GEAR TECHNOLOGY

The Journal of Gear Manufacturing

IMTS '96 SHOW ISSUE—

SEPTEMBER/OCTOBER 1996

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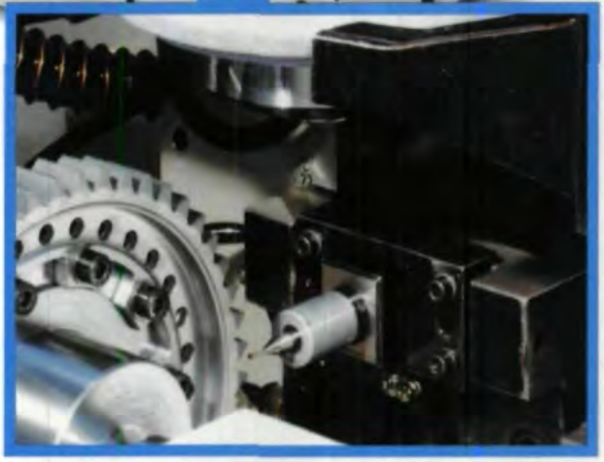

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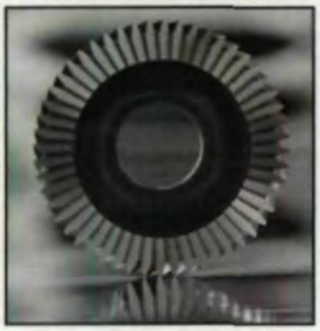

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GEAR TECHNOLOGY

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The Politics of Denial

A good many things bother me about election years—the annoying sound bites, the negative commercials, the endless political over-analysis. But what bothers me most about the coming election is this: So far (when I'm writing this, it's admittedly early in the campaign) there's little or no talk about what is one of the most critical national issues of the next thirty years—our growing government debt.

With yearly deficits, sometimes smaller, sometimes larger, this debt is like the creature in the movie "Alien." It keeps growing bigger and bigger inside our country, sucking the insides out of our treasury, and one day, I fear, it will burst forth and bring catastrophe to our society.

But no one running for national office seems to want to say that—not in an election year. Our debt and its accompanying burden of interest payments, caused in large part by our "entitlements" programs, remains a political "third rail." Politicians touch them at their peril. Talk about a state of denial.

Deficit economics is a complicated business, but certain conclusions are clear. In the July 1 issue of *Forbes*, a little chart illustrates the inevitable course to which we are already committed. *Right now* only 18% of the federal budget is "discretionary spending." Everything else is taken up with entitlements and interest. By 2030, the numbers will read this way: 93.6% of the TOTAL federal budget will go to entitlements and interest and 6.4% for discretionary income (everything else).

The *Forbes* chart shows some other disturbing numbers. In 1995 government spending accounted for 21.7% of our Gross Domestic Product. Thanks to a budget our leaders have promised to "balance" by 2002, it will be down to 18.5%. So far so good, but look a little farther into the future. By 2010 it will be back up to 24.6% of the GDP, and by 2030, the total federal government take will be 37.3%. And these numbers don't include the additional 11% that states will take, *assuming their spending remains unchanged for the next 35 years*. In 2030, the federal deficit alone will be 18.5% of the GDP, (up from only 2% now), and of this 10.7% will be interest.

Some optimistic scenarios suggest that as baby boomers age and enter their peak saving years, these numbers will begin to magically fix themselves. But that's a false promise. According to the July 15 issue of *Business Week*, while these savings increases will add $27.5 billion to annual personal savings, rising demand for health care among the aging will soak up the gains and then some. Medicare and Medicaid will cost more and more, pushing the deficit from $164 billion in 1995 to $376 billion in 2005. For every extra dime saved, the government will have to borrow 76 cents to meet its obligations.

The main culprit in these scenarios is that mass of "entitlements," (mostly, but not entirely, Social Security, Medicare and Medicaid) which we are already committed to paying out. We've created a giant Ponzi scheme where we're using the current payments (almost 15% of everyone's total earnings), which taxpayers believe are going to their retirements, to meet our obligations to present retirees. The present Social Security "surplus" is "loaned" to the Treasury to reduce each year's deficit. The Social Security system receives IOUs for this money, and it's left to the next generation and the vagaries of a future economy to pay back these loans.

Is it reasonable that people, regardless of their economic need, get back many many times what they put into the Social Security system and far more than they would get from the same money in any other investment? We've created a myth where people are "entitled" to this

"Everybody, sooner or later, sits down to a banquet of consequences."

Robert Lewis Stevenson

money, and now we're living with the consequences of it.

But soon we'll have no option but to face reality. Unchanged, our present course could lead to disaster within the next generation or two. In 1960 there were 5.1 taxpaying workers to support every retiree. By 2040 there will be 2, or possibly less.

This is a recipe for guaranteed generational warfare. At some point, those two future taxpayers, burdened with responsibilities for their own families and whatever other taxes they will have to pay, will say, "Enough! I won't take it any more! I'm not paying for these old geezers to sit around and play shuffleboard in Florida while I work three jobs to keep a roof over my head." At the same time, the "old geezers," understandably enough, will fight to keep their "entitlements" coming. It's not rocket science to figure out what that will mean for any sense of national unity.

But the scenario gets even worse. We already have very little left over after meeting current obligations to do the other things we expect our government to do, such as provide for defense, maintain our position of leadership in the world, care for our physical infrastructure and educate our children and grandchildren. And as time goes on, we'll have less and less.

It's a cliché to say that election year is the time to send a message to Washington. Sometimes I wonder what the point is. Messages were sent in 1992 and 1994, and it didn't seem to make any difference. Our elected leaders are still "fixing" the problem by putting Band-Aids on our fiscal internal bleeding.

At the time I'm writing this, we're again confronted with the insanity of both presidential candidates promising tax cuts within twenty-four hours of the announcement that part of Social Security will be bankrupt in five years.

What's wrong with this picture?

Ultimately democracies tend to get the governments they deserve. If we have a government of men and women who won't confront the hard questions of the coming years, maybe it's because we encourage them not to. It's frustrating when the candidates try to buy the election with minor tax cuts, cheap temporary fixes and denial, mixed with polarizing statements and demonization of the opposition. Compromise seems to have become a crime, and decency and civility in public discourse have become aberrations. Purity has become more important than progress. On the other hand, if we don't demand something else, then that's what we'll get from Washington.

The message we have to send—and mean—is this: Acting responsibly to control the deficit and deal with "entitlements" (not just the other guy's, but all of ours) is not a political "third rail"; failing to do so is. The message should be, "If you can't tell the truth and put the long-term good of the country ahead of the short-term interests of yourself, your party and the influential special interest groups, then we'll find someone who will."

While "government" is often part of the problem, only we, through our elected officials, can balance its inflow and outflow of funds. We must act in a fiscally responsible manner in our personal lives and in our businesses. Shouldn't we insist that our representatives be just as responsible when the question is government spending?

Fixing the deficit/entitlements mess won't be easy, but it's not impossible. It will mean, however, accepting some short-term pain for the sake of long-term gain; sacrificing some personal benefits for the sake of the general good; doing without present pleasures for the sake of our own and other people's children.

Our parents and grandparents and their parents for generations did it. It never occurred to them to do anything else. We can and must do it too. A country burdened with continuing deficits, low savings rates and an aging population will not remain economically healthy or politically strong.

We owe it to ourselves, our past and our future to get the debt under control: to ourselves because most of us will still be alive when the system starts to unravel; to our past because it was the self-denial and the dreams of our ancestors for us that have brought us the greatness and prosperity we're squandering away; and to our children because every day that we delay means more of the bill falls to them to pay. And that is neither right, nor fair, especially when they never agreed to accept the obligation in the first place, and when it's still within our power to relieve them of much of the burden.

I, for one, don't have all the answers. The exact means we use to solve the debt problem is an issue about which reasonable people may disagree. But we can't put off the question any longer.

There's no better time than an election year to remind ourselves and our elected officials that the real issue confronting us is not just "character" or "values," it's providing the leadership to deal with our crippling debt and enable us to become the kind of country we can and want to be in the next century.

Michael Goldstein

Michael Goldstein,
Publisher & Editor-in-Chief

IMTS Traveling Companion

The gear lover's guide to "da show," plus native dishes, language lessons, amazing factoids and other bits about our kind of town.

Da Staff

CHICAGO'S *an appropriate place for the largest machine tool show in North America. After all, the civic mantra here for nearly a hundred years has been to "make no small plans." But bigness has its downside. When you're in town for only a few days and have a lot of ground to cover, you have to zero in on the essentials.*

That's where the intrepid Gear Technology *staff comes in. We've spent months, weeks—well, half a day, anyway—picking out the good stuff. What follows is our exclusive gear lover's guide to McCormick Place and beyond.*

IMTS SHOW BASICS

Where: McCormick Place, Chicago.

When: Sept. 4–11, 9 am until 6 pm daily (except Sunday, Sept. 8, from 10 am until 4 pm)

Who: 1,300 exhibitors occupying 1.1 million sq. ft. of exhibit space and an expected 100,000+ attenders.

Registration: (800) 322-IMTS.

MANUFACTURING '96

Sponsored by SME, *Manufacturing '96* is a conference on various manufacturing topics. Gear industry attenders won't want to miss the following clinics:

- Fundamentals of Gear Manufacturing, Sept. 9.
- Coordinate Measurement Systems Fundamentals, Sept. 4.
- Failure Analysis of Tool Steel Components, Sept. 4.
- Fundamentals of Coatings for Cutting Tools, Sept. 6.

Many other clinics and tutorials are available. For prices and registration information, call SME Customer Service Center, (800) 733-4763. Registration for *Manufacturing '96* includes admission to IMTS 96.

IMTS AMAZING NUMBER FACTOIDS

9 MILES OF CARPETING TO COVER THE AISLES.

32 *football fields of exhibit space.*

35 MILLION POUNDS OF EQUIPMENT AND MATERIALS ON LOCATION.

$90 million will be spent by IMTS visitors on meals, rooms, transportation and services.

436,000 hot dogs sold per year at McCormick Place.

ENGLISH–HOW SHE ARE SPOKE

This is a polyglot town. You can hear almost any language spoken on the planet (and a few from elsewhere) in *some* Chicago neighborhood. But we also have our own take on English. Here are a few important words and phrases to make you bilingual here.

- Please, the name is "Chi-caw-go." Saying "Chi-ca-go" or, worse, "She-ca-ga" is like stamping "out-of-towner" on your forehead.
- You may drink "soda" elsewhere, but here we drink "pop."
- Chicago kids don't wear "sneakers." They wear "gym shoes."
- Out-of-towners, the upscale folks in Lincoln Park and the intellectuals at the U. of C. all say, "he said." The rest of us Grabowskis say, "he goes."
- When a native talks to you about "a hunnert" of anything, he means "a hundred."
- "Da Mare" or "Hizzoner" is Richard M. Daley, the mayor. In other towns, mayors and their friends and associates have "pull." Here they have "clout." Clout may be the most important word in Chicago-ese, with the possible exception of "Michael Jordan."

DA EXHIBITS

Below is a list of gear-related booths you won't want to miss. *Gear Technology* advertisers are shown in bold.

Your first stop should be the Gear Generating Pavilion, a space in the North Building, Upper Level, devoted exclusively to gear generating equipment.

Gear Pavilion—North Building, Upper Level

B1-6967 American Pfauter/Pfauter-Maag Cutting Tools. Dry hobbing machines, form grinders, gear measuring machines, gear shaving machines, hobs, shaper cutters, form cutters, rack cutters, thin film coatings.

B1-6991 Vermont USA Machine Tool Group. Fellows Corp. (gear shapers), Bryant Grinder (grinding machines), J&L Metrology (optical inspection) and Jones & Lamson (lathes).

B1-7145 Mitts & Merrill L.P. Keyseating machines.

B1-7148 Reishauer Corp. Gear grinding machines, dressing attachments.

B1-7157 Koepfer America. Gear hobbing machinery.

B1-7158 Star Cutter Co. Tool & cutter grinders, hobs, shaper cutters, gun drills, pressure coolant reamers, milling cutters, thin film coatings, plated products, grinding wheels & dressers, carbide pre-form blanks.

The new 5-axis CNC universal tool and cutter grinder will be demonstrated by Star Cutter Co.

B1-7170 The Gleason Works. Dry hobbing machines, hypoid cutting machines, large bevel gear cutting machines, shaving machines, honing machines, helical gear grinders, tooling, workholding and a PC-based knowledge system.

The Gleason Works will demonstrate dry hobbing and its new PC-based knowledge system on its PHOENIX 125GH hobbing machine.

B1-7176 Liebherr America/Sigma Pool. Dry hobbing machines, gear shaping machines, gear inspection & testing equipment.

Other important exhibits of interest to gear lovers are found throughout McCormick Place.

North Building—Upper Level

B2-6306 Kokusai Inc. Honing machines.

B1-6477 Inductoheat, Inc. Induction heat treating systems.

B2-6535 Balzers Tool Coating, Inc. TiN, TiAlN, TiCN and other thin-film tool coatings.

B1-6935 JRM International. Deburring machines, grinding machines, honing machines, milling machines and part marking systems.

B2-7060 Schunk Inc. Chucks, arbors, clamping blocks, vacuum components, expansion units, grippers and accessories.

B1-7114 American Wera. Chamfering, gear generating, shaping and turning machines.

B1-7125 Edgetek Machine Corp. High efficiency deep grinding machines.

B2-7200 LMT-Fette, Inc. Solid carbide gash hobs, heavy duty roughing hobs and hobs with indexable carbide inserts.

B1-7335 Sunnen Products Co. Honing machines.

B1-7489 National Broach & Machine Co. Broaching machines, grinding machines, hobbing machines, honing machines, inspection equipment, shaping machines, shaving machines, spline rolling machines and cutting tools.

North Building—Lower Level

C1-5007 Charmilles Technologies Corporation. EDM machines & tooling.

C1-5407 Agie USA Ltd. EDM machines and tooling.

East Building—Upper Level

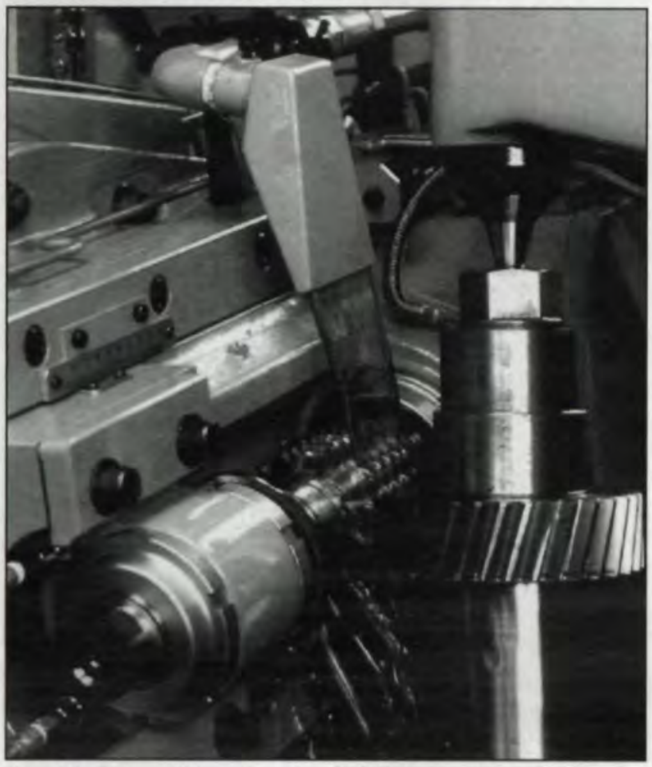
D2-4254 Mitsubishi Machine Tool. Hobbing machines, hobbing/finish rolling machines, gear shaving machines, high speed drilling machines.

D2-4358 WMW Machinery Company. TOS Celakovice Gear Hobbing machines, Mikromat worm and thread grinders and other machine tools.

The TOS Celakovice OFA 32 R hobbing machine will be shown by WMW Machinery Company.

D2-4359 Carl Zeiss, Inc., Industrial Measuring Technology Division. High precision inspection and measurement equipment.

D1-4385 Ikegai America Corporation. Gear hobbing machines and hob sharpening machines.



Ikegai America will introduce the SX-15 high-speed gear hobber.

D1-4619 Brown & Sharpe Mfg. Co., Custom Metrology Division. High precision inspection and measurement equipment.

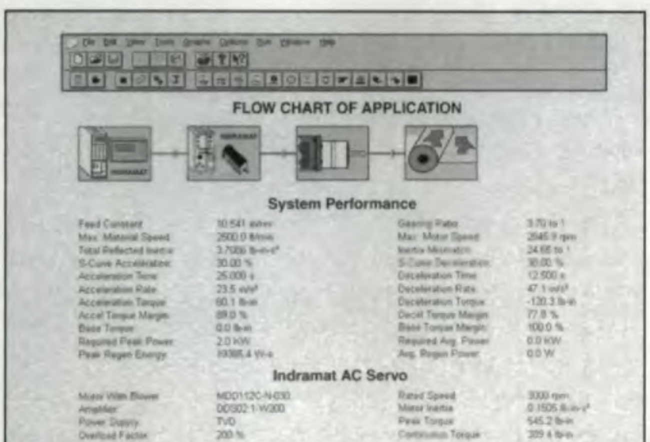
D1-4633 Giddings & Lewis, Sheffield Measurement Div. High precision inspection and measurement equipment.

D1-4700 Deckel-Maho-Gildemeister America. Universal machining centers, lathes, turning centers.

East Building—Middle Level

E1-2429 CIMNET/JNL Industries. Shop management & SPC software.

E1-2561 Indramat Division, Rexroth Corporation. Motion control systems.



Indramat introduces a Windows-based program for analyzing machine requirements and selecting motion control components for best performance.

E1-2640 Ingersoll Cutting Tool Company. Milling cutters and other cutting tools.

