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Gear Expo
OCTOBER 20-23, 1991

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THE JOURNAL OF GEAR MANUFACTURING

JULY / AUGUST 1991

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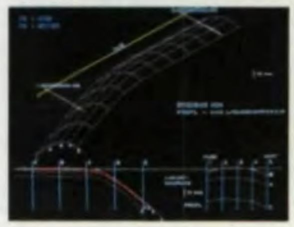

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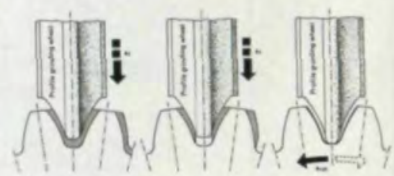

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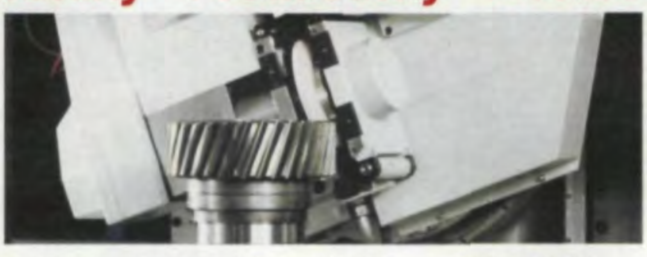

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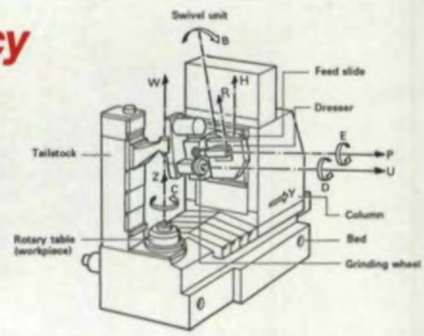

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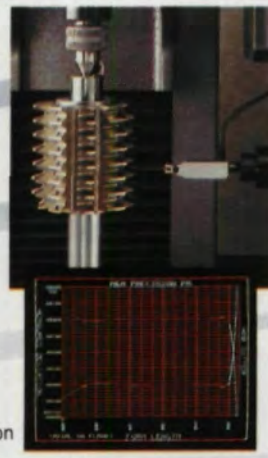

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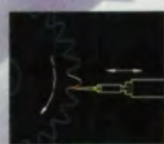

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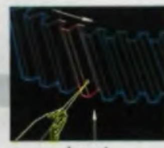

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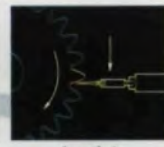

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Propping Up The Falling Sky

"So Henny-Penny, Cocky-Locky, Ducky-Daddles, Goosey-Poosey, Turkey Lurkey, and Foxy-Woxy all went to tell the king the sky was a-falling."
- Old English Fairy Tale

PUBLISHER'S PAGE

The sky may not be exactly falling for the gear industry, but things have been better. Last year's "332 Report" and the recent Department of Commerce "National Security Assessment of the U.S. Gear Industry" have confirmed what many of us have known all along - that the U.S. gear industry, if not actually sick, is certainly not in the best of health. Meanwhile, Joe Arvin's report, found elsewhere in this issue, suggests that overseas gear manufacturing is getting healthier. If the situation remains unchanged, the future of our industry could be bleak indeed.

The question now is, what do we do about it?

The problems facing the industry are tough ones. Many of them, like an adequate supply of trained workers, are part of much larger problems that confront our nation as a whole. Others, like the internal industrial policies or the differing cultural and economic expectations of our overseas competitors, are not open to "quick fixes" - assuming they're "fixable" at all. Some, such as the disparity between wages paid here and elsewhere, while still significant, are not as crucial as they once were, but have been replaced by others just as tough, like our own crippling national debt and balance-of-trade problems.

But we can't just sit around wringing our hands; nor is it enough to go crying, "The sky is falling!" We have to start addressing the question of what SPECIFICALLY we are going to do to restore the health of our domestic gear industry.

First, we have to shed the notion that simply lobbying Congress will be the solution to all our problems. Government can help the gear industry in some ways, but we have to go to our supporters in Congress with specific plans and ideas. If investment tax credits, subsidies for training, seed money for research, or more protectionist trade legislation are what we think we need, then let's ask for them. Meaningless cries to be provided with a "level playing field" or a generalized wail to "Do something!" are not enough.

Furthermore, we should remind ourselves that the ride on the government gravy train is coming to an end. Government money, either from outright grants or tax breaks, is going to be much harder to come by in the future. Ironically, our crushing national debt is part of our problem and adding to it, no matter how good the short-term goal, doesn't help.

We also have to face the fact that some of our problems cannot be solved by the government. The internal economic policies of other countries are beyond our control, as are the customs and practices that may give them a competitive edge. Instead we need to reform our own practices.

The solutions to the gear industry's hard, intractable problems will have to come from within the industry itself. Perhaps gear manufacturers need to begin to take a different view of who their competition is. The term "global village" is fast becoming a cliché, but it is an accurate description of the current gear manufacturing climate. If investment and training decisions are made on the basis of only what domestic competitors are doing, U. S. manufacturers are missing the big picture.

If present conditions remain unchanged over the next fifteen years, many current domestic gear manufacturers may not be in business. A more prudent strategy might be to take a long hard look at what the most successful manufacturers, no matter what their location, are doing. They're the people whose game plans U.S. gear manufacturers should be studying and trying to better.

For whatever reasons, our European and Pacific Rim competitors are more willing or able to spend money on state-of-the-art equipment and long-term training. A telling comment on the present state of gear manufacturing in the U.S. is a report from Gleason Works: They have taken orders for over 100 of their new Phoenix gear systems, a little less than 20% of which are going to the U.S. market. The rest are going overseas.

Why is that?

It seems as though the American gear manufacturing industry is waiting for the other guy to

PUBLISHER'S PAGE

move first. No one wants to spend the money or risk the short term profits for the sake of a long-term advantage. But refusal to run the risk is self-defeating in the long-run.

Perhaps U.S. gear manufacturers need to ask themselves the really hard question: Do I want to be manufacturing gears 20 years from now? If the answer is "No," then the present course fine. If that vision is not one that is attractive, then the time has come to make some other choices.

Someone is going to have to have the courage and foresight to make the first move and do the creative investment in the future.

Tough? Yes. Risky? Doing nothing is riskier. Necessary? Absolutely.

Only after we have gone through the soul-searching and brain work necessary to develop specific plans for pulling ourselves up by our own bootstraps, and then summoned the courage to make those plans work, can we legitimately go to others to demand solutions. Running about like Henny-Penny and her friends, crying, "The sky is falling!" is not enough.

Michael Goldstein
Publisher/Editor-in-Chief

VIEWPOINT

Pacific Rim Gives Stiff Competition To U.S. Gear Producers

Joe Arvin

This past fall, I had the opportunity to travel to Japan, Korea, Taiwan, and Singapore to witness first-hand the status of the power transmission and machine tool industries in these areas. Points of interest included equipment, material handling, computerization, wage and tax structures, inventory controls, and workforce attitude.

On this year's tour, I was accompanied by 24 other gear industry managers, government personnel, and people affiliated with the INFAC program. During our 17-day trip, we visited several manufacturers of loose gears, power transmission components, and machine tools, and gear training and research institutes.

Their Facilities

This trip was my third visit to the Orient. In view of what I had seen on my last visit in 1985, industry in these areas has experienced vigorous growth and has instituted massive reinvestment. We continually saw facilities equipped with the latest CNC equipment. Many companies were reporting sales that have doubled or even tripled in the last ten years.

For example, when I first visited the Kohara Gear Industry Company of Japan nine years ago, their facility and equipment could best be described as modest. On this trip, I found that they have doubled in size and have recently added a second location. They are now equipped with the latest machine tools and are doing CBN and diamond grinding of spur gears. They also have a Gleason Phoenix cutter and grinder on order. For any gear company to double its size in nine years is remarkable.

Another example of the growth rate of Pacific Rim manufacturers is Okubo Gear Company, also of Japan. Okubo has tripled in size since 1979 and now has a total land package of 13.8 acres with 5.9 acres of buildings combined in two locations. Considering the scarcity and high cost of land in Japan, the extent of their growth and reinvestment is quite evident.

A final example of the industrial growth in these areas is Tong Il, a Korean manufacturer of power transmission components and machine tools. In 1985, Tong Il was operating as a producer of transmissions and axles for automotive applications, and they were also just beginning to expand their product to include machine tools. With sales of $60 million, they were projecting 1990 sales of $253 million. Actual sales for last year were $430 million. They had underestimated sales by $177 million. We saw hundreds of Tong Il machines throughout Korea and Taiwan.

In terms of technology, the facilities we saw were as good or better than those of most gear companies in the U.S. They are no longer operating with just a few pieces of high precision equipment. In addition, the majority of the companies reported that substantial percentages of their annual sales were being allocated to research and development.

The remarkable growth of these companies was not the only change we saw. Other, more subtle changes were taking place as well.

While the participative approach made famous by Asian management in the 70s and 80s is still in use, there is less reliance on it now. The changing attitudes of the Asian workforce - which I will address later - may account for this change.

In the majority of the plants

Joe Arvin
is the President of Arrow Gear Co., Downers Grove, IL. Over the last nine years he has conducted industry informational tours to gear manufacturing countries around the world.

we visited, we saw extensive use of automated machine loading and material handling. We also saw that in most cases, operators were running several machines at one time.

While their equipment, technology, and (in most cases) housekeeping were quite impressive, we did see much less concern for operator safety than most of us are used to. In some plants we saw operators lifting heavy objects and people working around machines without safety glasses. In one plant we even saw operators wearing sandals in a work area with chips on the floor.

Throughout the 1980s, American industry was presented with numerous articles, books, and seminars which extolled the virtues of JIT (Just In Time) manufacturing. During our plant tours, we were able to see JIT in its "truest form". We saw thousands of finished goods waiting for shipment - not quite the 2-1/2 hour inventory we had heard so much about.

Their People

To better explain the characteristics of the Asian workforce, let me first mention the order in which the countries that we visited rank in terms of industrial development. First is Japan, followed in descending order by Korea, Taiwan, and Singapore. Wages in these countries are directly proportional to their status in this hierarchy. Hourly wages in Japan are $13-14; in Korea, $7-8; in Taiwan, $4-5; and in Singapore, $3-3.50. These countries experience an average unemployment rate of only 2%.

An interesting note on wages is that on my last visit to Korea in 1985, the average hourly wage was around $4-5. The current average of $7-8 is a substantial increase in just five years - a clear indicator of vigorous growth.

