D−002
.4593	.0158	150457.	2.00D+006	2.58D−002
.0861	.3793	65143.	6.22D+012	2.00D−007
.3947	.0158	139475.	7.76D+006	6.67D−003
.3681	.0158	134693.	1.45D+007	3.58D−003
	1.0000			1.0000

Baseline Hertzian Stress Sc = 2.22D+005 • Resultant Hertzian Life Nc = 3.27D+006 Cycles • Resultant Hertzian Life Nc = 9.66D+002 Hours

Part C – Bending Life – Pinion

Load Ratio	Cycle Ratio	Antenna AZ Box Bending Stress	Cycles To Failure	Damage Ratio
.0001	0	5.	3.60D+126	0.00D+000
.4282	.0199	22964.	1.32D+014	1.05D−004
.4593	.0199	24632.	1.51D+013	9.17D−004
.4904	.0398	26299.	1.98D+012	1.39D−002
.5502	.0796	29506.	5.63D+010	9.82D−001
.1459	.2389	7824.	3.96D+028	4.19D−018
.0861	.0002	4617.	4.89D+035	2.84D−028
.4498	.0796	24122.	2.88D+013	1.92D−003
.3947	.0398	21167.	1.65D+015	1.68D−005
.3684	.0199	19757.	1.39D+016	9.93D−007
.3421	.0199	18346.	1.38D+017	1.00D−007
.4282	.0158	22964.	1.32D+014	8.30D−005
.4593	.0158	24632.	1.51D+013	7.28D−004
.0861	.3793	4617.	4.89D+035	5.39D−025
.3947	.0158	21167.	1.65D+015	6.67D−006
.3681	.0158	19741.	1.43D+016	7.69D−007
	1.0000			1.0000

Baseline Bending Stress St = 5.36D+004 • Resultant Bending Life Nt = 6.95D+011 Cycles • Resultant Bending Life Nt = 2.05D+008 Hours

(continued on page 48)

APPLICATION OF MINER'S . . .

(continued from page 47)

Part D – Hertizan Life – Gear

Antenna AZ Gear

Load Ratio	Cycle Ratio	Hertzian Stress	Cycles To Failure	Damage Ratio
.0001	.7083	2220.	7.78D+037	7.97D−033
.4282	.0058	145274.	2.92D+005	1.74D−002
.4593	.0058	150457.	1.56D+005	3.25D−002
.4904	.0116	155467.	8.70D+004	1.17D−001
.5502	.0232	164673.	3.11D+004	6.52D−001
.1459	.0697	84799.	4.37D+009	1.40D−005
.0861	.0002	65143.	4.85D+011	3.61D−010
.4498	.0232	148893.	1.88D+005	1.08D−001
.3947	.0116	139475.	6.04D+005	1.68D−002
.3684	.0058	134748.	1.12D+006	4.54D−003
.3421	.0058	129849.	2.17D+006	2.34D−003
.4282	.0046	145274.	2.92D+005	1.38D−002
.4593	.0046	150457.	1.56D+005	2.58D−002
.0861	.1106	65143.	4.85D+011	2.00D−007
.3947	.0046	139475.	6.04D+005	6.66D−003
.3681	.0046	134693.	1.13D+006	3.57D−003
	1.0000			1.0000

Baseline Hertzian Stress Sc = 2.22D+005 • Resultant Hertzian Life Nc = 8.75D+005 Cycles • Resultant Hertzian Life Nc = 1.46D+003 Hours

Part E – Bending Life – Gear

Antenna AZ Gear

Load Ratio	Cycle Ratio	Bending Stress	Cycles To Failure	Damage Ratio
.0001	.7083	4.	1.59D+129	4.69D−115
.4282	.0058	16916.	5.84D+016	1.05D−004
.4593	.0058	18145.	6.67D+015	9.17D−004
.4904	.0116	19374.	8.77D+014	1.39D−002
.5502	.0232	21736.	2.49D+013	9.82D−001
.1459	.0697	5764.	1.75D+031	4.20D−018
.0861	.0002	3401.	2.16D+038	9.75D−028
.4498	.0232	17770.	1.27D+016	1.92D−003
.3947	.0116	15593.	7.28D+017	1.68D−005
.3684	.0058	14554.	6.15D+018	9.93D−007
.3421	.0058	13515.	6.09D+019	1.00D−007
.4282	.0046	16916.	5.84D+016	8.29D−005
.4593	.0046	18145.	6.67D+015	7.27D−004
.0861	.1106	3401.	2.16D+038	5.39D−025
.3947	.0046	15593.	7.28D+017	6.66D−006
.3681	.0046	14542.	6.31D+018	7.68D−007
	1.0000			1.0000

Baseline Bending Stess St = 3.95D+004 • Resultant Bending Life Nt = 1.05D+015 Cycles • Resultant Bending Life Nt = 1.76D+012 Hours

References

1. FUCHS, H.O., and STEPHENS, R.I. *Metal Fatigue in Engineering*, J. Wiley, 1980.
2. PALMGREN, A. "Durability of Ball Bearings", *ZDVDI*, Vol. 68, No. 14, 1924, p.339, (in German)
3. MINER, M.A. "Cumulative Damage in Fatigue", *Journal of Applied Mechanics*, Vol. 12, 1945, pp. A159-164.
4. HAPEMAN, M. J., "General Electric Motorized Wheel", 1971, AGMA P109.24, American Gear Manufacturers Association, Alexandria, VA
5. NELSON, D., "Cumulative Fatigue Damage in Metals", Stanford University, Ph.D., 1978, University Microfilms International, Ann Arbor, MI.

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