LOUIE "DA LIZZARD'S" GUIDE TO FINE DINING

Chicago's been about doing business since Jean Baptiste DuSable dealt in beaver pelts from his outpost at the mouth of the Chicago River. It's been about feeding the visitors since he diversified by adding a smokehouse and a vegetable garden to his holdings. Now Chicago has nearly 10,000 restaurants, and if you can't find something you like to eat, it's because you're not trying very hard.

As part of our mission to cut to the IMTS chase, we've sent our resident insider, gourmand, barfly and expert on all things Chicago, Louie "Da Lizzard," to report on the best places to eat, so you can devote your attention to the important things, like buying and selling gear equipment.

Like everything else about Louie, his list is unique. Most of the places here are not "hot." With few exceptions, you won't see movie stars or even aldermen with diamond pinky rings. These restaurants are about food, not glitz. The only criterion for inclusion is that the eatery has appeared on Louie's gastronomic radar in a positive light. So after you've fulfilled your promises to bring home T-shirts and menus from places belonging to sports mega-legends, check out the following: They're among the spots where the natives eat.

The Berghoff. 17 W. Adams. 312-427-3170. In the heart of The Loop, The Berghoff has been feeding the burghers of the city since shortly after the 1871 fire, and the ambience reflects a bustling Old Chicago of an earlier time. Reasonably priced, the food can best be described as Midwest/German. Louie's informant swears by the sauerbraten, the flourless chocolate cake and the private label dark beer. Warning: It's tempting to begin by scarfing down all the complimentary rye bread on your table with a mug of that beer. Careful or you won't have room for your entrée.

The Parthenon. 314 S. Halsted. 312-226-3377. West of The Loop in Greektown, a fairly short cab ride from McCormick Place. Lively atmosphere. Ohpa! and all that. Great lamb dishes. Louie says the *spanakopita* (spinach pie) is also fairly wonderful. One of his cronies always orders *skordalia* (a veggie and bread dip made of mashed potatoes, garlic and olive oil—trust Louie, this tastes much better than it sounds) and the octopus salad. Good value for your money.

Morton's—The Steakhouse. 1050 N. State (Newberry Plaza). 312-226-4820. This is for serious meat-and-potatoes people. For the most serious of them, there's a 48-oz. double porterhouse. Also live lobster, daily fresh fish choices and salads to keep up with the steak sizes. Pricey and a bit elegant. You might want to take your important clients here. Serves dinner only and reservations are a good idea. Louie says this is definitely a coat-and-tie spot.

The 95th. 875 N. Michigan (Top of the John Hancock Center). 312-787-9596. This is the other place you might want to take the people you most need to impress. Good food elegantly served and an unmatched view of the city, especially at night. Another spot where reservations, a coat, tie and expense account are in order.

Rosebud Cafe. 1500 W. Taylor (312-942-1117) or 55 E. Superior (312-266-6444). The W. Taylor site is the original restaurant in the heart of what was an old Italian neighborhood and now is near-neighbor to the University of Illinois at Chicago. It's closer to McCormick Place than its newer incarnation just off glitzy N. Michigan Ave. Italian food for grownups. Good meat and fish entrees, excellent pasta dishes, fresh hot breads and an impressive wine list. Another case of giant portions: Louie says bring a large appetite or prepare to take away a doggie bag. Rosebud was indeed THE eatery in town not too many years ago, and is still very popular, especially on weekends. Better make reservations.

Reza's. 5255 N. Clark (312-561-1898) or 432 W. Ontario (312-664-4500). Serves Persian food, which is Greek food with a foreign accent. Several types of meat kebabs are good safe choices, but if you're feeling adventurous, the cornish hen smothered in pomegranate-walnut sauce is very good. Some vegetarian items. A bit boisterous, but both rooms welcome large parties. Inexpensive by Chicago standards. The original Reza's on Clark is a bit of a cab ride from McCormick Place and most hotels, but it's in a marvelous, funky, multi-ethnic neighborhood, and many people prefer the atmosphere there. Louie's girlfriend reminds the women in your party: Reza's waiters were listed among the 25 sexiest Chicagoans in a recent issue of a local magazine (and this is without removing their bow ties and black jackets).

La Novita. 1232 W. Belmont. 312-404-8888. More Italian food for people who've gone beyond pizza and pasta with red sauce (although there is very good pizza on the menu). Not as crowded or upscale as Rosebud, it has the look and feel of a friendly neighborhood hang-out. Louie and his friends like the outdoor patio on warm evenings and the free parking behind the building. Owner/chef Giovanni Camaci takes his cooking seriously, and his careful attention shows up on your plate, the wine list and the well-stocked bar.

The Emperor's Choice. 2238 S. Wentworth. 312-225-8800. In the heart of Chinatown, this is only a short cab ride from McCormick Place. It specializes in Cantonese food. Seafood choices are a specialty. The menu here is extensive and elaborate, and Louie suggests the uninitiated might want to ask the waiters for advice. Serves both lunch and dinner and offers a good value for the dollar. ○

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If you found this article of interest and/or useful, please **circle 205.**

Tooth Modification and Spur Gear Tooth Strain

Fred B. Oswald & Dennis P. Townsend

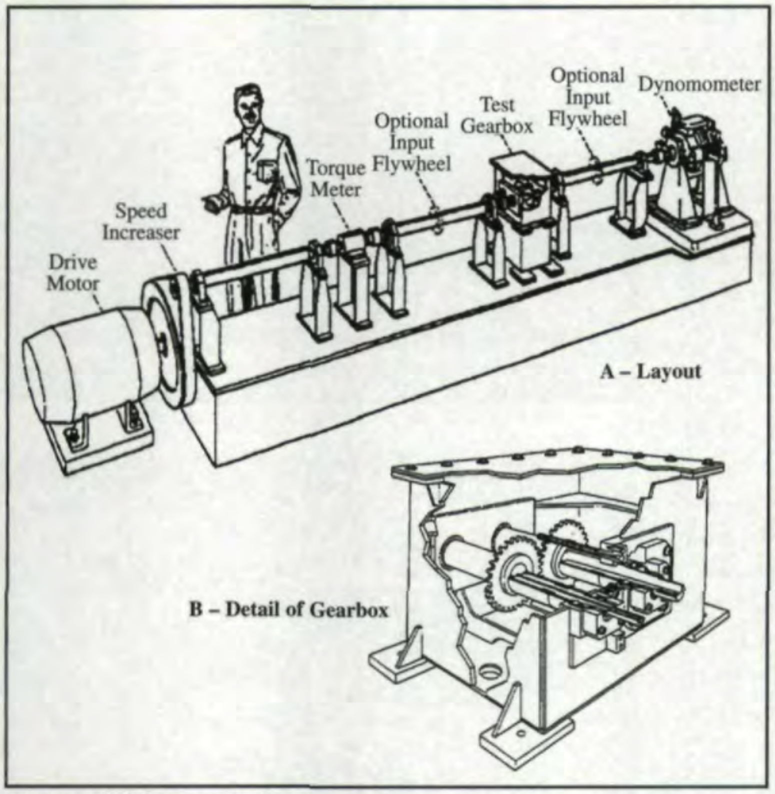

Fig. 1 — NASA gear noise rig.

Fig. 2 — Strain gage installation on test gear.

A major source of helicopter cabin noise (which has been measured at over 100 decibels sound pressure level) is the gearbox. Reduction of this noise is a NASA and U.S. Army goal.

Noise excitation in a transmission is caused by the load fluctuation as gear teeth enter and leave mesh. The cyclic variation in the numbers of teeth carrying the load causes a periodic change in the tooth stiffness and affects the relative position of the teeth. Any deviation in the angular position of the driven gear from its ideal position is called the transmission error. Transmission error arises from manufacturing and mounting errors and from tooth deflection under load.

High-quality gear designs often include modified tooth profiles (tip relief) to minimize transmission error. *Dudley's Gear Handbook* (Ref. 9) provides formulas for "first approximations" of tip relief based on the load and face width. Previous studies of spur gear profile modification include Refs. 2, 4, 5 and 8. Dynamic strain gage testing was reported in Refs. 6 and 7.

The goal for the research reported in this article was to examine the influence of tooth profile modification on dynamic tooth strain by means of controlled tests and to provide a database for further research. Data presented here include involute (tooth profile) charts for the test gears and time domain plots of static and dynamic gear tooth bending strain.

Apparatus

Tests were performed on the NASA gear noise rig (Fig. 1). The rig features a single-mesh gearbox powered by a 150 kW (200 hp) variable speed electric motor. An eddy-current dynamometer loads the output shaft. The gearbox can operate at speeds up to 6,000 rpm. The rig was built to carry out fundamental studies of gear noise and the dynamic behavior of gear systems. It is designed to allow testing of various configurations of gears, bearings, dampers and supports. The gearbox is extensively instrumented for strain, noise and vibration measurements.

A poly-V belt drive was used as a speed increaser between the motor and input shaft. A soft coupling on the input shaft reduces the input torque fluctuations caused by non-uniformity of the belt splice.

The gearbox oil inlet temperature was maintained at 70 ±2°C for these tests. At the mean temperature of 70°, the viscosity of the synthetic turbine engine oil (MIL-L-23699B) used in the tests is 9.5 centistoke.

Instrumentation. General-purpose, constantan foil, resistance strain gages with gage length 0.38 mm (0.015") were installed in the tooth-root fillets on both the loaded (tensile) and unloaded (compression) side of two adjacent teeth on the output (driven) gear (Fig. 2). To measure maximum tooth bending stress, the gages were placed at the 30° tangency location (Ref. 1). Two methods of signal conditioning were used on strain gage signals: For static measurement, a strain gage (Wheatstone) bridge was used. For dynamic measurements, the strain gages were connected via a slip-ring assembly to constant-current strain gage amplifiers.

An 8-channel, 12-bit digital data acquisition system was used to record the dynamic strain data. The sample rate was varied from 6.6–50 kHz per channel to provide 500 samples per revolution for each channel. An optical encoder on the input gear shaft produced an accurate once-per-revolution pulse. The encoder was adjusted so the leading edge of its pulse occurred at a known roll angle of the gear. This allowed us to determine the roll angle at any point in the data record.

Test Gears. The test gears were identical spur gears (at 1:1 ratio) machined to master gear (AGMA Class 15) accuracy. Test gear parameters are shown in Table 1. Profile modifications were chosen to compensate for tooth deflection under load. No additional allowance was made for manufacturing errors, since these errors were not more than one-tenth of the computed deflection at the nominal load of 71.8 N-m (635 lb-in).

Six different gear profiles (Fig. 3) were tested. These include an unmodified profile, three combinations of linear profile modification (tip relief) and two different forms of parabolic modification. Linear modification is defined by two parameters: The amount of modification at the tip and the roll angle at the start of modification. For parabolic modification, a third parameter is needed. We identified two forms of this third parameter: In type-1 parabolic relief (see Fig. 3e), the modified profile blends smoothly into the involute trace. (It is tangent to the involute at the start of modification.) In type-2 parabolic relief (see Fig. 3f), the modified portion of the curve blends into the edge break at the end of the tooth. (It has an infinite slope at the tip.) These gears were made to the AGMA Class 15 quality level. Even so, there is not much apparent difference between the traces for set E (type-1) and set F (type-2) gears.

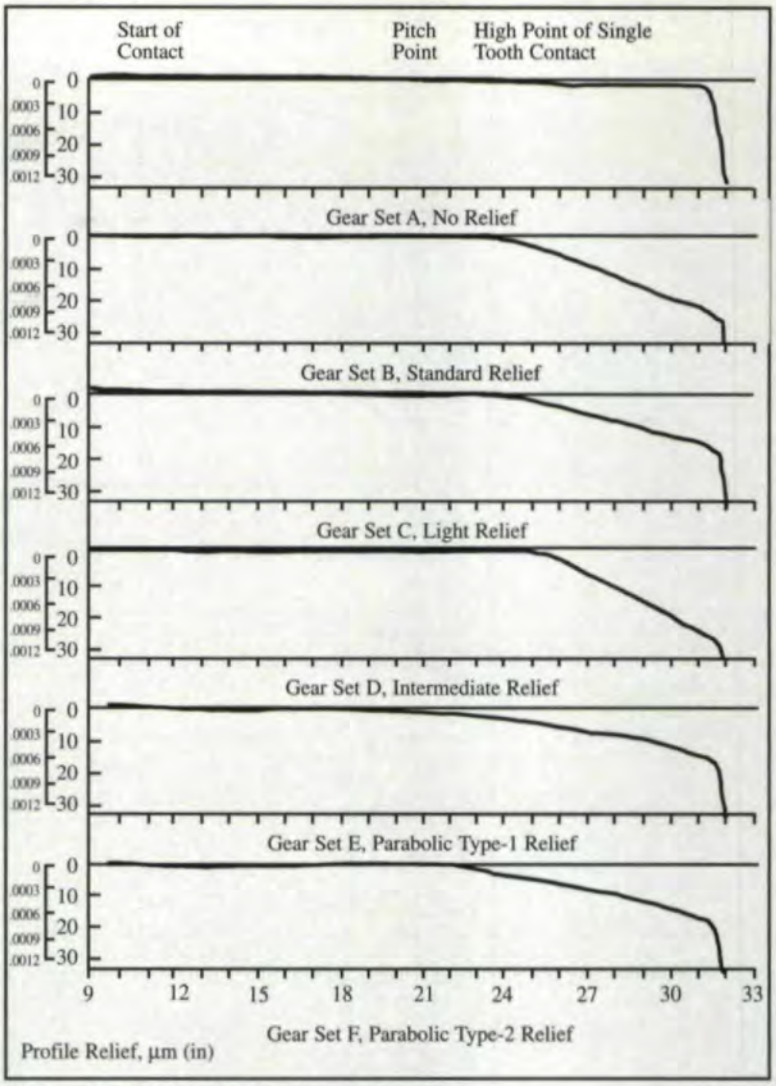

Fig. 3 — Profile charts for the six sets of test gears.

Table 1 — Test Gear Parameters	
Gear Type	**Standard, Full-Depth Tooth**
No. Teeth	28
Module, mm (Diametral Pitch, in^{-1})	3.175 (8)
Face Width, mm (in)	6.35 (0.25)
Pressure Angle, deg	20
Theoretical Contact Ratio	1.64
Tooth Root Radius, mm (in)	1.35 (0.053)
Max. Tooth Spacing Error, µm (in)	1.8 (0.00007)
Max. Profile Error, µm (in)	1.3 (0.00005)

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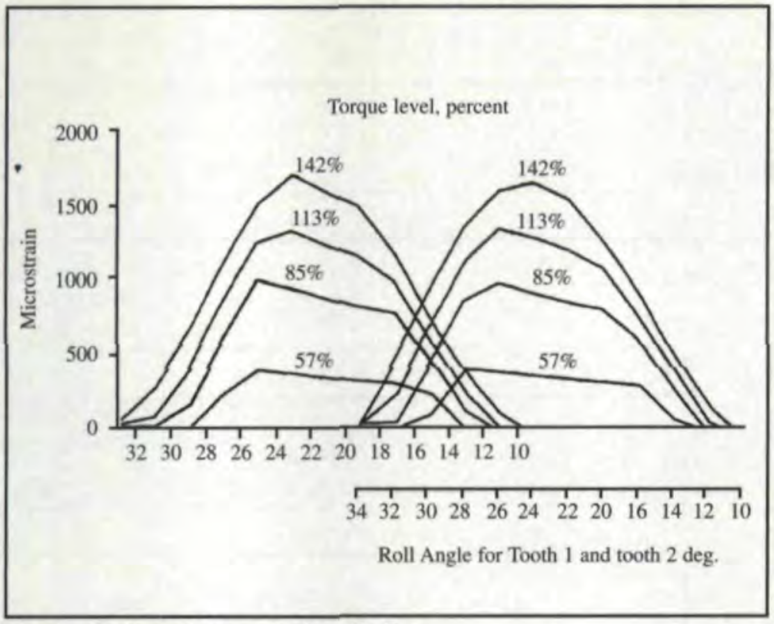

Fig. 4 — Static strain traces for Gear Set B.

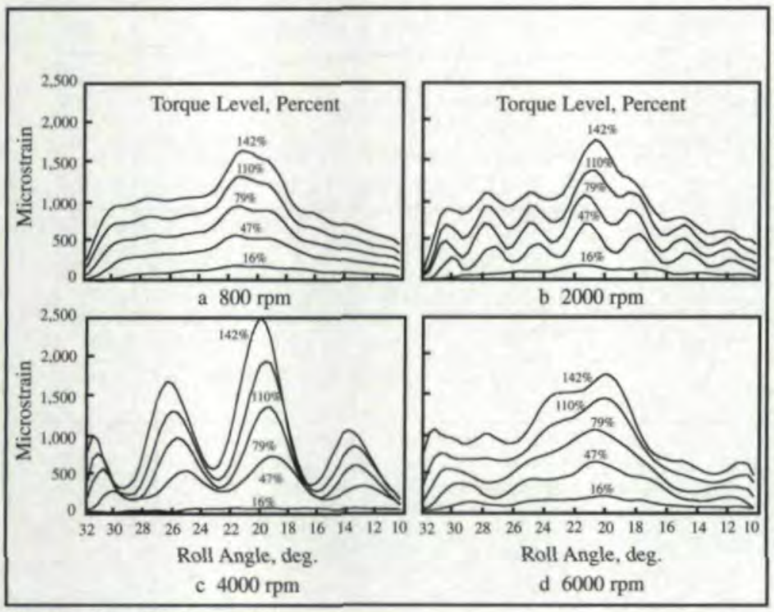

Fig. 5 — Gear Set A, no relief.

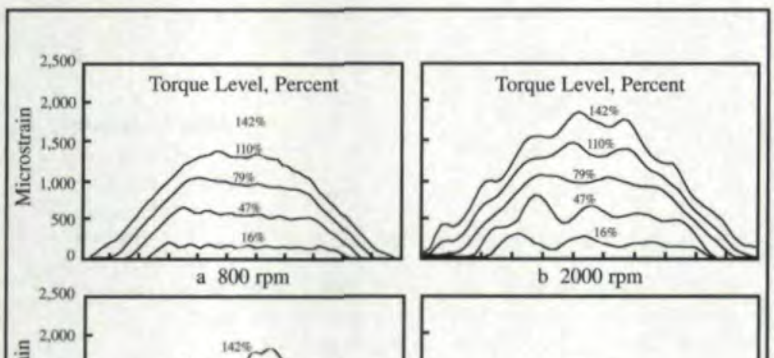

Fig. 6 — Gear Set B, standard relief.