The work ethic of the Asian workforce remains very interesting. Traditionally, Asian workers have a sense of duty to their company. Peer pressure to achieve peak productivity and avoid being a "disgrace" to the company is intense. This level of dedication seemed apparent in the plants we visited. In more than one instance, we saw operators running from machine to machine. But this ethic may be changing.

Each of our technical visits would end in a question and answer session with company personnel. During these sessions, we asked this particular question: "Of the following, what do you feel is your number one problem?

- Lack of sufficient capital;
- Personnel problems;
- Adequate engineering support to produce quality products that would be competitive in the world market."

Surprisingly each company's response was the same. "Personnel problems" were their number one concern.

After further explanation, it appeared that employees are starting to refuse overtime and prefer a 40-hour work week. One can speculate that this is because workers are starting to accumulate more disposable income, and they want more free time to spend it. Some managers also told us that their workers are becoming more "belligerent". Some of us felt that it was not necessarily belligerence, but rather that their workers were becoming more "westernized". Perhaps this is an inevitable side-effect of their industrial growth.

Training

As in earlier trips, I was impressed with the amount of training being done at these facilities. Training is just another accepted aspect of the Asian worker's job; and judging by the accelerated move into computerized industrialization, this commitment to training is essential.

Governments in the Pacific Rim, particularly in developing countries like Taiwan and Singapore, are actively involved in supporting industrial training.

While in Singapore, we visited two government sponsored training facilities - Philips Government Training Centre and Precision Engineering Institute (PEI). While PEI is larger than Philips, both facilities operate for the same purpose - to train young people in precision engineering and machining skills.

Both facilities offer two-year programs. In the first year, students learn basic machining techniques. In the second year, their studies involve computerized machining (CAD-CAM), where they acquire classroom and hands-on experience in the operation and programming of CNC machine tools. Their course of study also encompasses a full range of manufacturing processes. These facilities are equipped with state-of-the-art equipment. At PEI, we estimated approximately $25 million in equipment on the shop floor - the vast majority of which was brand new.

Graduates from these programs are sent into the workforce with high levels of skill, yet they will earn only modest salaries. After

VIEWPOINT

Asian workers are becoming more Westernized, sometimes refusing overtime and preferring a 40-hour work week. Some of their managers see these changing attitudes as "belligerent," when they may simply stem from the desire for more leisure time in which to spend increased income.

graduation, PEI students can expect to earn $330 per month. After five years of on-the-job training, they can progress to $720 per month.

I'm sure that any U.S. gear company would love for someone to come looking for a job and say, "I know how to program my own machine. I know how to operate turning, milling, drilling and grinding equipment. I understand tempering and rehardening and their major causes, and on top of that, I'll work for $3 per hour."

The fact is that the Pacific Rim's posture as a competitive force will continue to increase largely because it is so far ahead of the U.S. in the implementation of training.

And why doesn't the U.S. gear industry train its workforce to this extent? Speaking for Arrow Gear, the reason we do not is that our domestic competitors don't. Adding the cost of this extensive training to our product would result in an uncompetitive price. Yet, a trained workforce is essential in meeting the increasing precision requirements of our industry.

This no-win scenario has been a source of great concern in the gear industry for some time, although I believe we are finding a solution to this dilemma in the INFAC Program.

The INFAC Program, which stands for Instrumented Factory, is a government-sponsored program aimed at providing assistance with its training and research needs to the U.S. gear industry. Located at the Illinois Institute of Technology in Chicago, INFAC will offer formal training programs in gear technology. In addition, the facility will contain of a state-of-the-art machining shop where new processes can be studied and students can obtain hands-on experience.

I believe the benefits we stand to gain from this program will be essential in competing with the highly trained workforce and the high precision capabilities of Pacific Rim manufacturers.

In Conclusion

This visit renewed my concerns for the competitive ability of Asian manufacturers. Their massive reinvestment, technology, and trained workforce make them a formidable threat.

However, it's not too late for the U.S. gear industry. With a commitment to ongoing improvement and growth, cooperation, and plain hard work, we can succeed. But each of us has to do our best!

For anyone interested in additional information on the findings of this tour, a videotape of the trip's highlights will soon be available. Consisting of both video of our plant visits and comments from tour participants, this program will provide a detailed and interesting insight into Asian manufacturing. For more information on ordering, contact the office of Dr. Maurice Howes, Director of INFAC at (312) 567-4200. ■

Explore "The World of Gearing" in Detroit.

Kelli R. Hopkins

AGMA's Gear Expo '91, "The World of Gearing," opens October 20 and runs through October 23 at the Cobo Conference & Exhibition Center in "The Heart of the Manufacturing Industry," Detroit, MI. Gear Expo '91 is "the largest trade show ever specifically organized for the gear industry," according to Rick Norment, AGMA's Executive Director.

Gear Expo started in 1986 as a table-top exhibition and has grown into a full-fledged trade show. The Expo is held every two years on a rotation schedule among several cities. Detroit, the oldest city in the Midwest, especially lends itself to this trade show. The "Motor City" is the heart of America's manufacturing base and produces millions of gearsets each year.

Gear Expo '91 will provide 35,000 square feet of exhibits by 79 companies from around the world. This forum offers gear manufacturers and suppliers an opportunity to exhibit their products and gives visitors the chance to make comparisons of the products and ask company representatives questions right at the show. An index of Gear Technology advertisers who are exhibiting at this year's expo can be found on the adjoining page.

Products and processes on display include broaching, custom gears, cutting tools, finishing, forging, grinding, heat treating, hobbing, inspection, lubricating, milling, shaping, shaving, and testing, to name a few.

The Cobo Conference & Exhibition Center is on the banks of the Detroit River in downtown Detroit and provides easy access to a variety of hotels, restaurants, cultural attractions, and historical sites.

Transportation in the downtown area is a matter of personal preference - visitors can choose from a variety of unique options. Downtown Detroit's elevated transportation system, The People Mover, carries people on a 2.9 mile track around the central business district. Buses, rental cars, taxi cabs, and antique trolley cars are other travel possibilities. Parking is no problem - there are 2,200 spaces at the Cobo Center and 3,000 spaces at the adjoining Joe Louis Arena.

Show hours are 12:00 p.m. to 6:00 p.m. on Sunday; 10:00 a.m. to 8:00 p.m. on Monday; 9:00 a.m. to 6:00 p.m. on Tuesday; and 12:00 p.m. to 4:00 p.m. on Wednesday.

Once again the AGMA Fall Technical Meeting will be held in conjunction with Gear Expo. The FTM will be held on October 23-25 at the Westin Hotel, Renaissance Center, a short distance from the Cobo Center. This year's meeting has been expanded to allow for more presentations. The papers will feature a variety of gearing subjects, including 3-D contact analysis, gear tooth friction, gear stress distribution, oil jet gear lubrication, and low noise marine gears.

For more information on either the 1991 Fall Technical Meeting or Gear Expo '91, call AGMA Headquarters at (703) 684-0211. ■

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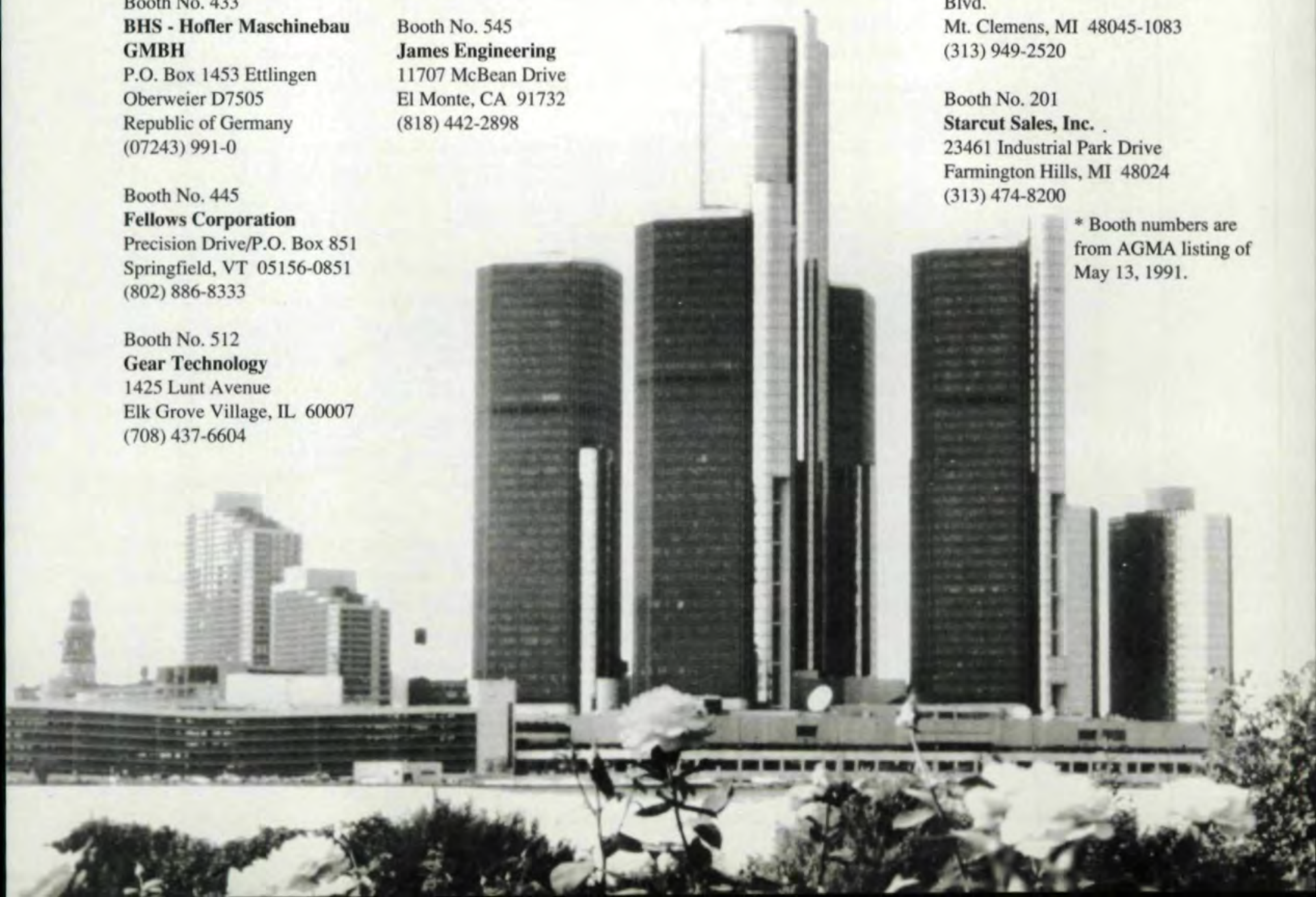

The Lubrication of Gears - Part III

Robert Errichello
GEARTECH, Albany, CA

Introduction

This is the final part of a three-part series on the basics of gear lubrication. It covers selection of lubricant types and viscosities, the application of lubricants, and a case history.