Test Procedure

Static Strain Data. Strain data was recorded under static (non-rotating) conditions. Static measurements provide information on load sharing characteristics of the gear pairs. Strains were recorded for roll angles from 32° to 10° roll angle for each tooth. (Note: the strain gages are on the driven gear, hence contact starts near the tip and proceeds towards lower roll angles at the root of the tooth.) Readings were taken at torque levels of 57%, 85%, 113% and 142% of the nominal torque of 71.8 N-m (635 lb-in).

Dynamic Strain Data. Dynamic strains were recorded for each of the six gear pairs over a matrix of load-speed test conditions: 4 speeds (800, 2,000, 4,000, 6,000 rpm) and torque levels of 16%, 47%, 79%, 110% and 142% of the nominal torque of 71.8 N-m (635 lb-in). The data were then digitally resampled using linear interpolation at 1,000 samples per revolution and synchronously averaged. Time domain synchronous averaging is a technique now in wide use in gear diagnostics (see Ref. 3), and was used here to reduce random "noise" effects (such as torque fluctuation caused by the belt drive). Its implementation requires at least two channels of data—a timing signal plus the data of interest. The timing signal provided the resample intervals needed for exactly one revolution of the gear.

Results and Discussion

Static strain data was collected primarily for use in an effort to compute normal and frictional loads between gear teeth (see Ref. 7). Fig. 4 shows static strain data from two consecutive teeth on gear set B (standard "long relief" profiles) taken at four different torque levels. Static strain readings were recorded at two-degree increments, except an extra reading was taken at 21°, which is near the pitch point (20.85°). Comparing readings from the strain gages mounted on adjacent teeth provides an indication of the accuracy and consistency of the strain gage installations. For all the gears tested, the maximum difference (worst case) in measured static strain between the two tensile gages was 4.6%.

The dynamic strain data for the six test gear designs is shown in Figs. 5–10. These figures show strains measured by one of the load-side gages as sets of parametric curves, each set for a single gear design, at a single speed, but at several torque levels.

Set A—Unmodified Gears. The gears designated Set A have essentially a true involute profile (see Fig. 3) up to the edge break at about 31° roll angle. Ref. 4 shows that the transmission error of a perfect involute gear is zero at no load (torque), but increases with the load. Therefore, we would expect these gears to show increasing dynamic action as torque increases. This is indeed the case. Low-speed measurements (Fig. 5a) show very smooth operation with little dynamic excitation. Tooth contact extends from about 32° to 10°. Two pairs of teeth are in contact except in the single contact zone, which extends from about 22° to 18°.

As the speed increased to 2,000 rpm (Fig. 5b), dynamic effects become apparent, especially at the higher torques. A regular pattern of waves can be seen in the strain curves. At 1,000 rpm, one tooth pitch period occupies about 13° of roll angle (360° divided by 28 teeth). In this span there are about four cycles of strain. This indicates that the dynamic loading frequency is four times the tooth mesh frequency. At 4,000 rpm (Fig. 5c), the dynamic action is much stronger, and there are two strain cycles per tooth pitch. At 6,000 rpm, the dynamic effects are not as strong and there is no regular wave pattern.

Set B—Standard Relief**.** The strain data from Set B gears is shown in Fig. 6. Because of dynamometer control problems at the time these data were taken, the highest torque (142%) curve is missing in Figs. 6a and 6d.

Set B gears have linear profile modification (Fig. 3) extending from the high point of single tooth contact to the tip. Munro calls this "long relief." (In contrast, "short" relief has a modification zone one-half as long.) The amount of relief at the tooth tip corresponds to the tooth deflection expected from a torque level of 115% of the nominal 71.8 N-m.

The low speed traces (Fig. 6a) show that the single contact zone is longer than in the Set A gears. The profile modification has apparently decreased the contact ratio. At higher speeds (Figs. 6b to 6d), there was smooth operation near the design torque, but much rougher operation away from design torque, especially at high-speed, low-torque conditions. At 4,000 rpm and 16% torque, there are two strain "spikes" at the beginning and end of tooth contact, and the strain is zero at the pitch point. This indicates that the teeth bounce out of contact. At 6,000 rpm and 16% torque, there is a single large strain spike. This indicates relatively high dynamic loading.

Set C—Light Relief**.** The gears from Set C are similar to Set B except the amount of relief is much less, corresponding to the deflection for 70% torque. As would be expected, these gears operate more smoothly at light torque, but with increased dynamics at higher torques.

Set D—Intermediate Relief**.** Set D gears have a shorter relief zone than Sets B or C. Munro in

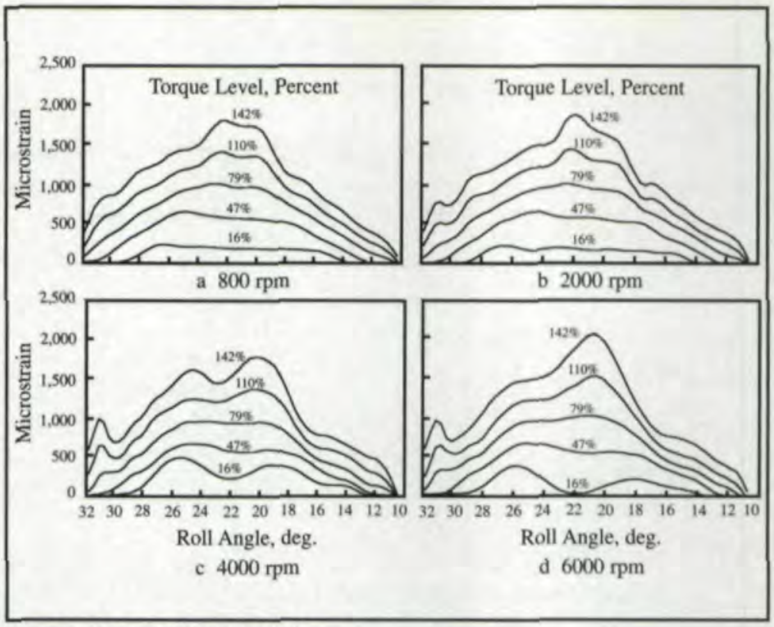

Fig. 7 — Gear Set C, light relief.

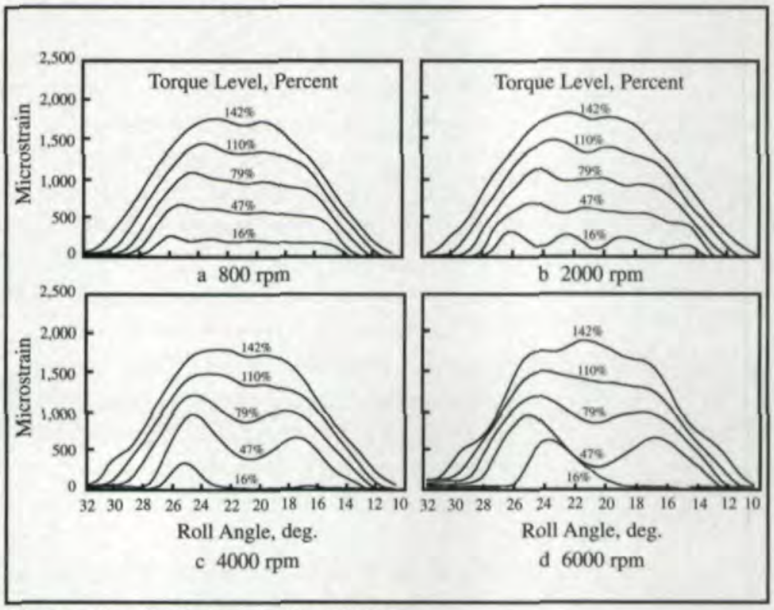

Fig. 8 — Gear Set D, intermediate relief.

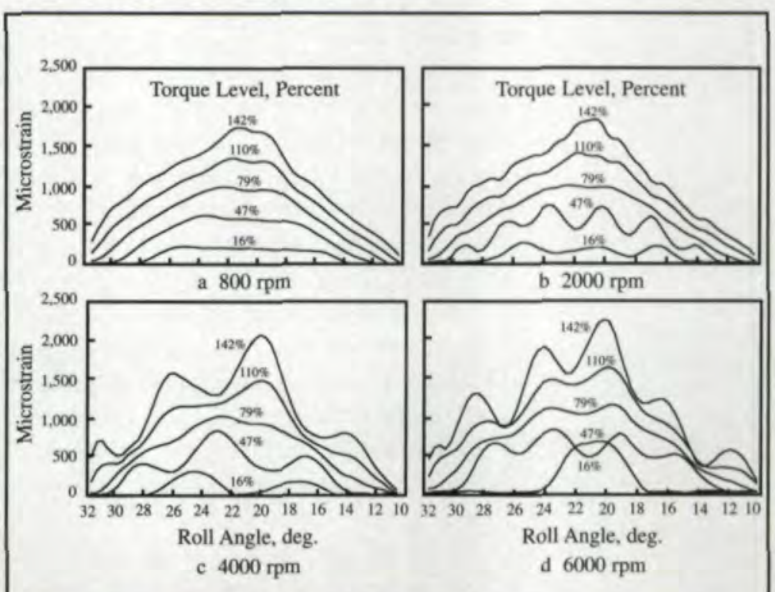

Fig. 9 — Gear Set E, parabolic type-1 relief.

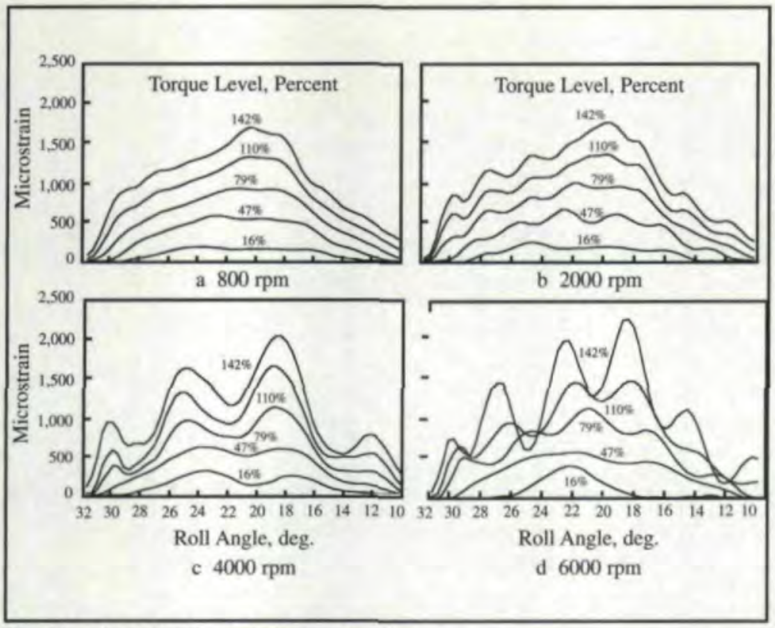

Fig. 10 — Gear Set F, parabolic type-2 relief.

Ref. 4 calls this "intermediate relief" and recommends it for gears which operate under a range of loads. The amount of relief at the tooth tip corresponds to 135% torque. The results (Fig. 8) show very smooth operation at high torque, and light torque operation is comparable to Set B. Since they show improvement at high torque and no worse operation at low torque (compared to Set B), intermediate relief is an improvement over long relief.

Set E—Parabolic Type-1. These gears are similar to Set C gears except the modification zone extends to the pitch point. (Munro calls this "extra long relief.") The modification at the tip corresponds to torque of 65%. These gears were difficult and expensive to manufacture. Their performance was disappointing. The dynamic strains were large at both high and low torques. Results may have been better if the relief zone were much shorter.

Set F—Parabolic Type-2. These gears are similar to Set E except the modification amount at the tip corresponds to the tooth deflection at 85% torque, and the modification zone is slightly shorter. Like Set E, these are difficult and expensive to manufacture. The performance was similar to that of Set E.

Summary and Conclusions

Low contact ratio spur gears with various profile modifications were tested in the NASA gear noise rig. Dynamic tooth bending strains were recorded for each gear design at 36 operating conditions. The experimental results were compared to examine the influence of the tooth profile on the dynamic behavior of the gears under various operating conditions. The following conclusions were drawn from the data:

1. The proper type and amount of tooth profile modification can significantly reduce dynamic loads in spur gears, especially for gears that operate at high speed and under high torque.

2. The parabolic modification gears tested here seem not to offer any advantage. This may be because the modification zone is too long.

3. Profile modification increases dynamic loads in gears that operate significantly below their design torque. This is especially so for a long modification zone.

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Alternative Lubrication Methods for Large Open Gear Drives

Dennis A. Lauer, P.E.

The type of lubricant and the method of applying it to the tooth flanks of large open gears is very important from the point of view of lubrication technology and maintenance. When selecting the type of lubricant and the application method, it is important to check whether it is possible to feed the required lubricant quantity to the load-carrying tooth flanks. This is necessary to avoid deficient lubrication, damage to the gear and operational malfunctions. It is important to determine the type of lubricant, which may be fluid or grease-like. The consistency of the lubricant will have a direct impact on the ability of the lubrication system to feed adequately the lubricant to the gear. The interactions between the common types of lubricant and the lubrication application methods for open gear drives are shown in Fig. 1.

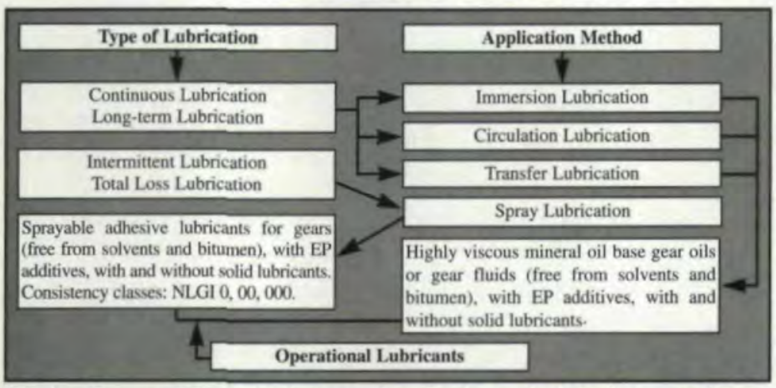

Fig. 1 — Common types of lubrication and application methods for large gear drives.

Basically, there are two types of lubrication for open gear drives: Continuous lubrication (long term lubrication) and intermittent lubrication (total loss lubrication). With both these types of lubrication, several application methods are possible.

Continuous Lubrication

Continuous lubrication means that the lubricant is fed to the friction point (the tooth meshing zone) without interruption. The lubricant volume at the load-carrying tooth flanks can vary between an extremely high and a minimal amount. Continuous lubrication may be provided by immersion lubrication, transfer lubrication with a paddle wheel or circulation lubrication.

Immersion Lubrication

Generally, immersion lubrication is one of the safest methods of applying a lubricant to an open gear drive. To function properly the lubricant bath must remain adequately filled and the gear guard cover properly sealed to avoid lubricant losses. Modern fluids, which were especially developed for open gear drives and whose consistency and flow behavior were designed specifically for this application method, increase the efficiency of immersion lubrication considerably.

It is important to know the operational limits of these modern fluids to help avoid critical situations caused by ambient temperature extremes. The safety of the immersion lubrication system is based on the fact that either the pinion or the gear

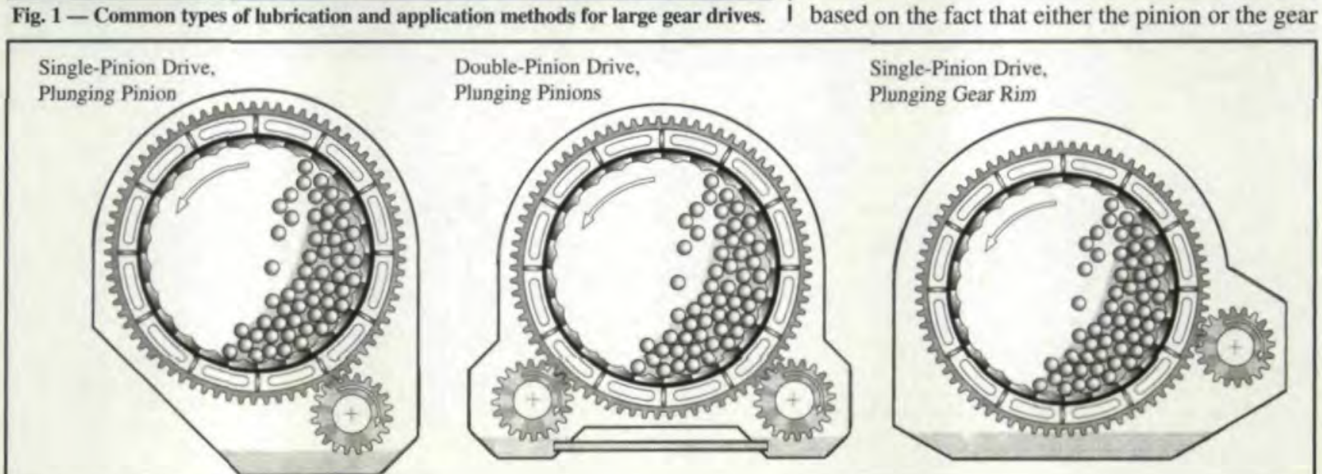

Fig. 2 — Types of immersion lubrication.

is in direct contact with the lubricant reservoir (Fig. 2). To ensure this system's reliability and prevent deficient lubrication, it is important to regularly compensate for lubricant losses from leakage or lubricant discharge through the gear rim seals. Inadequate sealing between the gear guard and the gear rim will result in dust, sand, clinker, water, etc. penetrating into the immersion bath. These contaminants will eventually reach the intermeshing gear tooth zone and result in increased abrasive wear of the tooth flanks. The immersion bath should be replaced at regular intervals to prevent damage to the tooth flanks caused by contamination.

The lubricant used in the immersion system should meet the following requirements to ensure a reliable and safe operation of the gear drives for a long period of time. It should

- be solvent-free.
- have good back flow behavior; channeling should not occur at ambient temperatures.
- have suitable viscosity/temperature behavior so that the bath must be neither heated nor cooled.
- have low evaporation losses.
- have easy replacement and disposal.
- have load-carrying capacity and antiwear behavior confirmable on the FZG gear testing rig.

Transfer Lubrication

In this type of lubrication, also called paddle wheel lubrication, paddle wheels plunge into the lubricant reservoir and then transfer the lubricant to the driving pinions. This method has the advantage that smaller amounts of lubricant are transferred to the tooth flanks and a smaller amount of excess lubricant is in circulation at the drive. Fig. 3 shows an example of paddle wheel lubrication of a rotary kiln with a double-pinion drive. The paddle wheels are located directly below the pinions and are driven by them. Paddle wheel lubrication is only suitable for spur-toothed gear drives and is mainly found on slowly operating kiln drives.

Circulation Lubrication

In circulation lubrication, the lubricant is transferred by means of externally driven pumps. The main advantage, as compared to immersion lubrication, is the fact that the lubricant is filtered and then applied in an excess amount to the tooth flanks almost without contaminants. Circulation lubrication will be successful only if the gear guard is sealed properly and the penetration of contaminants from the environment into the lubricant reservoir is prevented as much as possible. Most circulation lubrication systems used today are designed for the application of gear oils. Some systems are also suitable for applying high-viscosity open gear fluids used in running-in lubrication, as well as operational lubrication. The typical circulation lubrication system is suitable for lubricating drives on kilns and mills and can also be retrofitted to existing installations. For large open gear drives, its main advantage is that the lubricant is continuously cleaned by filters and that it is applied to the tooth flank surfaces very efficiently through special lubricant distribution pipes. Fig. 4 is a schematic of a circulation lubrication system for a double-pinion drive.

A schematic of a circulation lubrication system with various accessories is identified in Fig. 5. From the lubricant reservoir (1.1), which is filled directly from the gear guard (4), or by means of a transfer pump (5), the lubricant is fed by gravity flow to the lubricant pump (2.1) via a

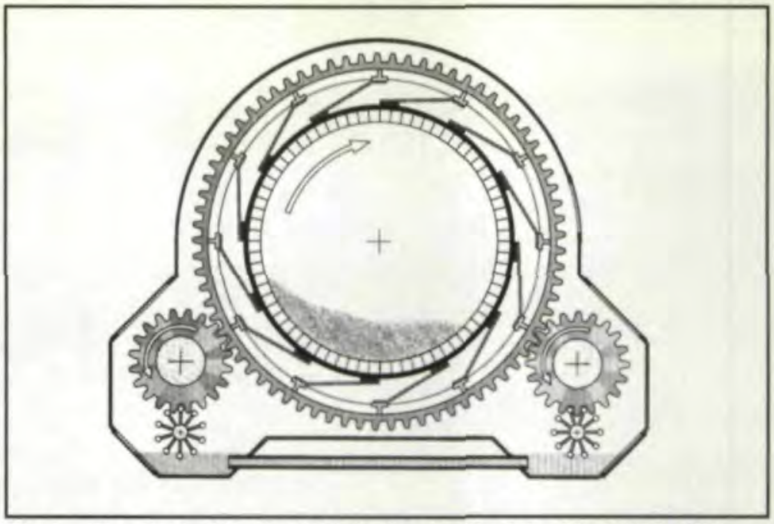

Fig. 3 — Transfer (paddle-wheel) lubrication, double-pinion kiln drive by FLS.

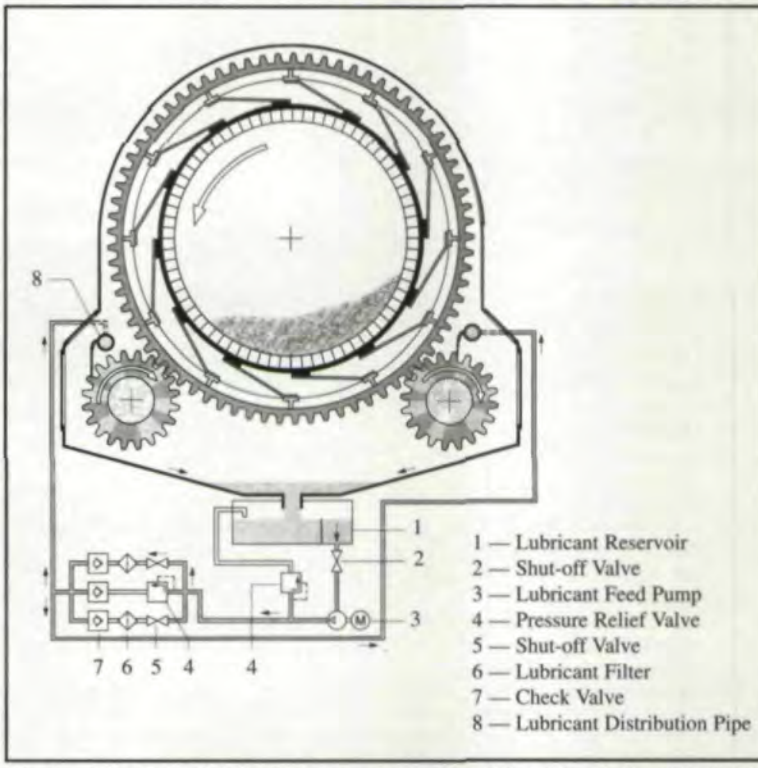

Fig. 4 — Circulation lubrication system for double-pinion drives (schematic view).

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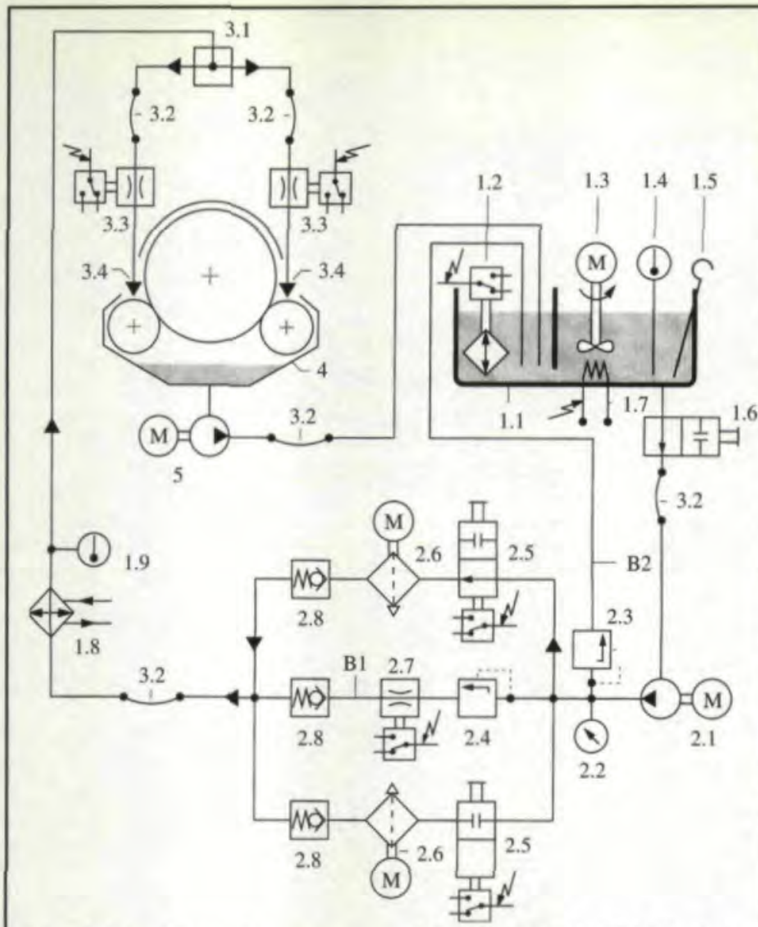


Fig. 5 — Schematic of a circulation lubrication system with various accessories.

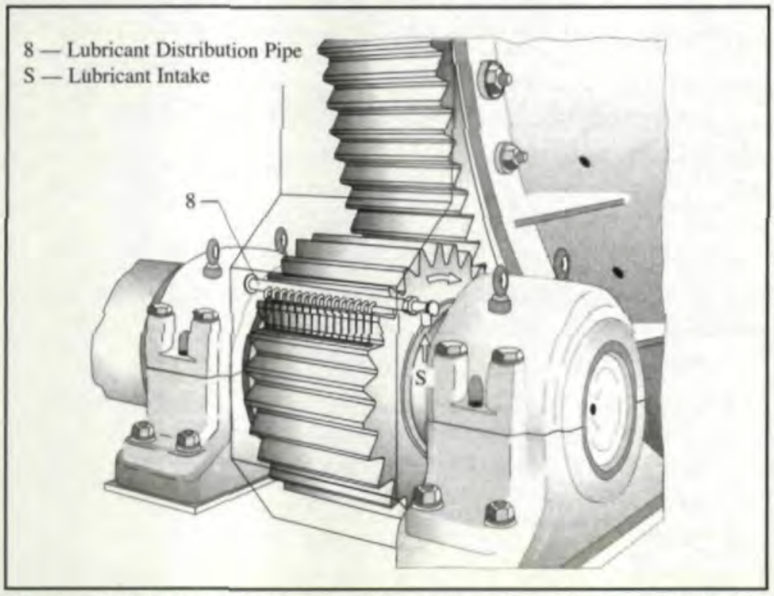


Fig. 6 — Lubricant application through specially designed lubricant distribution pipes.

pipe system. Depending on the position of the valves (2.5), it is transferred to one of the two filter units (2.6), which are equipped with a motor-driven cleaning mechanism. Since only one filter is operating at a time, the other can be maintained and cleaned without interrupting the flow of lubricant to the gears. Switching from one filter unit to the other can be accomplished manually or automatically. The recommended filter size is between 80 and 100 microns.

After passing the filter unit, the lubricant is fed through a distributor (3.1) to the distribution pipes (3.4). A check valve (2.8) prevents the lubricant from flowing backwards. If there is not enough flow through the operating filter unit (2.6), the pressure relief valve (2.4) will open. The lubricant is then supplied to the distribution pipes via the bypass line (B1). A flow control unit (2.7) installed in the bypass line will trigger an alarm, which will warn the operator to change the flow to the other filter line. The lubricant flow is then directed through the second filter unit, and the pressure relief valve (2.4) will close as soon as the pressure is reduced. The monitoring of system alarms is very important in order to prevent prolonged application of unfiltered lubricant to the tooth flanks. It is also possible to have an automatic system to control the flow direction into each filter unit based on pressure drop in the filter lines.

Flow control monitoring is also located before the distribution pipes to ensure that there is lubricant flow to the gear teeth. Fig. 6 shows the location of the lubricant distribution pipe.

If required, the lubricant reservoir (1.1) can be equipped with a heating system (1.7) and agitator (1.3). If cooling of the lubricant is necessary, a heat exchanger (1.8) can be installed in the lubricant line in front of the distribution lines (3.4). If the space available directly below the gear guard is restricted and it is not possible to install a lubricant reservoir there for a gravity flow, then a transfer pump (5) can be used.

Intermittent Lubrication

Intermittent lubrication means that the lubricant is applied at intervals. This type of lubrication is always total loss lubrication; therefore cost effectiveness has to be taken into account. Just as in the case of continuous lubrication; there are various methods of application of intermittent lubrication. For modern and safe lubrication of large girth gear drives, however, only two methods of application are used: Manual lubrication by spray gun and automatic spray lubrication.

Manual Lubrication by Spray Gun

Currently, the most effective type of manual lubrication for large gear drives is by means of pressurized manual spray guns. They are best suited for operational lubricants in cases where stationary automatic lubrication systems cannot be installed and where lubrication is required during operation. A typical manual spray lubrication system is depicted in Fig. 7. This portable equipment consists of a pressurized lubricant container with valves, a heavy-duty spray gun and connecting hoses for the lubricant and pressurized air. This system is easy to handle and only needs to be connected to the plant air system before it can be put into operation. With the spray gun, which can be adjusted to a round or flat jet, different types of lubricants can be applied:

• Operational lubrication with very thin lubricating film and economical consumption.

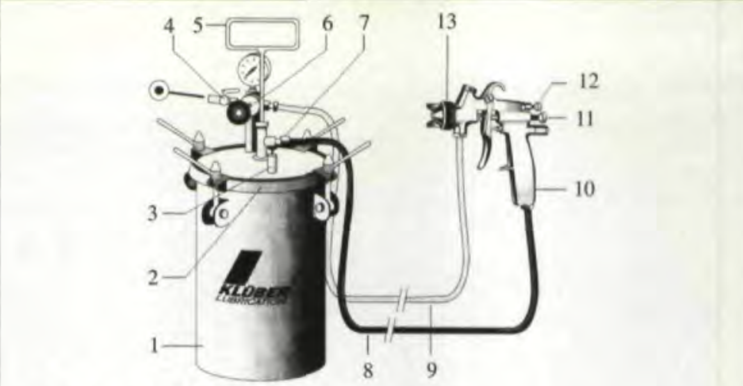
• Priming of open gear drives to check the contract and load-carrying pattern.

• Lubricant application for repair correction or forced run-in lubrication.

Automatic Spray Lubrication

Manual lubrication by means of a spray gun as described in the previous section has its limits in running-in and operational lubrication, especially when large drives have to be lubricated adequately and reliably. Automatic spray lubrication is especially suitable for gear drives that, due to their design and inefficient sealing of the gear guard, are not suitable for other types of lubricant application, and where heat dissipation by the lubricant is not necessarily required. Various manufacturers offer automatic spray lubrication systems of different designs, all of which are state of the art and are capable of handling operational lubricants with a high solid lubricant content, as well as special running-in lubricants. Ideally, the amount of lubricant sprayed onto the tooth flanks maintains a minimum thickness of lubricant film. This thickness is constantly diminished by the motion of the two flanks against each other. If the necessary film thickness is maintained, it will prevent scuffing caused by insufficient lubrication. However, this is hardly feasible for technical reasons.

It is quite obvious that during operational lubrication it is not possible to continuously apply a lubricant film of optimum thickness to the tooth flanks. A good compromise is found in interval lubrication, consisting of periods of excess lubrication. For operational reliability, the most decisive factor is the duration of the individual spraying pulse. It is best when the entire circumference of the pinion or the girth gear is sprayed and covered with lubricant in one pulse. The lubricant amounts must ensure a film thick enough to reliably last through the ensuing



1 — Deposit 10 l, Max. Operating Pressure 3 bar
2 — Removable Cover
3 — Pressure Relief Valve (Depressurization)
4 — Shut-off Valve, Connection to Pressurized Air System
5 — Carrying Handle
6 — Adjustable Pressure Reducing Valve to Set the Lubricant Pumping Pressure
7 — Connection for Tube Nozzle
8 — Lubricant Tube
9 — Compressed-Air Tube
10 — Spray Gun, Fan or Circular Jet
11 — Adjustment Screw to Control the Lubricant Throughput
12 — Adjustment Screw to Control the Compressed Air Throughput (Fan or Circular Jet)
12 — Spray Head, Vertical or Horizontal Adjustment of the Fan Jet

Fig. 7 — Manual spraying equipment.

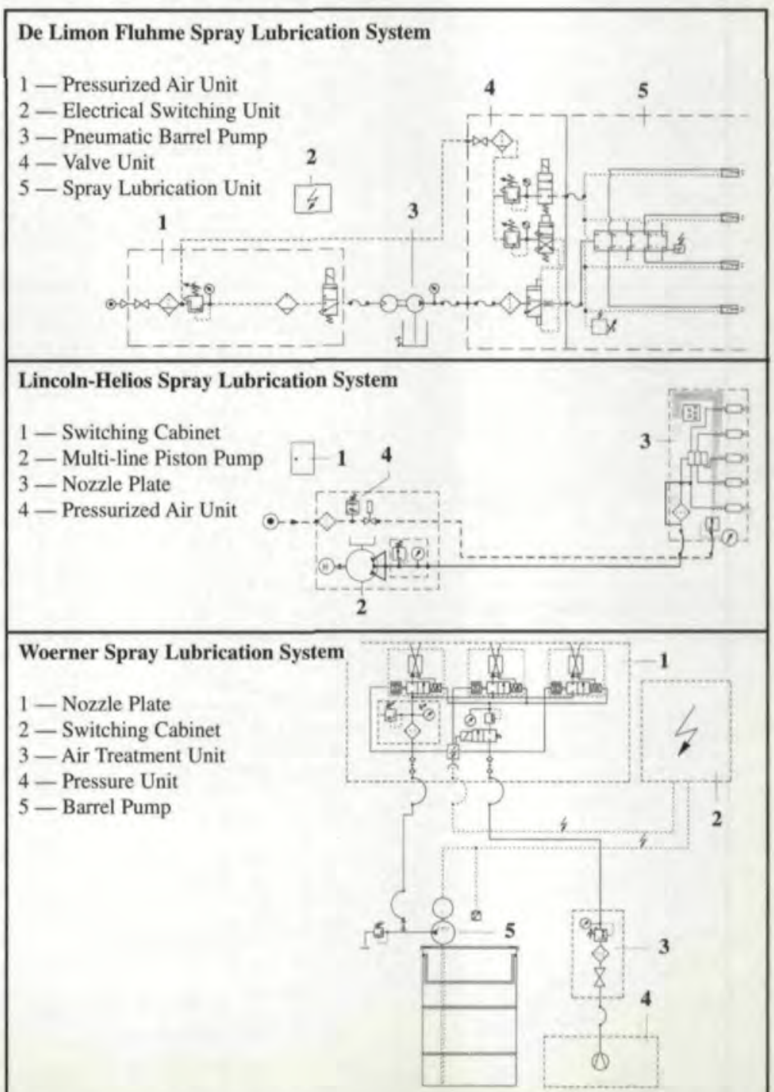


Fig. 8 — Spray lubrication systems.