Selecting Lubricant Type

The choice of lubricant depends on the type of gearing and enclosure, operating speed and load, ambient temperature, and method of lubricant application. Most gears are lubricated with one of the following types: oil, grease, adhesive open-gear lubricant, or solid lubricant. The optimum lubricant for any application is the least expensive, considering both initial cost and maintenance costs, that meets the requirements.

Oil is the most widely used lubricant because it is readily distributed to gears and bearings and has both good lubricating and cooling properties. Also, contamination may be readily removed by filtering or draining and replacing the oil. However, it requires an oil-tight enclosure provided with adequate shaft seals.

Grease is suitable only for low-speed, low-load applications because it does not circulate well, and it is a relatively poor coolant. Grease lubricated gears are generally boundary lubricated because the grease is either pushed aside or thrown from the gear teeth. Contamination from wear particles or other debris is usually trapped in the grease and requires costly maintenance to eliminate. Grease is often used to avoid leakage from enclosures that are not oil-tight. However, if all the factors are considered, it is usually found that an oil lubricant is more economical and reliable than a grease for gear lubrication.

Open-gear lubricants are viscous, adhesive semi-fluids used on large, low-speed, open gears, such as those used in iron ore and cement mills, antenna drives, bridge drives, cranes, etc. Gears in these applications run slowly, and they are therefore boundary lubricated. The lubricant must bond strongly to resist being thrown off the gear teeth. However, the squeezing and sliding action of gear teeth tends to push the lubricant into the roots of the gear teeth where it is relatively ineffective. These lubricants are applied by hand brushing or by automatic systems which deliver an intermittent spray. Some open gear lubricants are thinned with a quick-evaporating solvent/diluent to make them easier to apply. Open-gear lubricants share the disadvantages of grease lubrication, and they are especially costly (and messy) to maintain. For these reasons, the trend is away from open gears toward enclosed, oil-lubricated gearboxes whenever possible.

Solid lubricants, usually in the form of bonded, dry films, are used where the temperature is too high or too low for an oil or grease; where leakage cannot be tolerated; or where the gears must operate in a vacuum. These lubricants are usually molybdenum disulfide (MoS_2) or graphite in an inorganic binder, which is applied to the gear teeth and cured to form a dry film coating. Polytetrafluoroethylene (PTFE) and tungsten disulfide (WS_2) coatings are also used. Solid lubricants are expensive to apply and have limited wear lives. However, in many applications, such as spacecraft, they are the only alternative and can provide excellent service.

Only oil lubricants will be discussed in greater detail. Oil should be used as the lubricant unless the operating conditions preclude its use. Generally, the simplest and least expensive lubrication system for gears is a totally enclosed, oil-bath of mineral oil.

The lubrication requirements of spur, helical, straight-bevel, and spiral-bevel gears are essentially the same. For this class of gears, the magnitudes of the loads and sliding speeds are similar, and requirements for viscosity and anti-scuff properties are virtually identical. Many industrial spur and helical gear units are lubricated with rust and oxidation-inhibited (R&O) mineral oils. The low viscosity R&O oils, commonly called turbine oils, are used in many high-speed gear units where the gear tooth loads are relatively low. Mineral oils without anti-scuff additives are suitable for high-speed, lightly loaded gears where the high entraining velocity of the gear teeth develops thick EHD oil films. In these cases the most important property of the lubricant is viscosity. Anti-scuff/EP additives are unnecessary because the gear teeth are separated, eliminating metal-to-metal contact and the scuffing mode of failure. Slower speed gears, especially carburized gears, tend to be more heavily loaded. These gears generally require higher viscosity lubricants with anti-scuff additives.

Hypoid gears, such as those used for automotive axles, are especially prone to scuffing because they are heavily loaded and they have high sliding velocities. For these reasons, hypoid gear oils have the higher concentrations of anti-scuff additives.

For critical applications, the contact temperature should be calculated with Blok's[1] equation and compared to the scuffing temperature of the lubricant. This quantitative method is effective for selecting a lubricant with adequate scuffing resistance.

Worm gears have high sliding velocity which generates significant frictional losses. Fortunately, their tooth loads are relatively light, and they are successfully lubricated with mineral oils that are compounded with lubricity additives. These oils contain 3% to 10% fatty oil or low acid tallow. The polar molecules of the additive form surface films by physical adsorption or by reaction with the surface oxide to form a metallic soap which acts as a low shear strength film, improving the "lubricity" or friction-reducing property.

Synthetic lubricants are used for applications, such as aircraft gas turbines, where the oil must operate over a wide temperature range and have good oxidation stability at high temperature. Ester and hydrocarbon synthetic lubricants have high viscosity indices, giving them good fluidity or low viscosities at very low temperatures and acceptable viscosities at high temperatures. The volatility of esters is lower than that of mineral oils of the same viscosity, thus reducing oil loss at high temperature. Despite their long service life, the extra cost of synthetic lubricants generally cannot be justified for oil-bath systems unless there are extreme temperatures involved, because the oil must be changed frequently to remove contamination.

Selecting Gear Lubricant Viscosity

The recommendations of AGMA 250.04[2] should be followed when selecting lubricants for enclosed gear drives that operate at pitch line velocities up to 5,000 fpm. AGMA 421.06[3] should be consulted for high-speed drives (> 5,000 fpm).

In our discussion of gear failure modes, we found that viscosity is one of the most important lubricant properties, and the higher the viscosity, the greater the protection against the various gear tooth failures. However, the viscosity must be limited to avoid excessive heat generation and power loss from churning and shearing of the lubricant by high-speed gears or bearings. The operating temperature of the gear drive determines the operating viscosity of the lubricant. If the lubricant is too viscous, excessive heat is generated. The heat raises the lubricant temperature and reduces its viscosity, reaching a point of diminishing returns where increasing the starting viscosity of the lubricant leads to a higher operating temperature and a higher oxidation rate, without a significant gain in operating viscosity.

Gear drives operating in cold climates must have a lubricant that circulates freely and does not cause high starting torques. A candidate gear lubricant should have a pour point at least 5°C (9°F) lower than the expected minimum ambient start-up temperature. Typical pour points for mineral gear oils are 20°F while synthetic gear lubricants have significantly lower pour points of about -40°F. Pour point depressants are used to tailor pour points of mineral lubricants for automotive hypoid gears to be as low as -40°F.

The pitch line speed of the gears is a good index of the required viscosity. An empirical equation for determining required viscosity is

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$$\nu_{40} = \frac{7000}{(V)^{0.5}} \tag{1}$$

where

ν_{40} = lubricant kinematic viscosity at 40°C, cSt

V = operating pitch line velocity, ft/min

V = 0.262 d n

d = operating pitch diameter of pinion, in.

n = pinion speed, rpm

Caution must be used when using AGMA recommendations for viscosity. The author knows of an application where two gear drives were considered to be high-speed. The pinion speed was 3,625 rpm, qualifying the gear units as high-speed gear drives per AGMA 421.06. The gear drives were supplied with oil having the recommended viscosity per AGMA 421.06 of ISO 68. However, because the pinion was relatively small, its pitch line velocity was only 3,000 fpm. This qualifies the gear drives as slow-speed per AGMA 250.04, which recommends a viscosity of ISO 150. Both gear drives failed within weeks of start-up by pitting fatigue. The empirical equation for this application give

$$\nu_{40} = \frac{7000}{(3000)^{0.5}} = 128 \text{ cSt} \tag{2}$$

This indicates that the viscosity per AGMA 421.06 (68 cSt) is much too low, and the viscosity per AGMA 250.04 (150 cSt) is appropriate. Hence, definitions of high-speed versus slow-speed must be carefully considered, and pitch line velocity is generally a better index than shaft speed. The gear drives were rebuilt with new gearsets and the ISO VG 68 oil was replaced with ISO VG 150. The gear drives now operate without overheating, and the pitting has been eliminated.

For critical applications, the specific film thickness should be calculated with Dowson and Higginson's[4] equation. The specific film thickness is a useful measure of the lubrication regime. It can be used with Fig. 1 as an approximate guide to the probability of wear-related surface distress.

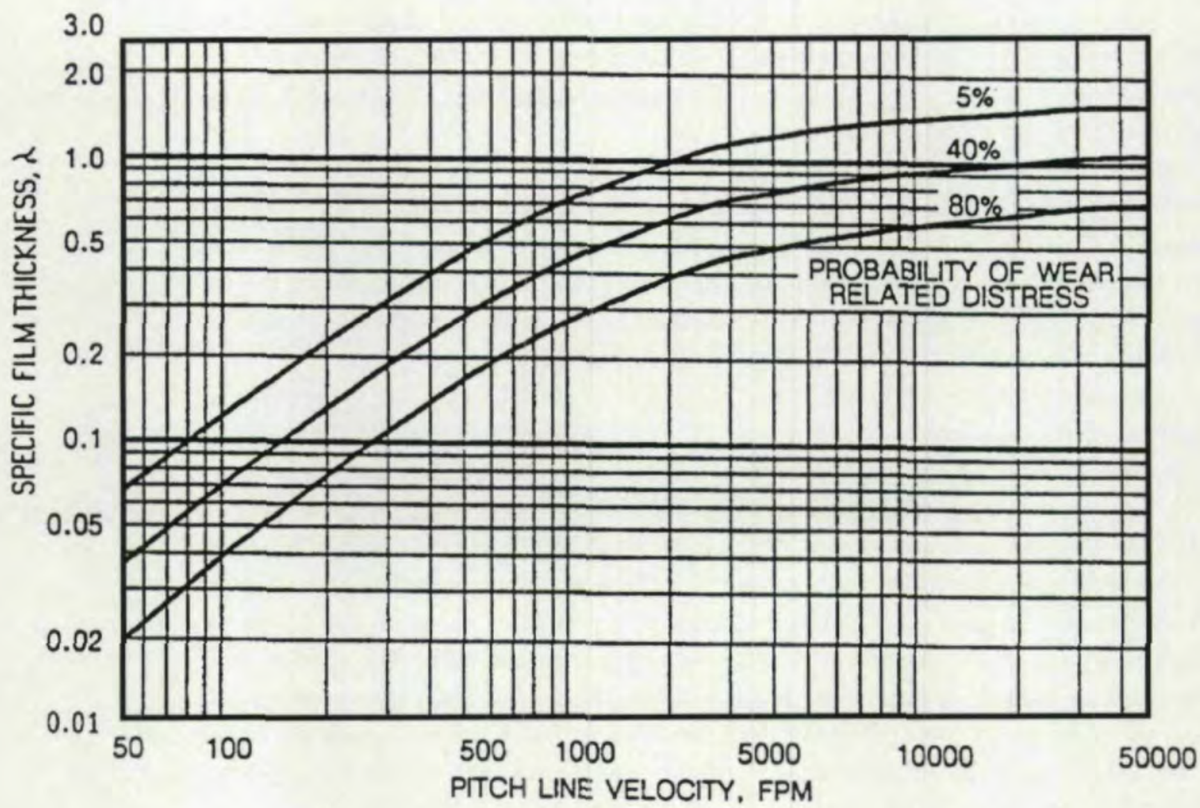

Fig. 1 - Probability of wear distress, percent. (From AGMA 2001-B88, 1988.)