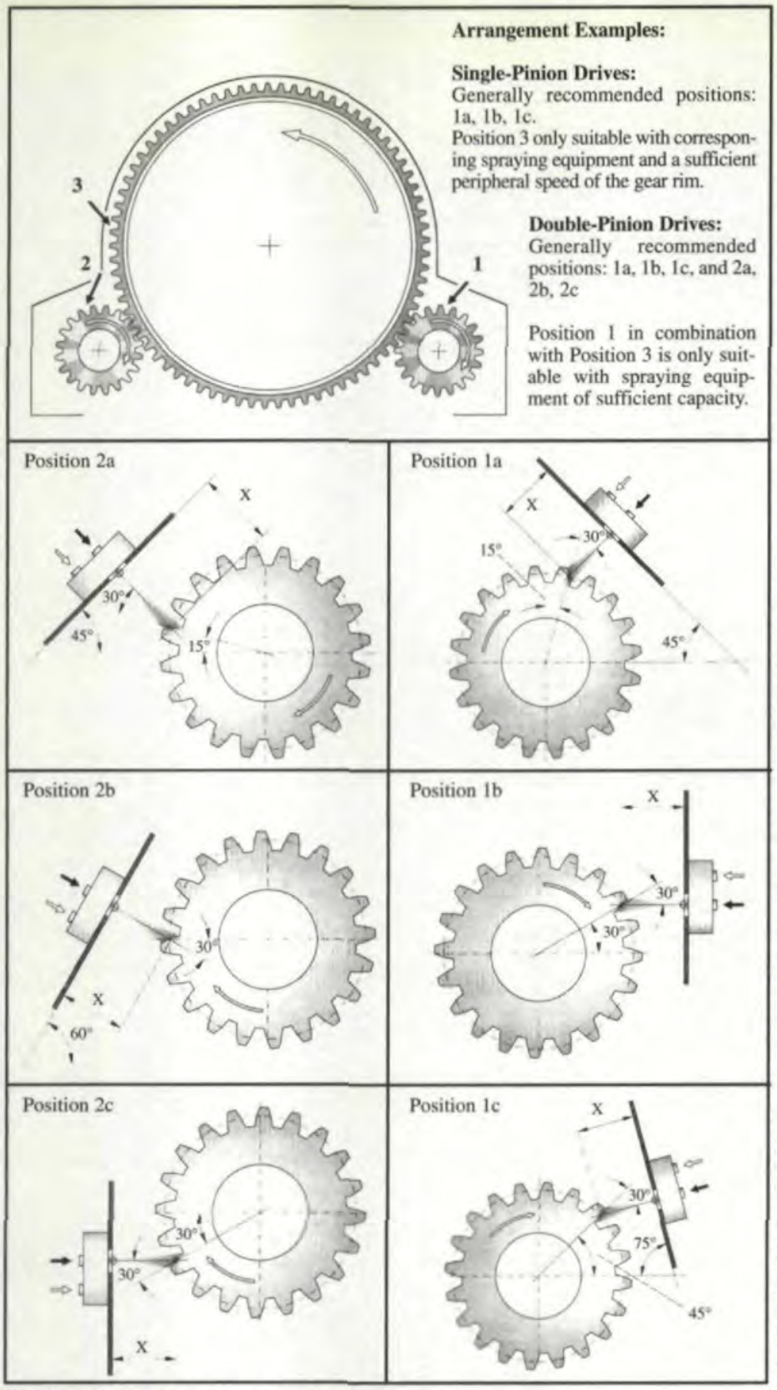


Fig. 9 — Nozzle plate arrangement.

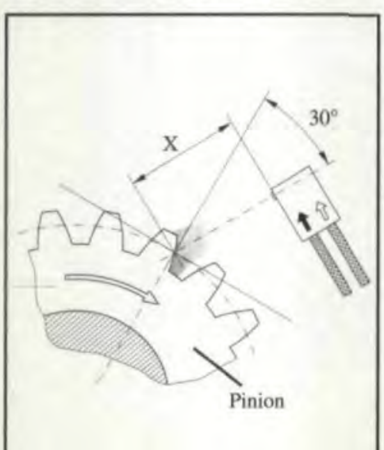


Fig. 10 — Preferable orientation of a spray nozzle.

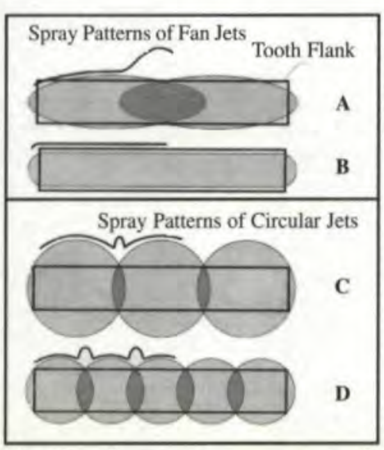


Fig. 11 — Spray patterns on gear tooth flank surface, depending on the number of nozzles and the spray geometry.

pause cycle. More frequent spray cycles with the shortest possible pauses result in the maximum operational reliability in spray lubrication.

Fig. 8 shows schematic diagrams of spray lubrication systems from different manufacturers that are used today for spraying lubricants on large girth gear drives. These systems differ in their design and components as follows:

• Lubrication pumps, e.g. electromechanical multi-line piston pumps (container pumps) with manual filling and automatic filling by means of pneumatic barrel pumps, or direct pumping by means of pneumatically or electromechanically driven drum pumps.

• Single or dual-line system design.

• Lubricant feeding to the spray nozzles either direct from the lubricant pump or via an intermediate progressive distributor.

• Auto-control spray nozzles (controlled by the lubricant and/or air) or externally controlled nozzles with or without monitoring units.

Fig. 9 shows the optimum arrangement of nozzle plates found by practical experience. Positions 1 and 2 are generally preferred. Spraying of the girth gear in position 3 should only be considered if it can be ensured that the entire circumference of the gear is covered with lubricant during one spray pulse. This applies especially to slow running kiln gears. Depending on the design of the gear guard, the nozzle plates should be arranged at the angles and positions indicated in Fig. 9. The spray nozzles should not be installed in a vertical upward direction, since they could become clogged by used lubricant. This would result in insufficient spray patterns and eventually in a complete failure of the lubricant supply system. When determining the position of the nozzles, especially when the girth gear is to be sprayed, the safety of the maintenance personnel when working on the system must be considered.

In Fig. 10, the orientation of the spray nozzle is indicated. A spraying angle of 30° is best to achieve a good distribution of the lubricant on the load-carrying tooth flanks. The nozzle distance should be between 150 and 250 mm. The exact distance depends on the position and the type of nozzle. Each type of nozzle requires a specific nozzle distance, which depends on the air pressure as determined by the manufacturer and which ensures the entire width of the tooth flank is covered with lubricant. This distance also must take into account the number of nozzles and the type of lubricant. Operational safety is reduced if the nozzles are not oriented properly.

An important factor of operational reliability is a perfect spray pattern without any gaps.

The lubricant must be distributed evenly over the entire height and width of the tooth flanks (Fig. 11). Even if the spray nozzles are automatically controlled, periodic checking of the spray pattern is an essential part of maintenance. Due to their design, older spray systems have the disadvantage that the spray pattern in high-speed drives, such as mills, can only be checked when the drive is stopped. This disadvantage can be removed by newly developed nozzle plates that make it possible to check the spray pattern while a drive is in operation.

Fig. 12 shows various nozzle plate designs. This is of special advantage on kiln drives, which cannot be stopped at random. In addition, this type of checking ensures maximum protection against accidents because it is not necessary to remove the gear guard.

Depending on the system design, the lubricant can be delivered from a container to a lubricating system by different means:

- Spray lubrication systems with container pumps (Fig. 13).
- Spray lubrication systems with drum pumps (Fig. 14).

The lubricant containers in the first system are filled manually or by means of a transfer pump out of the original package. The transfer pump is preferred because it prevents contamination of the lubricant. In the second system, the lubricant is fed directly to the spray system with the required operating pressure by means of a drum pump that is put into the original lubricant drum. Feed pumps may be driven electromechanically or pneumatically.

Summary

The goal of the lubrication system for large gear drives is to provide the optimum amount of lubricant at all times to the gearing system. This will prevent premature wear and expensive gear replacement. All of the systems discussed can provide this goal as long as they are maintained and operated properly. The cost effectiveness of each system must be analyzed by the individual user.

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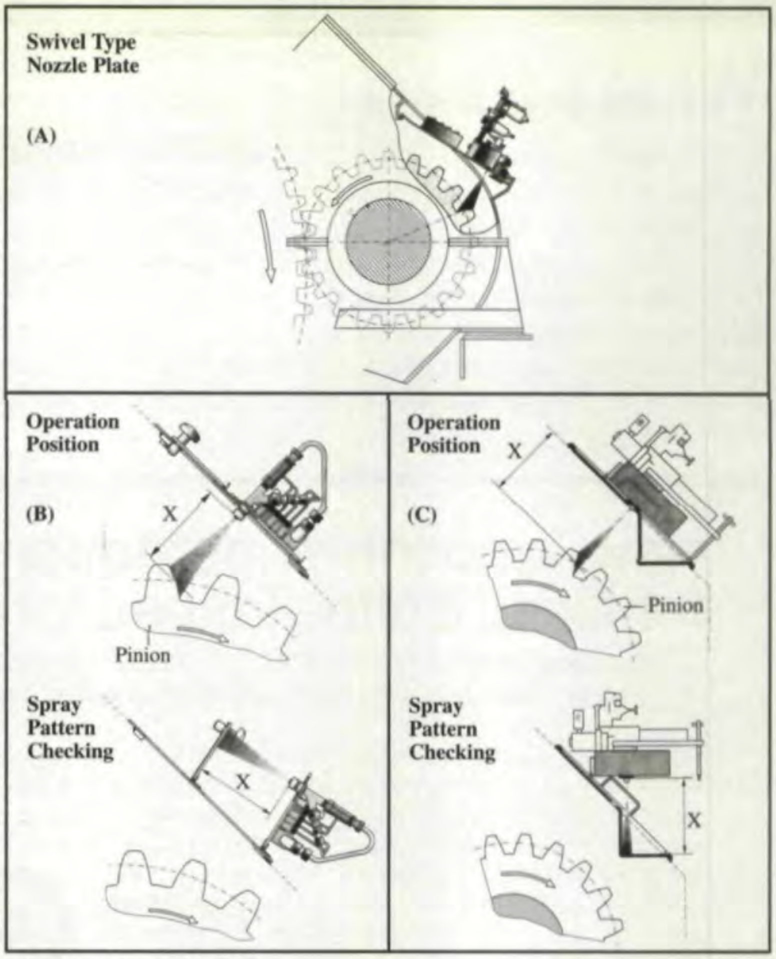

Fig. 12 — Nozzle plate systems.

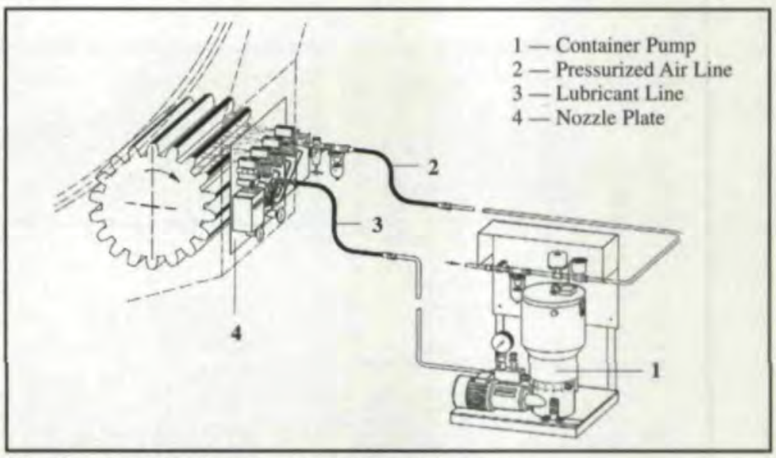

Fig. 13 — Spray lubrication system with container pump

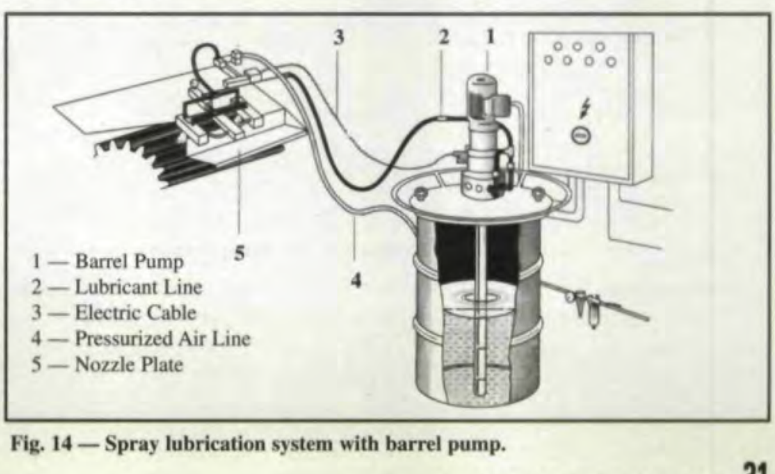

Fig. 14 — Spray lubrication system with barrel pump.

What's News

The Latest Moves

Louis W. Ertel has joined **Foote-Jones/Illinois Gear**, Chicago, IL, as vice president/general manager. Foote-Jones is a manufacturer of custom gearing, enclosed drives, gearboxes and other power transmission products. Mr. Ertel comes from **Prager, Inc.** of New Orleans, LA, where he was vice-president of manufacturing.

Hotline

Nord Gear Corporation of Waunakee, WI, announces the opening of its new 24-hour, 7-day-a-week customer service hotline, which will give customers access to Nord service personnel. A phone call to the 800 number will activate a beeper where the customer can leave a phone number for immediate response from a technician. Nord promises to be able to ship new product to anywhere in the U. S. within three hours of a phone call.

Also Known As . . .

LMT-Fette, Inc., formerly known as **Fette Tools Systems Inc.**, is part of the new LMT-Fette Group, composed of Fette, Boehlerit and Kieninger. LMT-Fette will have a new, enlarged cutting tool program including products for the mold and die making, automotive, aircraft and general metalworking industries.

Cybercompanies

Sunnen Products Co., manufacturers of industrial honing machines and automotive shop equipment, have opened a site on the World Wide Web. Accessible at *http://www.sunnen.com,* the site allows users to review listings of all available Sunnen catalogs, product brochures, technical bulletins and reference material. Information on machines, abrasives, tooling and honing fluids is also available.

CSI, provider of integrated maintenance systems and services for machinery diagnostics, has opened a web site at *http://compsys.com.* The site provides product information, news releases, training, services and customer support information.

South American Connection

Surface Combustion, Inc., manufacturer of industrial heat treating furnaces, has announced a product and technology license agreement with **Sauder Equipamentos Industriais Ltda.** of Brazil. Sauder will represent Surface Combustion in Brazil including product sales, service and manufacturing. Sauder Equipamentos is headquartered in Sao Paulo, Brazil.

Michael J. Huber (center) of Surface Combustion with Roberto Novak (left) and Jose Fernandes of Sauder Equipamentos.

500th Machine

The Gleason Works will ship is 500th Phoenix® CNC machine for the production of precision gears in August. The Phoenix, introduced in 1988, is the world's first spiral bevel and hypoid gear production machine requiring just six axes of CNC-controlled motion to produce all bevel gear tooth geometries for cutting and grinding. Since its introduction, the line has been expanded to include a helical hobber and gear testing and gear lapping machines.

Materials Guarantee

Hoeganaes Corporation of Riverton, NJ has announced the most comprehensive guarantee for ferrous metal powders ever offered in the P/M industry. The guarantee covers all the company's Ancorbond® and Anchodense™ processesed materials and goes into effect immediately. In the guarantee, Hoeganaes assures the performance properties of the materials agreed upon with the fabricator and that these materials will work with the specific applications for which they were developed. The company will provide online support, complete laboratory services and on-site personnel at the fabricator's facility to run development trials.

New Gage System

Albion Devices, Inc., of Solana Beach, CA, an engineering firm specializing in temperature compensation of precision measurements, recently delivered a GageComp-M2 multichannel temperature compensation system to K. J. Law Engineers, Inc. The unit was installed on an automatic gage developed by Law for American Axle & Manufacturing, Inc., to measure brake drum and pilot bores. The Albion system uses sensors to monitor the temperature of the workpiece, master and gage fixture.

Brakedrum in new Albion gage.