Fig. 1 is based on the data of Wellauer and Holloway,[5] which were obtained from several hundred laboratory tests and field applications of gear drives.

Applying Gear Lubricants

The method of applying the lubricant to the gear teeth depends for the most part primarily on the pitch line velocity.

Splash lubrication systems are the simplest, but they are limited to a pitch line velocity of about 3,000 fpm. The gears should dip into the oil bath for about twice the tooth depth to provide adequate splash for pinions and bearings and to reduce losses due to churning. The gear housing should have troughs to capture the oil flowing down the housing walls, channeling it to the bearings.

The range of splash lubrication can be extended to about 5,000 fpm by using baffles and oil pans to reduce churning. However, above 3,000 fpm, providing auxiliary cooling with fans and improving heat transfer by adding fins to the housing is usually necessary.

Above 5,000 fpm, most gears are lubricated by a pressure-fed system. For gearboxes with antifriction bearings, spraying the oil at the gear mesh only and relying on splash to lubricate the bearings is permissible up to a pitch line velocity of 7,000 fpm maximum. Above this speed, and for gear drives with journal bearings, both the gears and bearings should be pressure-fed.

The oil jets should be placed on the incoming side of the gear mesh for pitch line velocities up to 8,000 fpm. Above 8,000 fpm, more oil is needed for cooling than for lubricating, and the oil flow removes heat best by being directed at the outgoing side of the gear mesh where the oil jets can strike the hot, drive-side of the gear teeth.

For very high-speed gears[6] (above 16,000 fpm), there is a danger that the amount of oil carried to the incoming side of the gear mesh may be inadequate, and it is prudent to add a supplementary flow at the incoming side of the gear mesh. Generally, about 2/3 of the oil flow should be supplied to the outgoing side of the mesh for cooling, and 1/3 of the flow directed at the incoming side for lubrication. The placement of the oil jets is a crucial factor when pitch line velocities exceed 20,000 fpm. At speeds this high, experiments are required to find the optimum number and location for the oil jets.

In pressure-fed systems, the following parameters must be considered to ensure adequate lubrication and cooling of the gear mesh: Quantity of flow, jet size, feed pressure, and number of jets. There are general guidelines, based on experience and experimentation, for specifying these parameters, but each application must be evaluated independently based on its particular operating conditions and requirements.

An empirical equation used to calculate the quantity of oil flow in gallons per minute is

$$q = P/c$$

where c is taken from Table 1

P = transmitted power, hp

q = oil flow rate, gpm

For a typical industrial application transmitting 200 hp, where weight is not critical, the designer might choose the constant c = 200hp/gpm, resulting in a copious flow of 1 gpm. On the other hand, for a high efficiency aviation application transmitting 200 hp, where weight is critical, c = 800 might be chosen, resulting in a lean flow of 0.25 gpm. Some applications may require different flow rates than those given by Table 1. For instance, wide-face, high-speed

Table 1 - Typical Oil Flows Per Gear Mesh

c (hp/gpm)	Flow Conditions	Comment
200	Copious	General industrial
400	Adequate	Typical aviation
800	Lean	Light-weight, high-efficiency aviation
1000	Starved	Only for unusual conditions

gearing may require a higher flow rate to ensure uniform cooling and full-face coverage.

The proper jet size, feed pressure, and number of jets must be determined to maintain the proper flow rate, jet velocity, and full-face coverage. The diameter of a jet can be calculated for a given flow rate and pressure based on the viscosity of the oil at the operating temperature.[7] There are practical limitations on jet size, and the minimum recommended size is 0.03". If a jet smaller than this is used, contaminants in the oil may clog it. Typical jet diameters range from 0.03"- 0.12".

The feed pressure determines the jet velocity, which in turn determines the amount of oil that penetrates the gear mesh. Typical feed pressures range from 20-100 psig. Industrial application feed pressures are typically 30 psig, and high-speed aerospace applications are typically 100 psig. In general, the higher the pressure, the greater the cooling,[8] but the higher the pressure, the smaller the jet diameter. Therefore, pressure is limited by the minimum recommended jet diameter of 0.03".

The number of jets should be sufficient to provide complete lubrication coverage of the face width. More than one jet for each gear mesh is advisable because of the possibility of clogging. The upper limit on the number of jets is determined by the flow rate and jet diameter; too many jets for a given flow rate will result in a jet diameter less than the minimum recommended.

Case History

In an industrial application, 24 speed-increaser gearboxes were used to transmit 346 horsepower and increase speed from 55 rpm to 375 rpm. The gears were parallel-shaft, single helical, carburized, and ground. The splash lubrication system used a mineral oil without anti-scuff additives with ISO 100 viscosity. After about 250 hours of operation, two gearboxes failed by bending fatigue. The gear tooth profiles were so badly worn determining the primary failure mode was impossible. Three other gearboxes with less service were selected for inspection. One had logged 15 hours, while the other two had operated for 65 hours each. Upon disassembly, no broken teeth were found, but all three

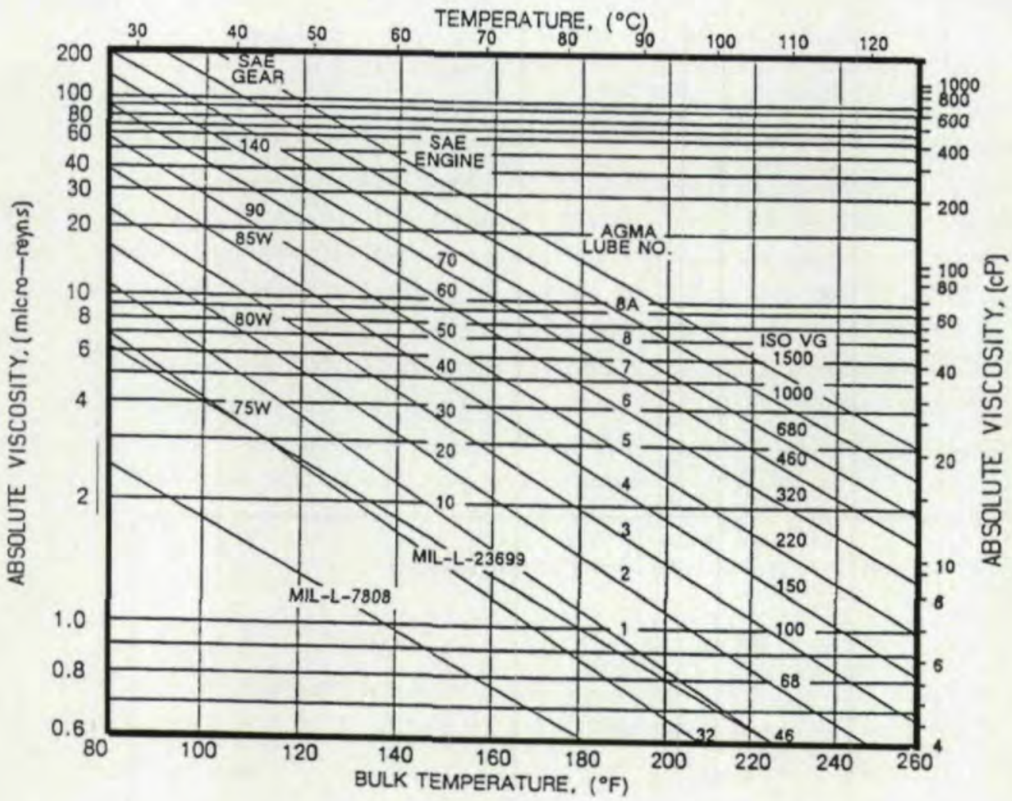

Fig. 2 - Absolute viscosity versus temperature for mineral gear lubricants with a viscosity index of 95. (From AGMA 2001-B88, 1988.)

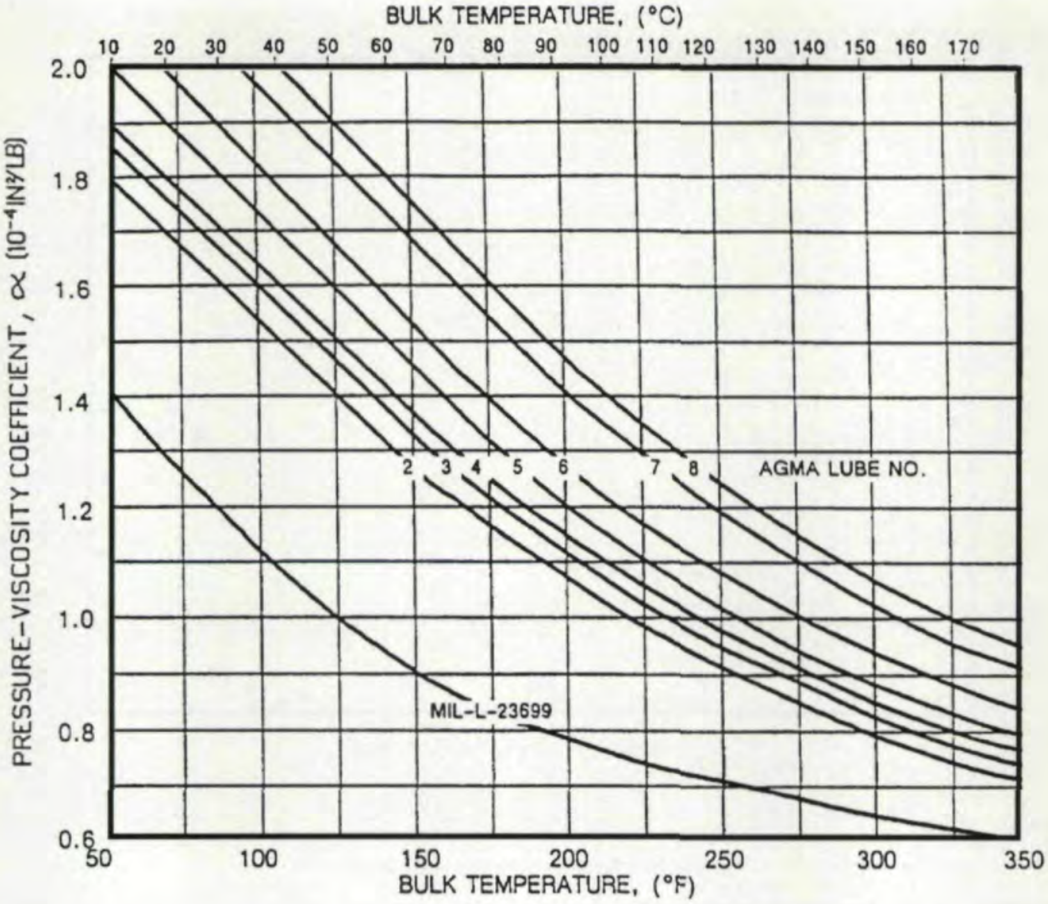

Fig. 3 - Pressure-viscosity coefficient versus temperature for mineral gear lubricants. (From AGMA 2001-B88, 1988.)

gearboxes had scuffed gear teeth. The primary failure mode was scuffing, and the earlier bending fatigue failures were caused by dynamic loads generated by the worn gear teeth. Subsequent inspection of the remaining gearboxes revealed that all had scuffing damage, which probably had occurred immediately upon start-up because the loads were not reduced during run-in.

Fortunately, a prototype gearbox had been run at 1/2 load for about 50 hours. When these gears were inspected, no signs of distress were seen on any of the gear teeth. The tooth profiles were smooth, with surface roughness estimated to be 20 μin rms, and the contact pattern indicated 100% face contact. This gearbox was reassembled and run under 1/2 load until its oil sump temperature reached equilibrium at 200°F. For this application, the ambient temperature was in the range of 50°F to 125°F. The center distance of the gears was 16 inches and the pitch line velocity was 400 fpm. Referring to AGMA 250.04, the recommended viscosity for these conditions is ISO 150 or ISO 220.

Using the empirical equation we get:

$$\nu_{40} = \frac{7000}{(400)^{0.5}} = 350 \text{ cSt} \tag{3}$$

Hence, the empirical equation recommends a viscosity close to ISO 320. It is apparent that the viscosity that was originally supplied (ISO VG 100) was too low.

The EHD film thickness was calculated with a special computer program.[9] The gear bulk temperature was assumed to be 230°F (30 degrees hotter than the measured oil sump temperature). The following data for the ISO VG 100 lubricant was obtained from Figs. 2 and 3:

$\mu_0 = 6.6$ cP(0.96×10^{-6} Reyns)
$\alpha = 1.02 \times 10^{-4}$ in^{2}lb

Fig. 4 shows a plot of the film thickness versus position on the pinion tooth. The minimum film thickness occurs low on the pinion

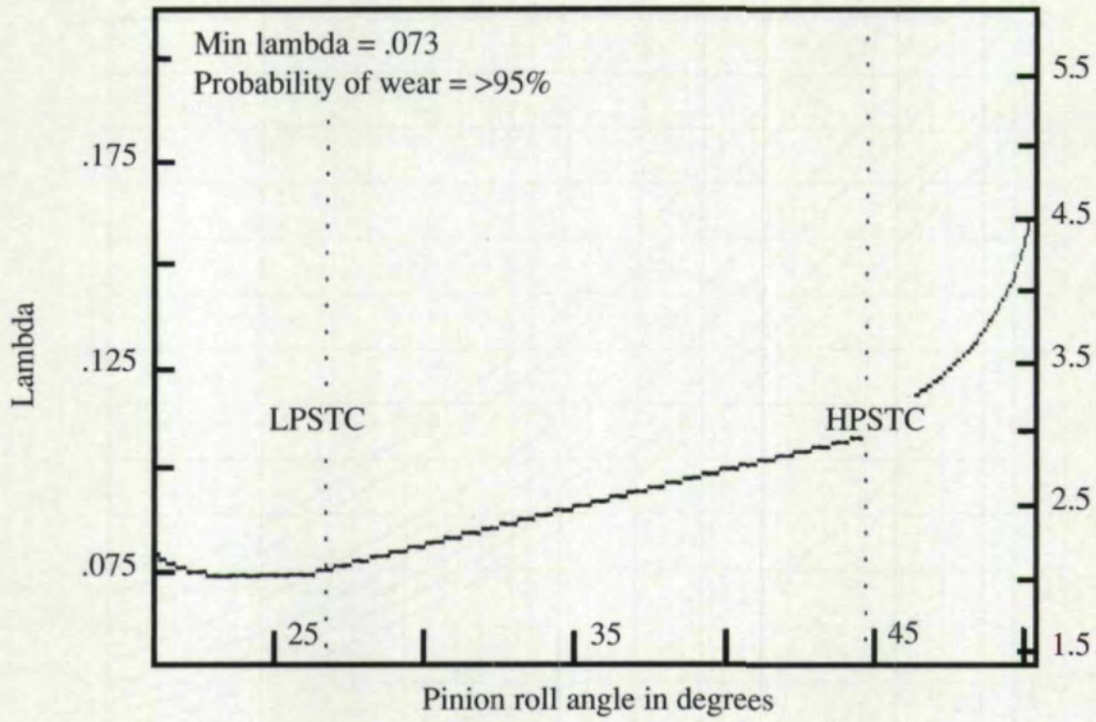

Fig. 4 - Film thickness versus pinion roll angle for gear tooth geometry of scuffed gearset.

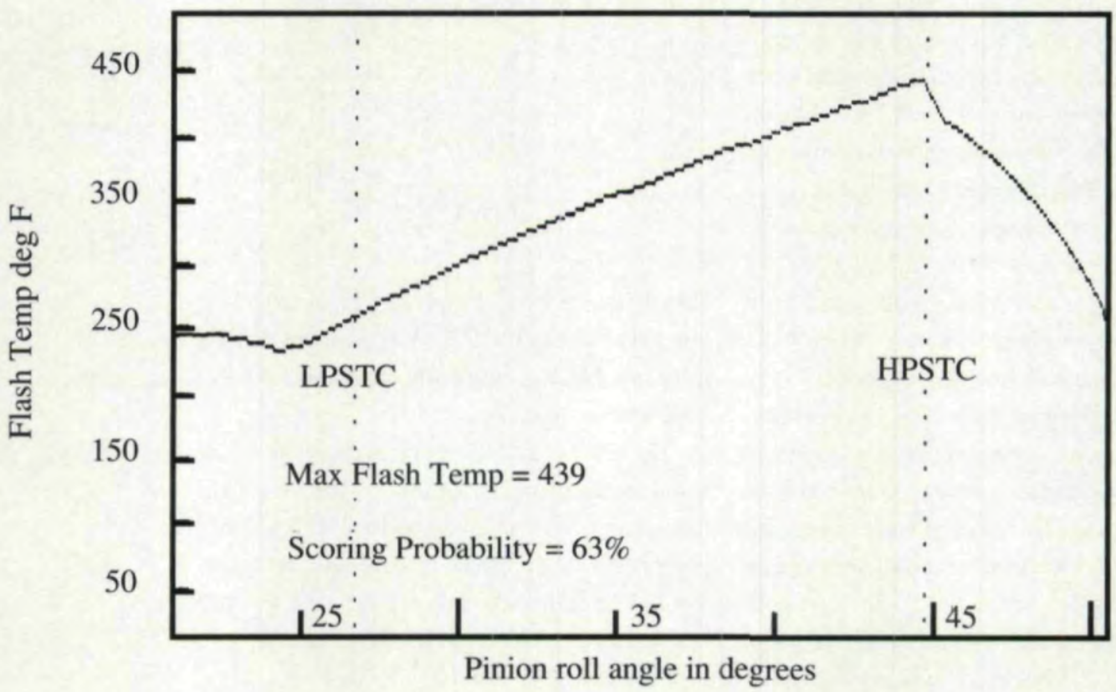

Fig. 5 - Flash temperature versus pinion roll angle for gear tooth geometry of scuffed gearset.

tooth near the lowest point of single tooth contact (LPSTC) where h_{min} = 2.1 micro inches. The specific film thickness, based on 20 μin rms surface roughness for both profiles, is $\lambda = 0.073$. Fig. 1 shows that the gears operate in the boundary lubrication regime. The program predicts that the probability of wear is greater than 95%.

The contact temperature was also calculated with the program. The scuffing temperature for the ISO BG 100 lubricant was calculated with the equation for non-anti-scuff mineral oils:

$$Ts = 146 + 59\ln(100) = 418°F$$

Fig. 5 shows a plot of the contact temperature versus position on the pinion tooth. The maximum contact temperature occurs high on the pinion tooth near the highest point of single tooth contact (HPSTC) where Tc = 439°F. The program predicts that the probability of scuffing is 63%. This is considered to be a high risk of scuffing. The relatively high temperature peak near the tip of the pinion tooth was caused by the geometry of the gears. The designer selected a long addendum tooth for the pinion. Long addendum pinions perform well in speed reducers, where they increase the amount of recess action and decrease the amount of approach action of the gear mesh. Since recess action is much smoother than approach action, long addendum pinions give speed reducers smooth meshing characteristics. When operated as a speed increaser, however, the approach and recess portions of the gear mesh reverse, making a long addendum pinion rough running and vulnerable to scuffing.