Expanded Facilities

Romer, Inc., manufacturer of portable CMMs, has expanded its Dearborn, MI, operations with a new, larger sales/service/training facility. The 4,300 sq. ft., multi-level facility houses a Romer sales office for Michigan, Ohio and Indiana, plus a full service center for Romer portable inspection systems. The company has also added two more applications engineers to its staff to provide customer service support for portable CMM customers. In addition, Romer has contracted with Crescent Gage of Dallas, TX, to handle sales of Romer portable articulate arm CMMs and related hardware and software in Texas, Louisiana and Oklahoma. ○

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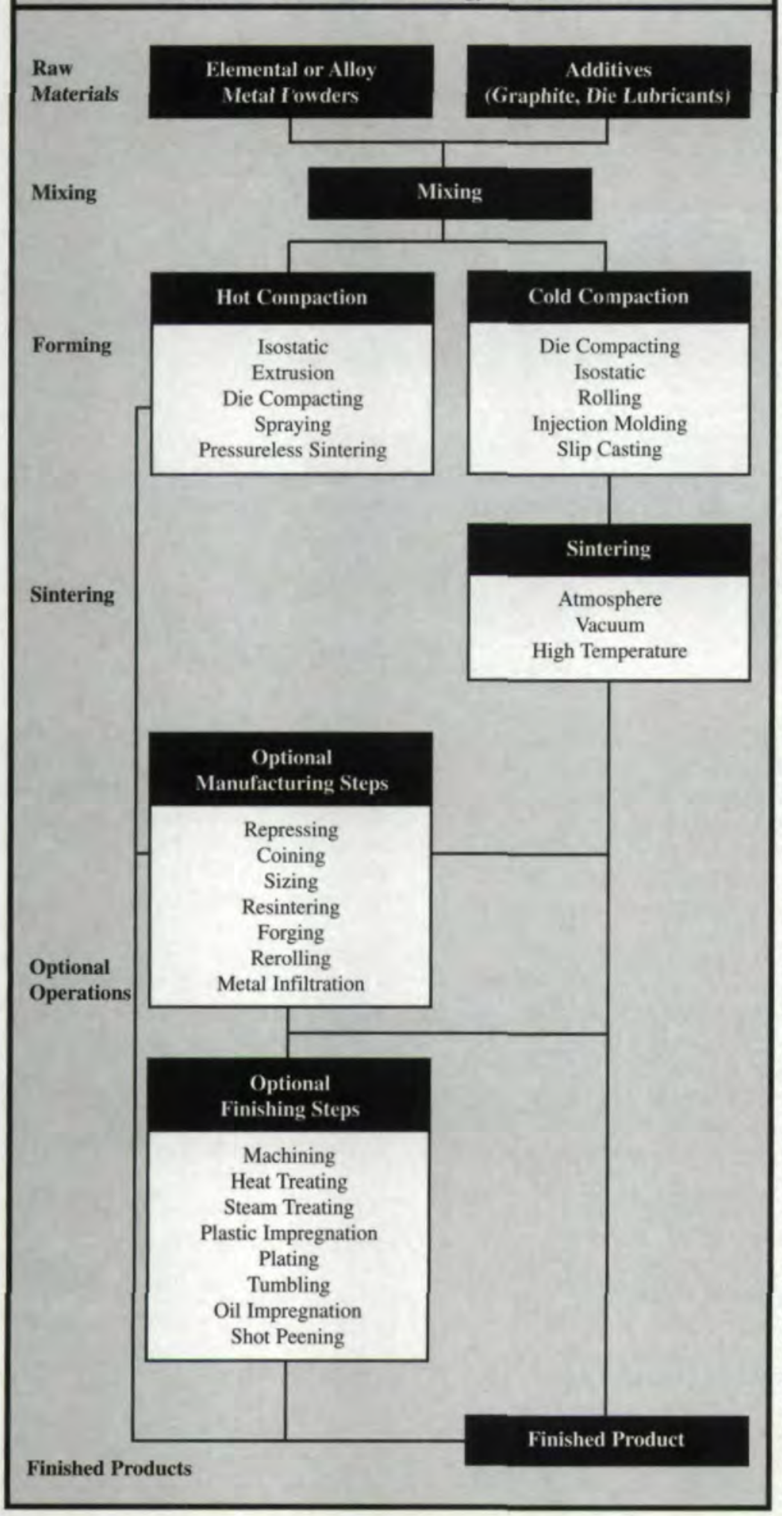
Powder Metal Gear Design and Inspection

Howard I. Sanderow

Table 1 — The Powder Metallurgy (P/M) Process



Powder metallurgy (P/M) is a precision metal forming technology for the manufacture of parts to net or near-net shape, and it is particularly well-suited to the production of gears. Spur, bevel and helical gears all may be made by powder metallurgy processing.

The P/M process is illustrated schematically in Table 1 (Ref. 1). There are three basic steps to producing parts: mixing, compacting and sintering. Variations to these basic steps, such as infiltration, double pressing/double sintering and powder forging, may be used to increase mechanical properties. Alternatively, a machining step may be added to qualify critical dimensions or to achieve a geometric feature not possible during rigid die compaction. Powder metal parts may be through-hardened or surface-hardened as required by the intended application.

In the pressing cycle, a charge of mixed powder is delivered to the die cavity by a feed shoe, and the upper and lower punches are used to compact the powder. After the upper punch is withdrawn, the pressed compact is ejected by the lower punch, and the feed shoe slides the part away from the die cavity. The cycle repeats as the feed shoe continues forward and refills the die cavity with another charge of powder.

Among the advantages of making gears by this process are that true involute gear forms are possible and that special features such as keyways, drive lugs, splines and cam contours may be incorporated during the compaction process. Lightening holes may be added to reduce part mass, and P/M gears can be made with blind corners, eliminating the undercut relief that is needed with cut gears (Ref. 2). P/M tooling provides consistent tooth form accuracy and surface finish over long production runs. Generally, because of their porosity, good surface finish and constant tool form accuracy, P/M gears reduce noise levels.

P/M gears can be made in a wide range of gear geometries for a number of applications, including gear motors for appliances, tractor transmissions, geared drives for cranes and a number of automotive applications.

Limitations of P/M Gears

Compared to wrought steel gears, P/M gears have lower impact resistance, fatigue strength and contact stress capability. This reduction in mechanical properties (due to the presence of porosity in the microstructure) may limit the same-size replacement of wrought and cast parts by P/M gears. However, it should be noted that the double-press/double-sinter process or warm compaction can overcome some of these limitations. Higher temperature sintering or case hardening can also improve the performance of P/M gears. Bending stress limitations may be overcome by using the versatility of P/M gear technology in the fillet radius design. If the design space permits, a larger P/M gear may be substituted.

For certain gear geometries, such as bevel and helical, the tooling motion cannot provide as high a tooth density as is possible in spur teeth. A portion of the applied pressure is lost to frictional effects in helical gears. Insufficient powder motion causes lower density in bevel gear teeth. Frequently copper infiltration is used to increase the density (and the corresponding mechanical properties) for both helical and bevel gears.

The face width of P/M gears is also limited. The constraints of the compaction process limit face widths to well under 3". Frictional losses between the powder and the die cause decreasing density along the face width, the lowest being at the midpoint. This is a reciprocal relationship, with greater face widths having larger density decreases. Dimensional variations can also occur during sintering and heat treating, which may lead to distortion, especially with larger gears.

Gear Design Data

P/M gear design data requirements are quite similar to those for machined gears. The AGMA Powder Metallurgy Committee has been working on the development of a new standard, "Powder Metallurgy Gear Specifications," for the past two years. This specification is expected to include sections for basic data, inspection data, calculation and process test data, as well as reference data.

Even after specifying the gear data in detail, certain geometric features or special notes should also be considered at the design stage:

1. Denoting the bore or central datum feature.

2. Providing a squareness of bore tolerance note if functionally critical.

3. Consideration of a tooling design to accommodate adherent burrs on the teeth edges.

a. Chamfers on teeth

b. A raised boss on the gear face to act as a 0.010" spacer between gears so small burrs do not interfere with proper rotation of the gears.

4. An allowance for minor surface defects, such as nicks or minor amounts of raised metal from a vibratory finishing operation.

5. Including an allowance for missing material (if possible).

6. An allowance for an identification mark on a gear face to assist in tool orientation during compaction and gear inspection after processing.

Proper consideration of these items during preliminary design discussions, either as specific design details on the part print or as off-line quality planning agreements, will ensure the P/M gear is specified correctly.

Mechanical Design Criteria

Once the geometric features of a P/M gear are determined, the next step is evaluating the mechanical loads on the product. Typically these requirements will include a normal operating load and a potential overload condition. Two types of failure modes should be considered:

- Tooth bending fatigue failure, and
- Tooth overload failure.

The AGMA Powder Metallurgy Committee is also reviewing methods of evaluating the mechanical operating characteristics of P/M gears. One of the goals of the committee is to provide a simple equation for determining the load capacity of P/M gears (spur, helical and bevel) once certain mechanical properties of the P/M steel materials are known. For instance, an equation of the form shown below has been suggested.

$$T = \frac{S\,K_1\,d\,F\,J}{2\,Pd\,K_2}$$

Where

T = torque load capacity (in-lbf)
S = design strength (lbf-in^2)
K_1 = constant
d = calculation diameter (in)
F = effective face width (in)
J = geometry factor
Pd = diametral pitch (in^{-1})
K_2 = constant

The design strength, S, would be the fatigue strength when determining the torque capacity under repeated loading or the yield strength when determining the torque capacity under a sudden, overloading condition. Once the material property, S, has been determined and the gear geometry is known, the torque load capacity, T, can be calculated. This value is then compared to the actual torque load expected by the gear. If the calculated T is less than the actual load, a failure is predicted; if T is greater than the actual torque load, then an acceptable material has been selected for the design conditions.

Howard I. Sanderow
is the president of Management & Engineering Technologies, Dayton, OH. He has written a number of articles and presented seminars on powder metal subjects.

Table 2 — Typical Manufacturing Tolerances for Heat Treated P/M Steel Gears

P/M Gear Quality Manufacture Process					
Gear Size Pitch Diameter mm	Max Runout µm	Tooth-to-Tooth Error µm	Total Composite Error µm	AGMA Class (Approx.)	Secondary Operation
16	35	40	75	7	None
	25	30	55	8	Size
	20	20	40	9	Bore
40	45	40	85	7	None
	35	25	60	8	Size
	25	20	45	9	Bore
100	75	55	130	6	None
	60	40	100	7	Size
	45	40	100	7	Bore

Values for DP = 32 Module = 0.75
Finer pitch requires closer tolerances to meet same AGMA class.

A second type of mechanical limit or failure mode encountered in gear designs is wear or pitting fatigue failure. This type of failure, also known as contact fatigue or surface fatigue, is normally associated with highly stressed, heat treated steel gears. Only recently has experimental work begun toward a systematic evaluation of the surface fatigue phenomenon as pertaining to P/M steel alloys (Ref. 3).

Until the AGMA Powder Metallurgy Committee completes its work in the area of mechanical performance characteristics, P/M gear designers will be forced to utilize other approximations for determining load capacity. One of the early studies of P/M gears (Ref. 4) suggested using the gear rating formula (as now found in AGMA 2001-B88), but modified for P/M steels. An approximation for the bending fatigue strength was to use 30% of the tensile strength. The approximation for the allowable contact stress was the tensile strength minus 10,000 psi.

These estimates are slightly conservative in light of more recent studies, which find the mean fatigue life closer to 32% of the tensile strength for heat treated P/M steels (Ref. 5), and the contact fatigue strength for a high density, heat treated nickel steel (FN-0205-180HT) about 10,000 psi greater than the tensile strength (Ref. 3). A recent Japanese study (Ref. 6) evaluated actual P/M gears for tooth bending fatigue. Contact fatigue limits were developed from a sliding roller test rig. Gear running tests were also conducted using a power circulating testing machine for comparison with these fatigue tests. Their results indicate improvement in bending fatigue strength through

- increased density,
- increased sintering temperature,
- shot peening after case hardening,
- heterogeneous microstructure.

The contact fatigue limit of about 190,000 psi for the 4% nickel steel in the Hirata study was lower than the value reported by Prucher et al. due to the lower density (7.1 vs. 7.4 g/cm^3) and difference in heat treatment (case-hardened vs. through-hardened). These investigators also noted that contact fatigue (spalling) was the predominant failure mode in the gear running tests, but that the contact fatigue life was increased 15–20% compared to the roller RCF tests. They concluded that the load bearing capacity of case-hardened P/M steel gears was higher than that of Tufftrided® AISI 1045 wrought steel gears.

P/M Gear Inspection

The quality of P/M gears is determined using the same type of inspection equipment as is used for machined gears:

- A rolling or composite gear checker for fine pitch gears,
- An involute or element gear checker for coarser pitch gears.

Typical manufacturing tolerances that can be expected for heat treated P/M steel gears are summarized in Table 2. As the size of the gear increases, the tolerance limits must also increase, since size changes in sintering and heat treatment are direct percentages of lineal dimensions. For small-to-medium size gears, an AGMA Q7 gear is quite feasible with no extra processing. A sizing or boring operation is needed to reach Q8 or Q9 tolerances. For much larger gears (4–6" pitch diameter), the as-manufactured tolerances are greater, resulting in a Q6 gear without any secondary operation.

Mechanical testing of a statistically representative sample from each production lot is used to demonstrate the mechanical quality of the product. Test methods include tooth torque tests or tooth breaking strength tests (either static or impact loading).

Advances to Further Improve P/M Gears

Efforts are already underway at specific P/M gear manufacturers, as well as at powder producers and process equipment suppliers, to improve both the dimensional control and the mechanical properties of P/M gears.

Dimensional Control. The dimensional consistency of gears can be improved either in the sintered condition or through the heat treatment process. One method, which is said to create AGMA Q9 gears from a Q6 gear, (Refs. 7–8) is the surface rolling of a sintered P/M gear against a master gear. Sinter-hardening P/M gears can reduce the distortion associated with more conventional quench and temper heat treatment processes.

Tooth Density Increase. Four primary methods to increase P/M gear tooth density are under study.

1. *Roll Densification*. This effective dimensional control method also densifies the gear tooth surface. Improvement in bending fatigue strength of 32% and a 3.5 times increase in contact fatigue stress has been reported for case-hardened 4600-type P/M steel gears subjected to this process (Takeya et al.). Contact fatigue strength of 96% of that of case-hardened wrought AISI 4118 steel has also been reported.

2. *Warm Compaction*. This process is much the same as conventional powder metal part compaction, except that both the metal powder and the tooling are heated to approximately 300°F before processing. In preliminary tests, gears manufactured using warm compaction showed an increase of 30% in tooth break load. When the process was coupled with high-temperature sintering, the improvement was over 50% (Ref. 9).

3. *Rotopressing*. The Rotopressing process, which subjects parts to large tangential stresses, causing intense, local plastic flow and densification, has been reported to give densities greater than 7.6 g/cm^3 in sintered P/M gear teeth, leading to excellent fatigue and wear properties (Ref. 10).

4. *Ausrolling*. This process combines surface densification and heat treatment (Ref. 11). Ausrolling of a conventional P/M steel is reported to have reduced the surface porosity from 14% to less than 2%, increased the rolling contact fatigue endurance by more than 10 times and led to substantial improvements in gear accuracy and surface finish (Ref. 12).

Conclusions

The powder metallurgy process can provide gear designers with a cost-effective alternative to machined, wrought steel gears. This net shape process offers dimensional tolerances and mechanical properties compatible with many market applications. As current process improvements and material developments are incorporated, more high performance transmission gears will be converted to powder metallurgy. ○

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Acknowledgements: *Longer versions of this material were presented at the 1995 AGMA Fall Technical Meeting and at the 1996 SME 1st International Advanced Gear Processing & Manufacturing Conference. They are available from AGMA as 95FTM13 and from SME as "Powder Metallurgy Gears—Expanding Opportunities" by James & Sanderow.*

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AGMA & MPIF Develop Standards, Information Sheet for Powder Metal Gears

AGMA and members of the Metal Powder Industries Federation (MPIF) are three years into a joint project to develop specifications and an information sheet on rating powder metal gears. According to committee vice chairman Glen A. Moore of Burgess-Norton Mfg. Co., the first phase of the project, the publication of AGMA Standard "6008-AXX, Specifications for Powder Metallurgy Gears," should be completed in late 1996 or early 1997.

The object of the standard is to spell out through descriptions and examples the minimum data needed to adequately specify P/M gears. It covers external spur, external helical and straight bevel gears, the types most often made by the conventional P/M process. The subjects covered include gear tooth geometry data, gear drawing specifications and gear material specifications.

Moore points out that 6008-AXX is still a work in progress. The draft is almost complete, but has yet to go through the formal AGMA process of review and approval.

Courtesy of Burgess-Norton Mfg. Co.

P/M gears are available for a wide varity of applications.

The standard addresses the long-standing powder metal industry concern about the lack of information available to gear purchasers to properly convey their gear design intent to the gear producer. The lack of properly detailed gear data often leads to confusion and delays in quotations and deliveries and in manufacturing gears for particular applications.

"This standard addresses a long-term need," says Moore. "As powder metal gears appear in more and more applications, it's important that this information becomes available."

Powder metal part producers like Burgess-Norton welcome this joint effort, which has produced the first AGMA standard for powder metal gears. They feel it constitutes recognition of the fact that P/M is a viable alternative to wrought metal in some applications. Tables 1 and 2 are samples of this data.

The Bending Load Rating Information Sheet

The second part of this joint AGMA/MPIF project is an information sheet, "9XX-AXX, Calculated Load Capacity of P/M Gears," which will contain rating procedures and data on the failure of powder metal gears caused by bending loads, either as fatigue failures brought on by repeated loading or as breakage or yielding caused by a temporary overload condition.

The wear and pitting modes of failure have been omitted from the information sheet for a number of reasons. AGMA standards for wrought metal gears do not cover the wear mode of failure because it is successfully avoided in most cases by proper lubrication. Although the wear mode is common enough in P/M gears where adequate lubrication is not practical, the limited data available and the complexity of the issues involved forced the decision to follow the present practice, according to Moore.

Inadequate test data for pitting in P/M gears made eliminating it from the information sheet necessary as well. Furthermore, the committee felt that omitting pitting data would not be a significant handicap to the usefulness of the information sheet because this mode of failure is

Table 1 — Basic Data Required to Describe Gear in General Terms

	TYPE OF GEAR		
SPECIFICATION	**SPUR**	**HELICAL**	**STRAIGHT BEVEL**
Number of Teeth	X	X	X
Diametral Pitch	X		X
Normal Diametral Pitch		X	
Pressure Angle	X		X
Normal Pressure Angle		X	
Helix Angle		X[1]	
Hand of Helix		X	
Pitch Angle			X
Face Angle[2]			X
Root Angle[2]			X
Back Angle[2,3]			X
Tooth Form	X[4]	X[4]	X

Notes:
1) Value of helix angle when calculated from other data specifications may be approximate due to rounding.
2) These data items may be shown instead on an outline drawing of the gear blank.
3) The outer form of P/M gears is generally cylindrical, in which case the back angle is zero degrees and this specification item may be omitted.
4) Tooth form may be omitted when the tooth outline is fully defined by other data.