To explore the possibilities for reducing the scuffing risk, new gear tooth geometry was proposed with the pinion and gear addenda designed to minimize the flash temperature rise. The new gearset, analyzed with the program, assumed the lubricant was a mineral oil with anti-scuff additives, with a viscosity of ISO 220, and with the following properties:

$$\mu 0 = 10\ \text{cP}(1.45 \times 10^{-6}\ \text{Reyns})$$

$$\alpha = 1.09 \times 10^{-4}\ \text{in}^2/\text{lb}$$

$$Ts = 245 + 59\ln(220) = 563°F$$

Fig. 6 shows that the film thickness increases to h_{min} = 2.7 μin, and the specific film thickness increases to $\lambda = 0.097$. Fig. 1 shows that the gears still operate in the boundary lubrication regime, however, the probability of wear is reduced to 94%. Fig. 7 shows that the optimized gear geometry reduced the maximum contact temperature to Tc = 302°F. The combination of reduced contact

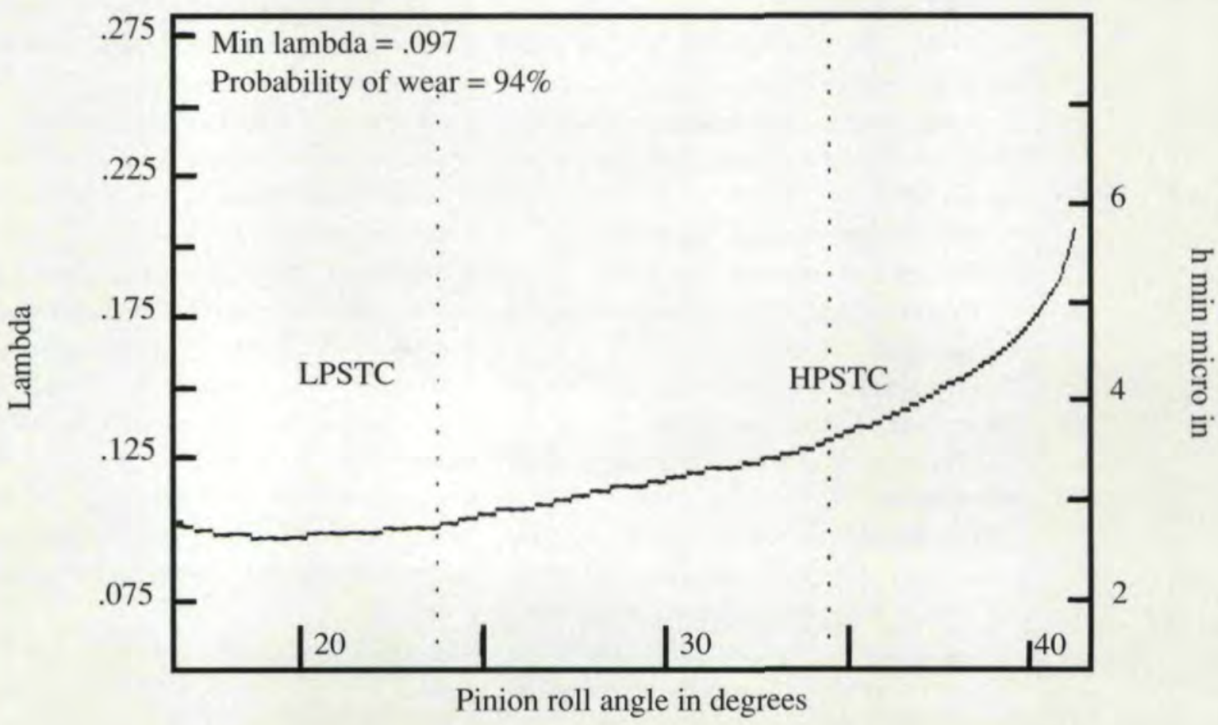

Fig. 6 - Film thickness versus pinion roll angle for gear tooth geometry optimized for maximum scuffing resistance.

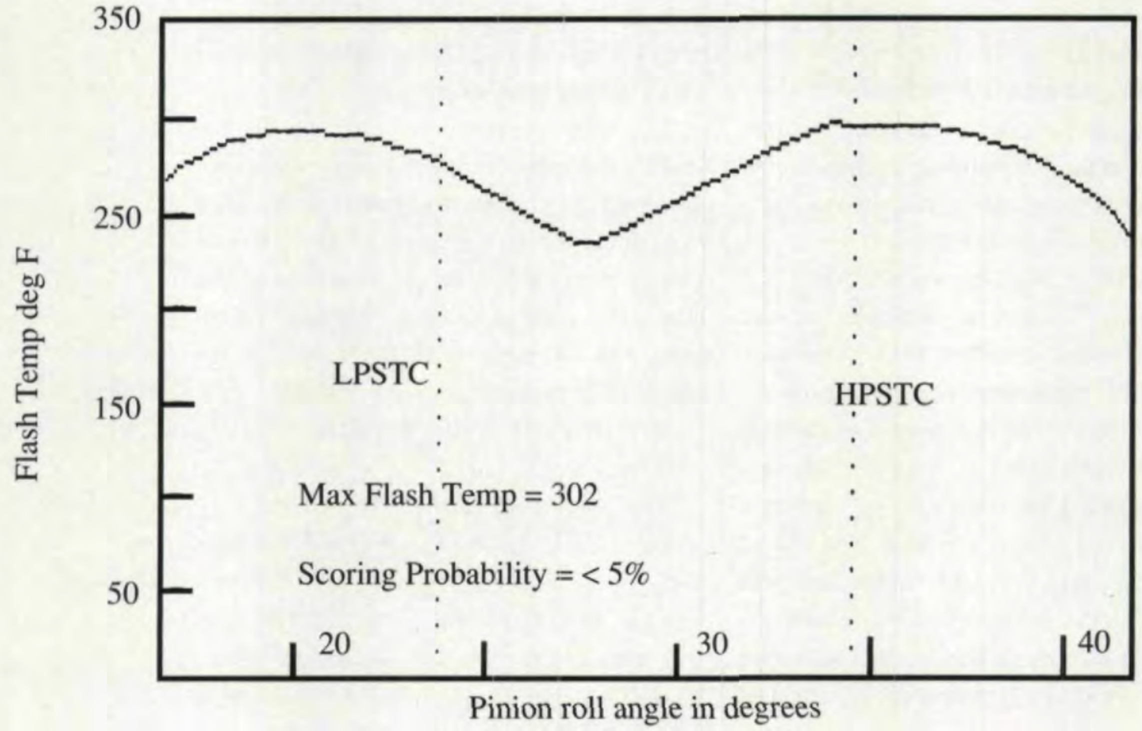

Fig. 7 - Flash temperature versus pinion roll angle for gear tooth geometry optimized for maximum scuffing resistance.

temperature and the increased scuffing resistance provided by the higher viscosity mineral oil with anti-scuff additives reduces the scuffing probability to < 5%.

Typical of many gear failures, this case history shows that several factors contributed to the failures:

•The lubricant viscosity was too low.

•No anti-scuff additives were used.

•A gearbox designed as a speed reducer was used as a speed increaser.

•The gear teeth were provided with a coating or plating to ease running-in.

•The gears were not run-in properly under reduced loads.

Gear failures, as exemplified by the case history, can be avoided if designers and operators recognize that the lubricant is an important component of a gearbox, and appreciate that the tribology of gearing requires the consideration and control of many interrelated factors.

Acknowledgement: *Reprinted by permission of Society of Tribologists and Lubrication Engineers.*

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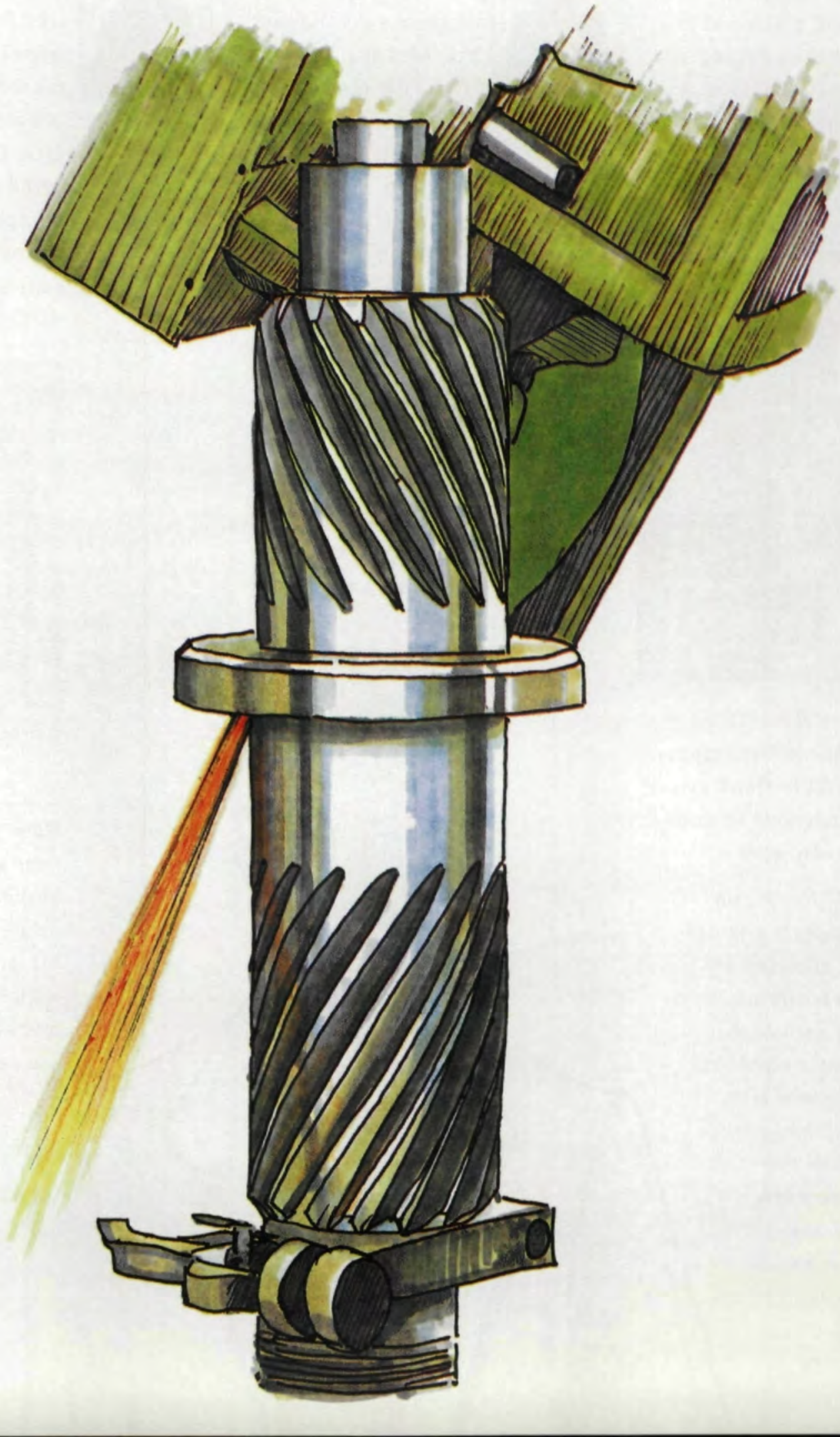

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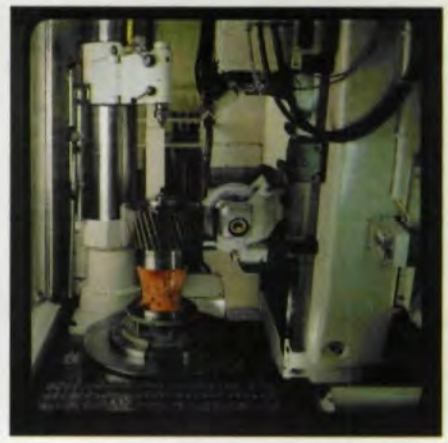

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Multi-Thread Hobs With New Cutting Diagrams

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Abstract:

In spite of the use of high speed steel hobs and state-of-the-art hobbing machines, increased hobbing efficiency is still demanded. However, this desire is often checked by the corner wear of hob teeth. A new method of analyzing the wear mechanism is proposed. The article provides the results of the research on the work of hobs with different cutting diagrams. It is also presents a new cutting diagram which makes it possible to decrease hob wear two to four times and the cutting force by 20% to 40%.