Table 2 — Inspection Data (For Use With Composite Action Testing) Appropriate For Each Gear Type

	FILLET FORM		
SPECIFICATION	**SPUR (EXTERNAL)**	**HELICAL (EXTERNAL)**	**STRAIGHT BEVEL**
Specification	X[2]	X[2]	X[2]
Outside Diameter	X[3]	X[3]	
Root Diameter	X	X	X
Master Gear (Spec or Data)			X
Pitch Apex to Back	X	X	X[4]
Total Composite Tolerance	X	X	X[4]
Tooth-to-Tooth Composite Tolerance	X[2]	X[2]	
Test Radius	X[2]	X[2]	X[2]
Tip Radius	X		

Notes:
1) Value of helix angle when calculated from other data specifications may be approximate due to rounding.
2) These data items may be shown instead on an outline drawing of the gear blank.
3) The outer form of P/M gears is generally cylindrical, in which case the back angle is zero degrees and this specification item may be omitted.
4) Tooth form may be omitted when the tooth outline is fully defined by other data.

much less common than the tooth bending mode in P/M gears.

According to Moore, efforts are being made now to collect pitting data, and when it becomes available, to include the pitting mode in the information sheet will be reconsidered.

Developing the data for the information sheet will be a long process, says Moore. "It took three years to develop the specification, and the information sheet will probably take as long," he says.

The committee will take the existing AGMA formulas and plug in appropriate factors for P/M to make similar gears. Then test gears will have to be manufactured and tested to compare the actual results with the theoretical ones. Drafts of the procedures will have to be written and discussed and their calculations compared with available field experience with gears in real applications. Only at that point will the draft be approved and circulated as an information sheet.

Eventually, as field experience increases, the information sheet could go through other revisions and be upgraded as a rating standard.

The next meeting of the AGMA/MPIF committee will be Sept. 16–17 at AGMA headquarters in Alexandria, VA. Information about participation and other committee matters is available from Mr. Bill Bradley at AGMA.

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OHIO STATE GEAR NOISE COURSE

A three-day course on gear noise, covering measurement, analysis, diagnosis, noise reduction and dynamics, will be held at The Ohio State University **Sept. 10–12**. Contact Carol J. Bird at 614-292-3204 or fax 614-292-3163 for more information.

ASM EDUCATIONAL SEMINARS

ASM is offering the following seminars on metallurgical subjects: **Sept. 17–19**, Hardness Testing: Methods & Applications; **Sept. 23–27**, Principles of Failure Analysis; **Oct. 7–11**, Principles of Heat Treating; **Oct. 14–18**, Metallurgy for the Non-Metallurgist. For registration information, contact ASM Member Services at 216-338-5151, fax 216-338-4634 or e-mail *mem-serv@po.asm-intl.org*.

AGMA GEAR SCHOOL

One-week sessions on basic gear manufacturing. Both classroom and hands-on work covering hobbing, shaping and inspection. Held at Daley College, Chicago, IL. The next session is **Sept. 23–27**. Call AGMA at 703-684-0211 or fax 703-684-0242 for more information.

ASME 7TH INTERNATIONAL POWER TRANSMISSION & GEARING CONFERENCE

In San Diego, CA, **Oct. 6–9**, this four-day conference will focus on major technical aspects of power transmission component design, manufacturing and applications. Subjects will include gear design, wear, lubrication, noise and vibration, as well as belts, bearings, couplings, chain and helicopter transmissions. Contact ASME at 212-705-7057 for registration information.

AMERICAN PFAUTER/PFAUTER-MAAG CLINIC

The companies are offering a 3-day seminar on **Oct. 8–10** at their facilities in Loves Park (Rockford), IL. The subject is modern metal removal and measuring techniques for spur and helical gears. Designed for manufacturing and process management personnel, the clinic will provide an in-depth look at modern methods, applications and hardware. For more information, contact Lisa Alexander, 815-282-3000, x328 or fax 815-232-3075.

AISE LUBRICATION SEMINAR

AISE (Association of Iron and Steel Engineers) is holding a Basic Lubrication Seminar **Oct. 20–22** at the Sheraton Station Square Hotel, Pittsburgh, PA. Subjects covered include basic concepts, test methods, bearings, gears, conservation, hydraulics, additives and more. Contact Laura Evert at 412-281-6323 x151 for more information.

AGMA 1996 FALL TECHNICAL MEETING

This year's meeting will be **Oct. 28–30** in Cincinnati, OH, and will cover topics including bevel gear manufacturing, sound and vibration, scuffing, fatigue life testing and more. To register, contact AGMA Headquarters at 703-684-0211 or fax 703-684-0242.

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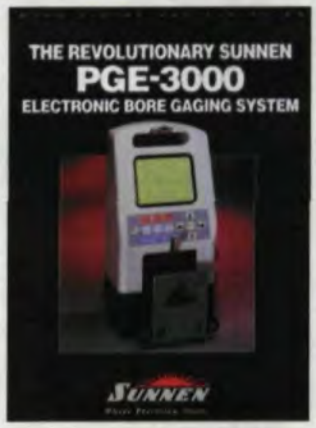

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CIRCLE 188

Gear Wear Caused By Contaminated Oils

Douglas Godfrey

Introduction

The diagnosis and prevention of gear tooth and bearing wear requires the discovery and understanding of the particular mechanism of wear, which in turn indicates the best method of prevention. Because a gearbox is a tribologically dependent mechanism, some understanding of gear and bearing tribology is essential for this process. ***Tribology*** is the general term for the study and practice of lubrication, friction and wear. If tribology is neglected or considered insignificant, poor reliability and short life will result.

One of the most common causes of practical industrial wear problems is contaminants in the oil. In this article the author shares his experience with the diagnosis and solution of such problems and their relationship to basic tribology.

Kinds of Lubrication Occurring in a Gearbox

All types of gears and their support bearings utilize the three kinds of lubrication.

Hydrodynamic Lubrication (HDL) exists when a gas or liquid film completely separates moving surfaces, preventing solid-to-solid contact. HDL is the most desirable regime of lubrication because friction is low and wear can be extremely low. In gears it occurs wherever the sliding of engaging teeth, especially just above or below the pitch line, allows for the formation of a wedge of oil that can completely separate the teeth. Journal bearings in the gearbox also operate in the HDL regime.

The primary HDL property of the oil is its viscosity. The oil film thickness must be greater than the sum of the surface roughnesses to minimize wear. A rule of thumb to apply here is that the oil film thickness is proportional to the square root of the viscosity. For example, if gear oil viscosity is increased from ISO 150 to ISO 320, film thickness will be increased 46% for a given load and sliding speed. It is important to note that in spur gears, HDL is discouraged by the automatic reversal of sliding direction on the addendum and dedendum.

Elastohydrodynamic Lubrication (EHL) implies that a full oil film is formed between moving surfaces that are elastically deformed. It occurs in concentrated contacts, such as rolling contact at the pitch line of the gears. The metals at the line of contact deform elastically, and the oil trapped between them is subjected to extremely high Hertzian pressures. The oil viscosity increases with high pressures by many orders of magnitude. For example, a mineral oil with a viscosity of 15 cSt at atmosphere pressure would be 100,000 cSt at 140,000 psi. The oil may be considered a solid for the short time it is at that pressure. The pressure-viscosity coefficent is a measure of change.

As a result of combined elastic deformation and high viscosity, an extremely thin film of oil completely separates the surfaces and prevents metal-to-metal contact. EHL also occurs between the rolling elements of rolling element bearings and their raceways. For EHL, oil viscosity at atmospheric pressures is still important for dragging the oil into the contact.

The EHL theory was developed by English tribologists Dowsen and Higgensen in order to understand zero tooth wear on a 30-year-old operating gearbox. They developed an equation for the calculation of minimum oil film thickness of engaging gear teeth (See Table 1).

Boundary Lubrication (BL) occurs when HDL and EHL fail, and metal-to-metal contact occurs, such as when the tip of a spur gear tooth slides near the root of the opposing tooth. BL also occurs during

Table 1 — Equation for the Calculation of Minimum Film Thickness in the Elastohydrodynamic Lubrication Regime. *

$$h_{min} = \frac{1.63\alpha^{0.54}(\mu_o V_e)^{0.7} \cdot \rho n^{0.43}}{(X_r W_{Nr})^{0.3} Er^{0.13}}$$

The specific film thickness is given by

$$\lambda = \frac{h_{min}}{\sigma}$$

where

σ = composite surface roughness

$\sigma = (\sigma_1^2 + \sigma_2^2)^{1/2}$

σ_1, σ_2 = surface roughness, rms (pinion, gear)

μ_o = absolute viscosity, Reyns (lb sec/in^2)

α = pressure-viscosity coefficient, (in^2/lb). The pressure-viscosity coefficient ranges from α=0.5 x 10^{-4} to α=2 x 10^{-4} for typical gear lubricants.

Er = reduced modulus of elasticity given by

$$Er = 2\left(\frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}\right)^{-1}$$

where

υ_1, υ_2 = Poisson's ratio (pinion, gear)

E_1, E_2 = modulus of elasticity (pinion, gear)

ρn = normal relative radius of curvature

$$\rho n = \frac{\rho_1 \rho_2}{(\rho_2 \pm \rho_1)\cos\Psi_b}$$

ρ_1, ρ_2 = transverse radius of curvature (pinon, gear)

Ψ_b = base helix angle

V_e = entraining velocity given by

$V_e = Vr_1 + Vr_2$

where

Vr_1, Vr_2 = rolling velocities given by

$Vr_1 = \omega_1 \rho_1$

$Vr_2 = \omega_2 \rho_2$

ω_1, ω_2 = angular velocities (pinion, gear)

W_{Nr} = normal unit load given by

$$W_{Nr} = \frac{W_{Nr}}{L_{min}}$$

where

W_{Nr} = normal operating load

L_{min} = minimum contact length

* These equations are taken from Reference 2, Errichello, "Lubrication of Gears."

start-up, shut down, at high loads and high temperatures and in worm gears. The important oil property for BL is the soluble anti-wear and anti-scuff additives in the oil, which react with the metal surfaces to form a thin, tough film that reduces metal-to-metal contact.

In worm gears, which resemble a screw thread with continuous sliding, lubricity additives are required to reduce friction. Lubricity is defined as that property of a lubricant that reduces friction. Oil with good lubricity also improves gearbox efficiency.

Mixed Lubrication. HDL, EHL and BL may not occur ideally and isolated in gears, but in a mixed condition. During one tooth engagement all three lubrication mechanisms may occur consecutively or simultaneously. Unfortunately, ideal lubrication mechanisms frequently fail and allow high wear.

Many articles such as Refs. 1 and 2 provide information on gear lubrication.

Kinds of Gear Wear

The AGMA booklet (Ref. 3) and other literature describe the numerous kinds of gear tooth failures. Table 2 shows a list of the most common wear mechanisms related to contaminated lubricants.

Table 2 — Common Kinds of Gear and Bearing Wear

Gears		
	Abrasion	
	Scuffing	
	Contact Fatigue and Spalling	
	Polishing Wear	
	Pitting	
Bearings		
	Rolling Element	
		Nicks and Dents
		Contact Fatigue
		Abrasion
		Fretting Corrosion
	Journal	
		Wiping of Babbitt
		Excessive Embedment

Wear Caused by Contaminated Gear Oil

In the author's experience, contamination of the lubricant accounts for about 80% of gear tooth wear and bearing distress. *Therefore for reliability and long life, keep the oil clean and dry.* Contaminants in lubricated machines may come from the fresh oil or the new machine, be generated by the system while running or ingested from external sources. These issues are discussed in detail in Ref. 4.

Following are the major kinds of oil contamination and the types of gear tooth and bearing wear they cause.

- Ingested dirt
  - Any solids interfere with oil film formation.
  - Hard particles cause abrasion.
  - Fine particles cause polishing wear.
  - Dirt makes dents in rolling element bearings, leading to fatigue failure.
- Water
  - Interferes with oil film formation.
  - Causes rust.
  - Degrades the oil by acting as a catalyst for oxidation.
- Manufacturing debris
  - Same as ingested dirt, plus metal chips bridge the oil film and initiate scuffing.
- Chemicals
  - Cause corrosion and oil degradation.
- Wear debris
  - The accumulation of gear wear debris promotes oil degradation.

- Wear debris embeds in babbitt bearings and destroys the surface.
- Wrong oil
  - If viscosity is too low, oil film is too thin.
  - If viscosity is too high, gearbox efficiency decreases.
  - If additives are too chemically active, corrosion and polishing wear result.
  - If additives are not chemically active enough, the risk of scuffing increases.

Abrasion

Hard particles in the oil are the most frequent cause of gear tooth wear. Abrasion is the cutting of metal by hard particles or a rough surface. Abrasion is readily recognized by parallel furrows in the direction of sliding. If the abrasion is caused by hard particles, such as sand or grinding grit in the oil, it is called three-body abrasion. Prevention is obvious—remove the hard particles. If abrasion is caused by one tooth cutting the opposing one, such as when a hard, sharp gear tooth tip slides against a softer root, two-body abrasion occurs. The result is the same as filing. Smoother surfaces and chamfered tips will prevent this type of abrasion.

Rolling element support bearings suffer when solid particles and water contaminate the oil. The hard particles cause nicks and dents on the raceway and the rolling elements and abrade the sliding areas, such as between the cage and the rolling element bearings. The dents shorten life and start contact fatigue failure. Water in the oil accelerates crack growth, thus shortening fatigue life. Ideally, the rolling element bearings would be less stressed if they had their own clean and separate oil supply and didn't have to run on the oil contaminated with gear wear particles.

Other well-known tribology problems are caused by improper metallurgy, excessive loads, speeds and temperatures and misalignment.

Detection of Contaminated Gear Oil

The degree of contamination and the type of contaminants are usually determined by an oil analysis laboratory. However, a rough idea of contamination can be obtained on site using a simple process.

Collect some oil in a clean, tall, narrow glass bottle, (a 2-oz. olive bottle is about right). Look at the oil with a bright light. If it is hazy, it is probably contaminated with water. Smell the oil. If it has a bad odor compared to fresh oil, it is probably oxidized. Now set the bottle aside for at least two days and allow the dense contaminants to settle; then examine the material in the bottom. Metal wear fragments will be present and if ferrous, they will move with a magnet. Water will be a clear layer; sand and clay will also be on the bottom, but will not move with a magnet. To get a better look, carefully pour off most of the oil above the contaminants, add clean paint thinner, shake up and allow the contaminants to settle again. *Any* visible magnetic fragments, sand or water is bad news.

Reducing Contamination

Contamination can be reduced and oil cleaned by following some simple procedures.

- Change used oil frequently.
- Filter oil with a circulating system with a full-flow, 3-micron filter and reduce ingestion of dirt and water by means of a fine air filter on the vent or breather tube.
- Before operating a new gearbox, clean the gears and gearboxes of all manufacturing debris by vigorous flushing. Grinding and cutting chips, welding splatter, sand from sand blasting, steel shot,

Table 4 — Common Wear Problems Related to Lubricants & Hydraulic Fluids*

Names of Wear	Other	Definition	Susceptible Machine Parts	Conditions Promoting Wear	Unaided Eye
Mild Adhesion	Normal[a]	Generally, transference of material from one surface to another due to adhesion and subsequent loosening during relative motion. Mild adhesion involves transfer and loosening of surface films only.	All.	• Moderate loads, speeds & temperatures. • Good, clean, dry lubricants. • Proper surface finish.	• Low rates of wear. • No damage. • Deeper original grinding marks still visible.
Severe Adhesion	• Scuffing • Galling • Scoring[b]	Cold welding of metal surfaces due to intimate metal-to-metal contact.	• Piston rings and cylinder barrels. • Valve train. • Rolling & sliding bearings. • Gears. • Cutting tools. • Metal seals. • Chains.	• High loads, speeds and/or temperatures. • Use of stainless steels or aluminum. • Insufficient lubricant. • Lack of antiscuff additives. • No break in. • Abrasive wear.	• Rough, torn, melted or plastically deformed metal, broad or streaks. • High temperature oxidation. • High friction, high rates of wear. • Possible seizure.
Abrasion[e]	• Cutting • Scratching • "Wire Wool" damage • Gouging • Scoring	Cutting and deformation of material by hard particles (3-body) or hard protuberances (2-body).	All surfaces in relative motion.	• Hard particles contaminating oil. • Insufficient metal hardness. • Hard metal with rough surface against soft metal.	• Scratches or parallel furrows in the direction of motion, similar to "sanding." • High rates of wear.
Erosion	• Solid Particles • Impact Erosion	Cutting of materials by hard particles in a high-velocity fluid impinging on a surface.	• Journal bearings near oil holes. • Valves. • Nozzles.	• High-velocity gas or liquid containing solids impinging on a surface[g].	• Smooth, broad grooves in direction of fluid flow. • Matte texture, clean metal. • Similar to sandblasting.
Polishing	Bore Polishing	Continuous removal of surface films by very fine abrasives.	• Cylinder bores of diesel engines. • Gear teeth. • Valve lifters.	• Combination of corrosive liquid and fine abrasive in oil[h].	• High wear, but a bright mirror finish. • Wavy profile.
Contact Fatigue	• Fatigue Wear • Frosting • Surface Fatigue • Spalling	Metal removal by cracking and pitting due to cyclic elastic stress during rolling and sliding.	• Rolling & sliding bearings. • Valve train parts. • Gears.	• Cyclic stress over long periods. • Water, dirt in oil. • Inclusions in steel.	Cracks, pits and spalls.
Corrosion	• Chemical Wear • Oxidative Wear • Corrosive Film Wear	Rubbing off of corrosion products on a surface.	• All bearings. • Cylinder walls. • Valve train. • Gears. • Seals and chains.	• Corrosive environment. • Corrodible metals. • Rust-promoting conditions[j]. • High temperatures.	Corroded metal surface.
Fretting Corrosion	• False Brinelling • Fretting • Friction Oxidation	Wear between two solid surfaces experiencing oscillatory relative motion of low amplitude.	• Vibrating machines. • Bearing housing contacts. • Splines, keys, couplings. • Fasteners.	Vibration-causing relative motion.	• Corroded, stained surfaces[k]. • Loose colored debris around real contact areas. • Rouge (Fe_2O_3) colored films, debris, grease or oil for steel.
Electrocorrosion	• "Erosion" • Electrical Erosion • Electrochemical Wear • Electrical Attack	Dissolution of a metal in an electrically conductive liquid by low amperage currents.	• Aircraft hydraulic valves. • Hydraulic pumps and motors.	• High-velocity liquid flow causing streaming potentials. • Stray currents. • Galvanic metal combinations.	• Local corroded areas. • Black spots such as made by a small drop of acid. • Corroded, worn metering edges.
Electrical Discharge	• Electrical Pitting • Sparking	Removal of metal by high amperage electrical discharge or spark between two surfaces.	• Bearings in high-speed rotating machinery, such as compressors, atomizers. • Static charge producers.	• High-speed rotation. • High-velocity, two-phase fluid mixtures. • High potential contacts. • Sparks.	Metal surface appears etched. In thrust bearings, sparks make tracks like an electrical engraver.
Cavitation Damage[m]	• Cavitation Erosion • Fluid Erosion	Removal of metal by bubble implosion in a cavitating liquid.	• Hydraulic parts, pumps, valves, gear teeth. • Cylinder liners, piston rings. • Sliding bearings.	Sudden changes in liquid pressure due to changes in liquid velocity or to shape or motion of parts[n].	• Clean frosted or rough-appearing metal. • Deep rough pits or grooves.