Introduction

During the last 20 years a great amount of research has been dedicated to the problem of increasing production efficiency in hobbing. Among the most efficient methods developed are the use of multi-thread hobs with optimized cutting conditions and diagrams and the search for a new method of gear cutting.

New Analysis of Hobbing

This article demonstrates a new method of analyzing the hob wear mechanism. Research on hob wear reveals that hob teeth are worn out on front and back edges non-uniformly. This is explained by the fact that the cutting edges of teeth cut layers of different thickness. They are also engaged in a non-free cutting condition. (They are cutting "V"-shaped layers when two cutting edges are applied to the work and "W"-shaped layers when three cutting edges are applied.) This brings up additional shear stress and compression stress, which depends chiefly on the angle (C_i) of chip flow, which lies between the perpendicular to the corresponding cutting edge and the direction of the chip flow. The greater the angle, the greater the value of additional deformation of the layers being cut. In the most simplified form it is possible to describe the tooth wear of a hob as:

$$h_i = A_i \int_0^{l_i} a_i \left[1 + \bar{A}_i \frac{l_w}{l_i} \cdot \mathrm{tg}\,(C_i)\right] dL \qquad (1)$$

where a_i = thickness of the cut layer, mm;
l_i = distance of cutting, mm;
l_w = distance of cutting on which the hob edge cuts complex "V" or "W" shape layers, mm;
$A_i, \bar{A}_i$ = constants.

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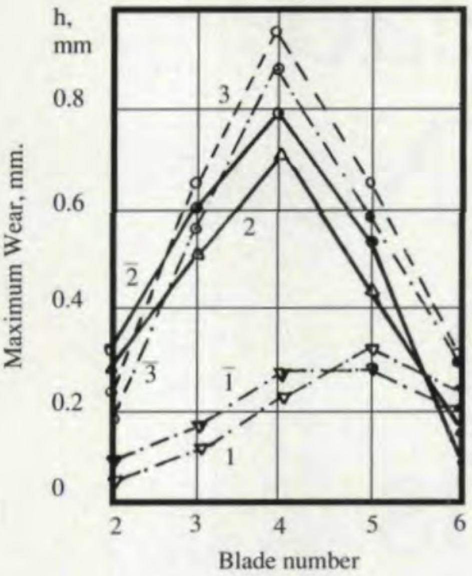


Hob: m = 4, 25, PA 20°, 3 RH, D = 120 mm, N = 12, P9K10;
Gear: HA 0°, Z = 48, b = 50, st. 40X;
Cut. Cond.: F = 2mm/rev., Conventional, V = 54m/min., T = 80 min. 1-top; 2-left; 3-right; $\bar{1}, \bar{2}, \bar{3}$ - experiment; 1, 2, 3 - calculated.

Fig. 1- Comparison of wear between experiment and that predicted by Model I.

The angle of chip flow was determined from independence of action of forces. For the top edge:

$$C_1 = \mathrm{arctg}\left\{\frac{[\mathrm{Cos}(\alpha_2)\cdot\int_0^{b_2} a_2\cdot db_2] - [\mathrm{Cos}(\alpha_3)\cdot\int_0^{b_3} a_3\cdot db_3]}{\int_0^{b_1} a_1\cdot db_1 + \sum_{i=2}^{3} [\mathrm{Sin}(\alpha_i)\cdot\int_0^{b_i} a_i\cdot db_i]}\right\} \quad (2)$$

where α_2 is the inlet leading profile angle, and α_3 is the outlet trailing profile angle of the side cutting edges of the hob.

The cutting thickness (a_i) and the width of cut (b_i) of each hob tooth, as well as length (l_i) and (l_w) can be calculated.[1] In order to know A_i and $\bar{A}_i$ we must make experiments in which the factors influencing the wear of the hob must be equal to the middle of their modified area. For example, if the area of feed is 1-3 mm/rev, module 2-4 mm, then the modified conditions of hobbing are f=2mm/rev and m=3mm. Fig. 1 illustrates wear on hobs.

It is clear from Formula 1 that hob (tool) wear can be decreased by dividing complex "V" and "W" shaped layers into elementary ones; that is, it is possible to divide complex cut layers into elementary ones by means of cutting diagrams introduced by Prof. S.N. Medvedichkov. Medvedichkov has found that in order to avoid the interference of the chips gathering from the top and both sides, the tooth profiles must be modified as shown in Fig. 2 (b). Hobs with this diagram are called "progressive". One profile is standard (low tooth-2, 4, 6, . . .), and another is higher by a value of e_1 and is reduced in tooth thickness (is narrower by e_2).

It is obvious from Fig. 2b that the top chip flow and the side flows do not interact at all. Medvedichkov has found that values e_1 and e_2 should be equal to the maximum thickness of layers cut by top a_1 and exit a_2 edges of the tooth, which engaged in cutting first. High teeth of progressive hobs get worn to a greater extent. Their wear is 2 to 3 times higher than that the wear of low teeth; however, the total wear of Medvedichkov's progressive hob (Fig.2b) is 2 to 3 times less than that of a standard hob (Fig. 2a).[3] Ueno, Terashima, and Hidaka[4] arrive at the same conclusion in their study of a similar situation.

The use of Medvedichkov's progressive hobs cuts down tool wear 2 to 3 times and cutting stress 20% to 40%. On the other hand, the use of the hobs increases the error of involute tooth profile. The author's hob reduces this error. (See Fig. 3.)

Hobs with New Cutting Diagrams

Hobs with new diagrams (Fig. 3a) differ from standard ones (Fig. 2a) by alternation of the standard tooth with protuberance with ones that are higher by a value of e_1 and narrower by a value of e_2 and have no protuberance.[5] Hobs without protuberance, but with the same cutting diagram, are shown in Fig. 3b. Cutting edges of high teeth are: edges 1-2; 5-6; 9-10. Standard teeth work with edges 1-2-3-4 and 7-8-9-10. In that case the error of generating involute tooth profiles by a standard (Fig. 2a) hob and by new ones (Fig. 3) are identical.

Besides, the author proposes to choose e_1 and e_2 so that the wear of "high" and "low" teeth are equal, which decreases the wear of new type hobs by 1.5 to 2.0 times compared to Medvedichkov's hobs. In this case the values e_1 and e_2 are determined as:

$$\left.\begin{aligned} e_1 &= a_1 - [a_1] + 0.5[h]\cdot \mathrm{tg}\tilde{\alpha}_1 \\ e_2 &= a_2 + 0.5[h]\cdot \mathrm{tg}\tilde{\alpha}_2 \end{aligned}\right\} \quad (3)$$

where [h] is allowed (extreme) value of hob wear in mm, and $\tilde{\alpha}_1$ and $\tilde{\alpha}_2$ are back clearance angles of top and side cutting edges. Values a_1 and a_2 in the prevented formula are equal to maximum thickness of layers cut by the top and inlet (leading) side cutting edges of the most worn tooth.

After calculating $[a_1]$ through Eq. (1) when $h_1 = h_2$, value e_1 can be obtained through Eq. (3). Such calculation of e_1 and e_2 has been done, but it is very complicated. In order to prove the calculation, a lot of experiments were made. Table 1 shows the experimental zones.

The experimental and calculated data are similar, but for determing values e_1 and e_2 it is better to use formula:

Table 1 – Experimental Zones

Dimension	Zone "from"	Zone "to"
Hob:		
Module (m), mm	2.5	5.0
Number of thread (n_h)	1	5
Number of gashes (z_h)	10	18
Gear:		
Number of teeth (z)	18	60
Cutting condition:		
Feed (f), mm/rev	1.0	4.0

$$\left.\begin{aligned} e_1 &= 0.7\cdot(n_h)^{0.7}\cdot(f)^{0.6}\cdot(z)^{-0.4}\cdot(m)^{0.7}\cdot(z_h)^{-0.9} + 0.5\cdot[h]\cdot tg(\alpha'_1) \\ e_2 &= 1.1\cdot(n_h)^{0.8}\cdot(f)^{0.4}\cdot(z)^{-0.4}\cdot(m)^{0.6}\cdot(z_h)^{-0.9} + 0.5\cdot[h]\cdot tg(\alpha'_2) \end{aligned}\right\} \quad (4)$$

Hobs with new diagrams (Fig. 3) and equal wear of "high" and "low" (standard) teeth were called "unloaded of uniform wear," and the hobs of the Medvedichkov diagram (Fig. 2.b) with values e_1 and e_2 determined by Eq. (4) were called "progressive of uniform wear." Fig. 4 shows the influence of value e_1 on the wear of such hobs. This function can be divided into four zones. In the first zone the value of e_1 is very small and the standard "low" tooth, engaged in non-free cutting condition, has limited wear (Fig. 4, line 1). When $e_1 = e'_1$, the wear of "high" and "low" teeth becomes equal. The "low" tooth cut "V"-shaped and "W"-shaped layers until $e_1 < e''_1$ (Zones 1 and 2). It is clear from Fig. 4 that value $[a_1]$ in Eq. 3 is $[a_1] = e''_1 - e'_1$. In Zone 3 "low" teeth cut elementary layers, and the wear of the "high" tooth is 2-3 times higher than the wear of the "low" tooth; however, the total wear of such hobs is 2-3 times less than that of a standard hob (Fig. 5). Further increasing of e_1 led to unfortunate results: the wear of the high tooth increased (Fig. 4, Zone 4). Such increasing of wear depends on deterioration of the cutting condition; the high tooth in such a case (rather like needle) engaged in cutting "V" shaped layers.