* From "Recognition and Solution of Some Common Wear Problems Related to Lubricants & Hydraulic Fluids," *Lubrication Engineering*, February, 1987. Reprinted with permission.

Symptoms Microscopically	Oil Analysis	Prevention Mechanical Changes	Lubricant Changes
• Smooth microplateaus among original grinding marks. • Slight coloration due to films.	• 1–5 ppm wear metals by emission spectroscopy. • Low % solids by filtration. • Metal salts (oxides, sulfides, phosphates, etc.) in wear fragments by X-ray diffraction.	None.	None.
• Rough irregular surface. • *Metal from one surface* adhering to other surface by spot tests or microprobe analysis.	Large metallic wear fragments of *irregular shape*[c].	• Reduce load, speed and *temperature.* • Improve oil cooling. • Use compatible metals. • Apply surface coatings such as phosphating. • Modify surface, such as ion implantation[d].	• Use more viscous oil to *separate surfaces.* • Use "extreme pressure" additives, such as a sulfur-phosphorous or borate compound.
• Clean furrows, burrs, chips. • Embedded abrasive particles. • In sliding bearings with soft overlay, embedded particles cause polished rings.	• High metal content in oil and high silicon (>10 ppm) by emission spectroscopy. • High % solids by filtration. • Chips and burrs by ferrography.	• Remove abrasive by improved air and oil filtering, clean oil handling practices, improved seals, flushing and frequent oil changes[f]. • Increase hardness of metal surfaces.	• Use oil free of abrasive particles. • Use more viscous oil.
• Short V-shaped furrows by scanning electron microscopy. • Embedded hard particles.	Elements of hard particles by emission spectrography. Chips and burrs by ferrography.	• Same as above. • Reduce impact angle to less than 15°.	Same as above.
Featureless surface except scratches at high magnification by electron microscopy.	Combination of fine metal corrosion products and fine abrasive by X-ray diffraction.	None.	• Choose less chemically active additive. • Remove corrosive contaminant. • Remove abrasive.
• Combination of cracks and pits with sharp edges. • Subsurface cracks by metallographic cross section. Numerous metal inclusions.	• Particles of metal with sharp edges. • Metal spheres by electron microscopy.	• Reduce contact pressures and frequency of cyclic stress. • Use high quality vacuum melted steels. • Use less abusive surface finish.	• Use clean, dry oil. • Use more viscous oil. • Use oil with higher pressure viscosity coefficient[i].
Scale, film, pits containing corrosion products. Dissolution of one phase in a 2-phase alloy.	• Detection of corrosion products of worn metal. • Detection of anions, such as chlorine, by X-ray *fluorescence.*	• Use more corrosion resistant metal (not stainless). • Reduce operating *temperature.* • Eliminate corrosive material.	• Remove corrosive material such as too chemically active additives and contaminates. • *Use improved corrosion* inhibitor. • Use fresh oil.
Thick films of oxide of metal. Red and black for steel.	Indentify metal oxide ($\propto Fe_2O_3$ for steel) by X-ray diffraction.	• Reduce or stop vibration by tighter fit or higher load. • Improve lubrication between surfaces by rougher (then honed) surface finish.	• Use oil of lower viscosity. • Relubricate frequently. • Use oxidation inhibitors in oil.
Corrosion pits, films, dissolution of metals.	• Detection of corrosion products. • Electrically conductive liquids[l].	• Decrease liquid velocity and velocity gradients. • Use corrosion-resistant metals. • Eliminate stray currents. • Use nongalvanic couples.	Decrease or increase electrical conductivity of lubricants of hydraulic fluids.
• Pits near edge of damage, showing once molten state, such as smooth bottoms, rounded particles, gas holes. • Rounded particles near pits welded to surface.	Detection of large rounded particles by microscopic examination of filtrate or in ferrography.	• Improve electrical insulation of bearings. • Degauss magnetic rotating parts. • Install brushes on shaft. • Improve machine grounding.	Use oil of higher electrical conductivity.
• Clean, metallic bright rough metal, pits. • Removal of softer phase from 2-phase metal[o].	Observation of large chunks or spheres of metals in oils.	• Use hard tough metals, such as tool steel. • Reduce vibration, flow velocities and pressures. • Avoid restrictions and obstructions to liquid flow.	• Avoid low vapor pressure, aerated, wet oils. • Use noncorrosive oils.

grinding grit, such as silicon carbide, paint chips and fibers are all frequently found in *new* gearboxes.

• Clean and assemble gearboxes in a dedicated clean room, not a dusty shop.

• Filter fresh oil as it is added to a gearbox.

This kind of simple, basic practice should greatly reduce the risk of lubricant contamination and consequent gear wear and failure.

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Douglas Godfrey *is the principal in Wear Analysis, a tribology consulting firm in San Rafael, CA. He has over 50 years' experience in tribology research and study and is a well-known author and lecturer in the field.*

Notes for Table 4

a — Mild adhesion is a desirable wear condition.
b — Scoring is not recommended because it implies a scratch or furrow cut by abrasion.
c — Emission spectroscopy usually misses large (> 5 micron) wear fragments.
d — Increasing metal hardness does not reduce scuffing.
e — The most common wear problem.
f — Do not shot peen, bead or sandblast any surface in a lubricated machine because abrasive cannot readily be removed completely.
g — Sandblasting embeds sand in surfaces.
h — An example of polishing combination in oil is active sulfur additive and Fe_2O_3 (jeweler's rouge).
i — A new additive reduces promotion of contact fatigue by water; some extreme pressure additives are suspected of promoting contact fatigue.
j — Rust (hydrated iron oxide $Fe_2O_3H_2O$) is common corrosion product of ferrous metal.
k — Damage on one surface is mirror image of damage on other.
l — Highly compounded oils can be electrically conductive—or electrolytes; phosphate ester hydraulic fluids are conductive.
m — Not to be confused with pump cavitation, which is a different phenomenon.
n — Corrosion and abrasive in oil increase cavitation damage.
o — Graphite phase in cast iron susceptible to removal by cavitation.

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PRODUCT NEWS

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Tribology Tool

Computational Systems Inc., offers a new maintenance tool, the OilView 51DV Digital Viscometer. It provides on-site measurements of viscosity, the primary indicator of a fluid's lubricating property. The 51DV detects misapplication of lubricants, including wrong or mixed oils; detects fuel dilution in liquid-fueled engine applications and establishes lubricant viscosity for trending.

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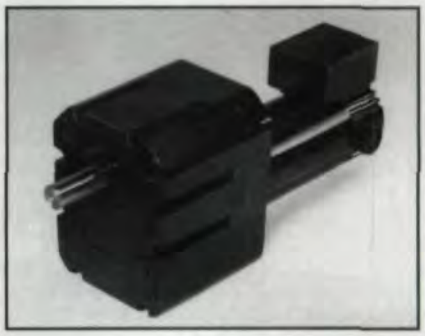

Improved High-Torque Gearmotors

Bison Gear announces an enhanced version of its new Series 650 parallel shaft gearmotors. This series has a gearbox measuring only 5" x 6.25" x 5.75", yet they deliver torques up to 720 in-lbs. They will now withstand occasional peaks up to 250% of their full-load torque. Gear ratio ranges are from 11:1 to 2206:1 and speeds range from 0.7 to 16 rpm. The units are lifetime oil lubricated and sealed with gaskets and O-rings. The gearmotors are available in 1/20, 1/6 or 1/2 hp and in 2-, 3-, 4- and 5-stage versions.

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Portable CMMs

Weighing only 14 lbs, the new **Romer 1000** series articulating arm CMMs are completely portable. The arm and its components are contained in a single suitcase, and setup takes less than five minutes. The arm is mountable in any orientation, even upside down. Arm lengths range from a 4-ft. to a 10-ft. diameter measuring envelope.

Measurement options extend its reach to 32 feet with no loss of accuracy. The system comes with a color notebook computer and software featuring graphic representation and on-screen relational storage of measured or theoretical geometric features. It interfaces to other popular CMM and engineering software and is Windows® 95-compatible.

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New Preservatives

Esgard, Inc. introduces the "P" preservatives. These coatings are used in preservation and packaging of metal items such as gears, turbines, engines, automotive parts, lubrication systems, steam components and fill and drain voids. They are qualified per MIL-C-16173E. All are VOC-compliant, easy to remove and safe.

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GEI International introduces the ONS-0180 series of collimating magnifiers specifically designed to eliminate parallax errors when viewing objects. They are available with magnifying powers of 7x and 10x, and they feature flat-field, achromatic, color-corrected lenses, which provide distortion-free viewing. Applications include scientific measurement using meters, scales, thermometers, etc., and most inspection operations involving the visual measurement of an object. May be used in normal room light or on a light table or box.

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The NOVAMATIC internal grinder series handles parts with ODs from 10 to 200 mm, while the NOVAMATIC external grinder series has an OD range from 10 to 350 mm. These machines from **Meccanica Nova** offer straight/spherical grinding through interpolation, quick changeover tooling, total machine accessibility for easy maintenance, machine enclosures to EEC standards for environment protection, CNC or PLC controls, U.S. electrics, in-process/post-process gaging, part handling automation and multiple workholding/wheel dressing methods.

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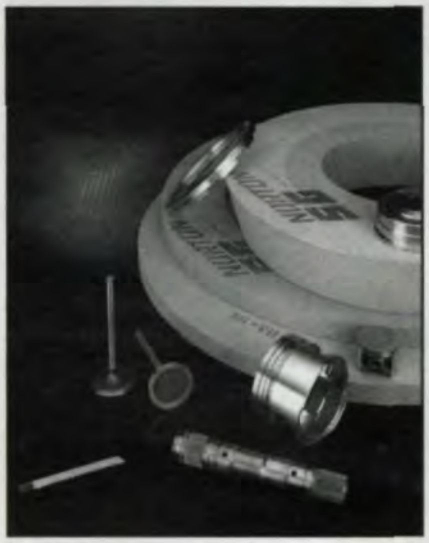

New Ceramic Abrasives

The Norton Co. announces its new Targa-VH Bond product line of advanced ceramic abrasives for demanding grinding jobs, particularly in cylindrical and plunge centerless grinding. The products are available in two concentrations: 50% (5TG) for difficult-to-grind materials, small contact areas and heavy stock removal, and 30% (3TG) for easy-to-grind materials, large contact areas and light stock removal. Grit sizes are 80/3, 80/4, 120/3 and 120/4. Wheel dimensions include >16–24" OD up to 6" thick, >24–36" OD up to 4" thick and >36–42" OD up to 3" thick.

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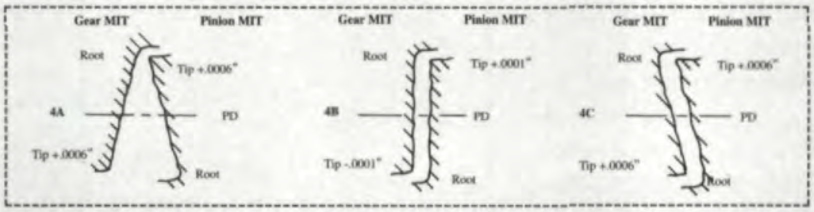
Duh!

Space aliens stopped at our editorial offices on their way to blowing up the White House and deliberately caused errors to appear in the Smith & Laskin article, "Noise Reduction in Plastic & Powder Metal Gear Sets." We regret any inconvenience this may have caused.

Fig. 4 should appear as shown:



The equation that appears in the middle of page 21 should read,

$$\phi_A = \phi - (57.3)^2 \frac{2\,(\text{MIS})}{d_B = (\varepsilon_2 - \varepsilon_1)\tan\phi}$$

The Bridges of Cook County and Other Sagas

Gear Technology's bimonthly aberration — gear trivia, humor, weirdness and oddments for the edification and amusement of our readers. Contributions are welcome.

In spite of being the "Second City," Chicago has always cultivated a reputation for bigness. We're known for big talk, big shoulders, big basketball players—and big gears. While not necessarily the biggest in the world (more about that later), some Chicago gears are among the hardest working.

Take, for example, our bridge gears.

Among the city's other claims to fame is that it is the home of the double-leaf trunnion counter-balanced bascule bridge—the lift bridge for short. The span of such bridges splits in the middle, each side lifting toward the shore to leave the river passage open.

Crucial to the working of the bridge is the rack and pinion system that raises and lowers the leaves (the equivalent of a block or more of city street) several hundred times a year in all weathers.

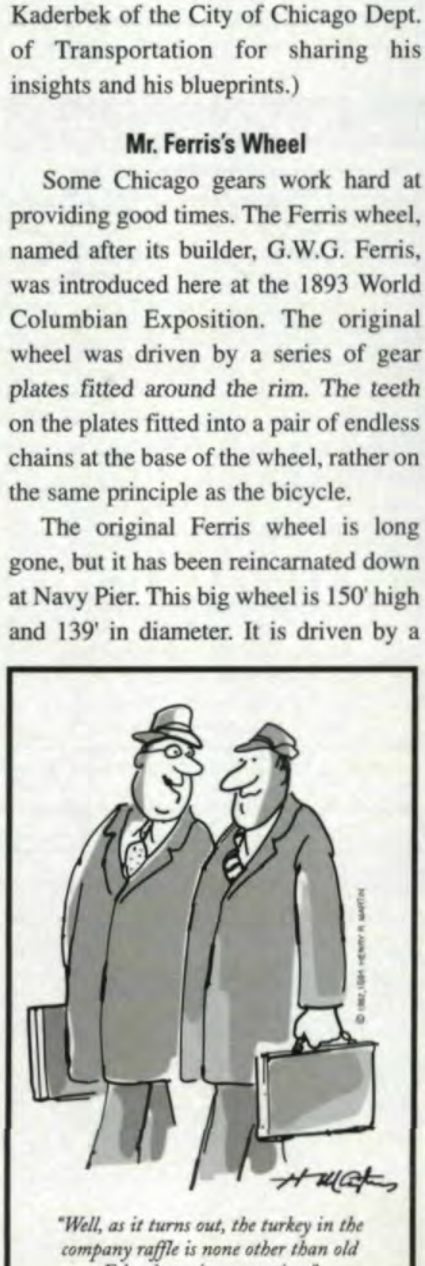
Each of the 32 bridges across the Chicago River is one-of-a-kind, but the one at E. 106th St. is fairly typical. Built in 1927, each of its leaves is lifted by a curved rack with 46 teeth and an 18' pitch circle radius. It moves through an arc of 78° 49'. It is driven by a 14-tooth pinion or bull gear with a pitch diameter of 31", a root diameter of 28" and an outside diameter of 36". Its pitch radius is 15.5". Both the rack and pinion are made of cast steel.

The reduction gears in the motor that drives the system aren't wimpy either. The pinion has 15 teeth and an outside diameter of 17". The three gears have 59, 71 and 125 teeth with outside diameters of 86", 73" and 50.8" respectively. According to the blueprints, the teeth all "conform to the Brown & Sharpe, standard, 14.5° involute, except that the pitch line thickness of the tooth shall be 0.49 times the circular pitch to provide backlash." (Our thanks to Stan Kaderbek of the City of Chicago Dept. of Transportation for sharing his insights and his blueprints.)

Mr. Ferris's Wheel

Some Chicago gears work hard at providing good times. The Ferris wheel, named after its builder, G.W.G. Ferris, was introduced here at the 1893 World Columbian Exposition. The original wheel was driven by a series of gear plates fitted around the rim. The teeth on the plates fitted into a pair of endless chains at the base of the wheel, rather on the same principle as the bicycle.

The original Ferris wheel is long gone, but it has been reincarnated down at Navy Pier. This big wheel is 150' high and 139' in diameter. It is driven by a DC drive consisting of eight synchronized motors.



"Well, as it turns out, the turkey in the company raffle is none other than old Edsy down in accounting."

Michael Emerson of VOA Associates, the pier's architects, points out that the Ferris wheel can be viewed as a giant toothless gear set. The eight motors drive eight balloon wheels (think giant rubber tires), which meet the outer diameter of the wheel and drive it by friction.

And Big Gears From Down Under

The World's Biggest Gear Saga, cont'd. Addendum has received a fax from Mr. Peter Mayo of Toronto, N.S.W., Australia. He tells us that the company Envirotech has designed and installed a 92 meter (302') diameter "red-mud" tailings thickener in Western Australia. Around the perimeter of the tank is a series of toothed racks which form a circle of 93 meters (305'). The rack forms the final drive for the thickener stirring device.

Mr. Mayo also raises an interesting philosophical question. Since neither this thickener rack nor the Bucyrus-Erie dragline rack mentioned in our last issue move, are they "real" gears? Or should that name be reserved for gears that rotate? If so, what do we call these big rack drivers? And what are "real" gears anyway? Real as opposed to what? Unreal? Fake?

Anyone ready to take on the metaphysics of gears? Addendum is up for it if you are. ○

The Addendometer: If you've read this far on the page and enjoyed it, please **circle 225.**