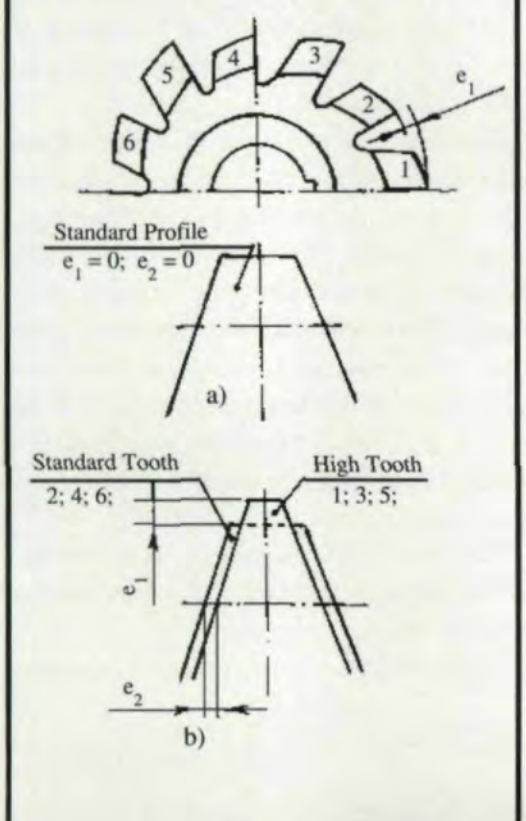


Fig. 2 - Standard (a) and progressive (b) hobs.

Now we can make a very important conclusion: modifying thickness of layers cut by the top cutting edge has influence on corner wear on side cutting edges. This fact corroborates the author's conclusion that the wear of hobs depends on flow chip angle C_i (Eqs. 1 and 2).

Fig. 5 shows the influence of cutting diagrams on the wear of hobs. Tests showed that the best wear resistance was obtained by use of hobs with progressive and "unloaded" cutting diagrams with uniform wear. (See Fig. 5.) Their wear resistance is 2-3 times higher than that of conventional hobs.

To determine the efficiency of multi-thread hobs

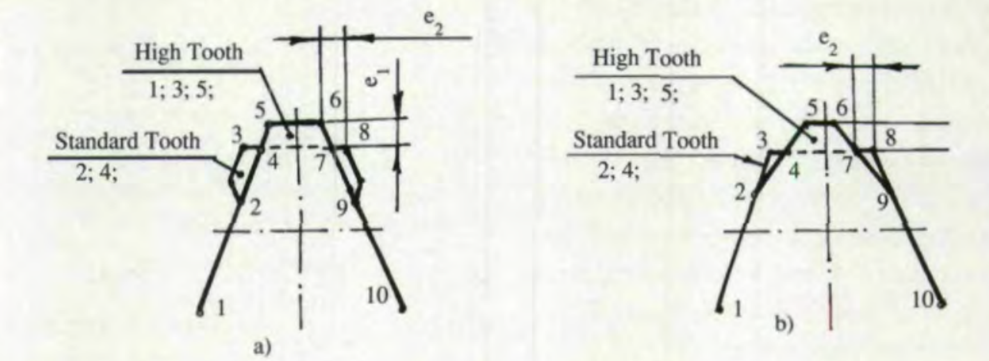


Fig. 3 - Hob with new "unloaded" diagram with protuberance (a) without protuberance (b).

with new cutting diagrams with respect to the standard hobs, wide tests were carried out (Table 2) which revealed that the substitution of standard cutting diagrams by progressive or unloaded of uniform wear increased the tool life of multi-threat hobs 2 - 3 times.

Experiments were carried out on the basis of the multi-factor method. Values of factors (x_i) vary on two levels: $x_i = +1$ and $x_i = -1$ (Table 2). Through data analysis, the following relations were received for calculation of hob wear:

standard hob (conventional hobbing)

$$h = A'_1 \cdot (V)^{1.87} (n_h)^{0.82} (T)^{0.85} (z)^{-0.35} \cdot (m)^{0.62} (f)^{[0.58+0.15 \cdot \ln(Z) - 0.34 \cdot \ln(n_h)]}, \text{ mm} \quad (5)$$

unloaded and progressive of uniform wear hob (conventional hobbing)

$$h = A'_2 \cdot (V)^{1.90} (n_h)^{0.70} (T)^{0.73} (z)^{-0.41} \cdot (m)^{0.50} (f)^{[0.26+0.15 \cdot \ln(Z) - 0.24 \cdot \ln(n_h)]}, \text{ mm} \quad (6)$$

where A_1 and A_2 are constants dependent on the type of the material under process. For example, if hob material is P9K10 and material of gears st. 40X, then $A_1 = 5.1\ 10^{-6}$, $A_2 = 7.0\ 10.$[6]

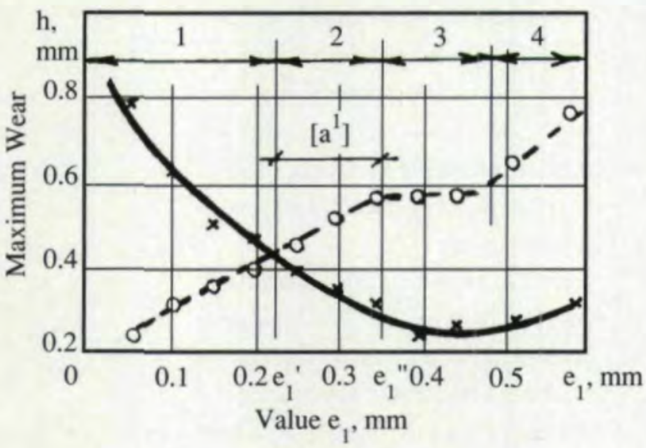
Operating capacity of hobs is characterized not only by their tool life, but also by the cutting force appearing during operation as well. Therefore, dynamic research of the work of multi-thread hobs was carried out. Both the cutting force and cutting temperature were simultaniously measured. In the course of the experiment, it was found that the process of tightness of chip formation at simultaneous work of three cutting edges (non-free cutting condition) effects not only the tool life of hobs, but also the cutting force and temperature.

The multi-factor experiment (Table 2) determined the function between the cutting force and different factors.

The cutting forces for conventional hobbing are:

standard hob

$$P_{t.max} = B_1 \cdot (V)^{-0.17} (f)^{0.78} (n_h)^{0.85} (m)^{1.40} (z)^{0.18} \quad (7)$$



Hob: m = 3.5 mm; n_h = 3; PA 20°;

Gear: z = 24; st. 40x; b = 50;

Cut. cond.: V = 54 m/min; Conventional

Fig. 4 - The influence of value e_1 on the wear of high (2) and standard (1) tooth.

progressive of uniform wear hob

$$P_{t.max}=B_2 \cdot (V)^{-0.17} (f)^{0.70} (n_h)^{0.82} (m)^{1.35} (z)^{0.15} \quad (8)$$

unloaded of uniform wear hob

$$P_{t.max}=B_2 \cdot (V)^{-0.17} (f)^{0.70} (n_h)^{0.75} (m)^{1.35} (z)^{0.15} \quad (9)$$

The application of new cutting diagrams decreases the cutting force by 20%-40% which positively effects on accuracy of gears.

Conclusions

Summarizing the above results, the following conclusions can be drawn:

(1) The relationship between hob/wear and chip flow for hobbing is obtained.

(2) Proposals were made for hobs with new cutting diagrams: unloaded uniform wear and progressive uniform wear were suggested.

(3) The experiments are carried out on the basis of the multi-factor method and the experimental relations are obtained for calculation of hob wear and cutting force when hobs with different cutting diagrams are on work.

(4) The application of new cutting diagrams cuts down hob wear 2-4 times and cutting force by 20%-40%.

Table 2 — Matrices of experiments (values of factors)

Factor	Values of factor, x_i= +1	0	−1
Cutting speed, V,m/min	75	64	54
Feed, f, mm/rev	2.0	1.4	1.0
Number of thread, n_h	1	2	3
Tool life, T, min standard hob	40	28	20
Prog. or unloaded	90	63	45
Number of gear teeth, z	48	34	24
Module, m, mm	4.25	3.5	2.5

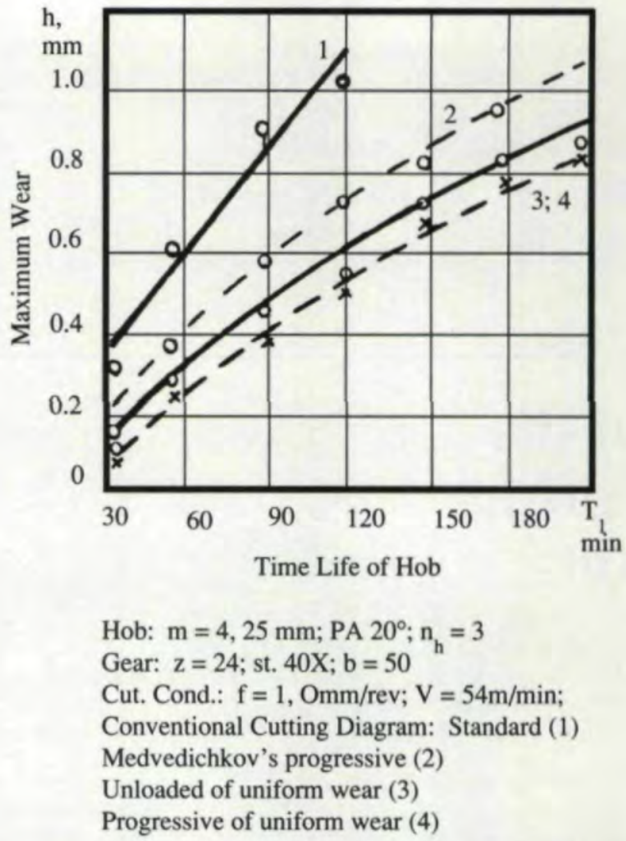


Fig. 5 - The influence of cutting diagram on the wear of the hob.

In conclusion, it should be stated that employment of multi-thread hobs with progressive and unloaded cutting diagrams of uniform wear result in an increase in hobbing efficiency.

Acknowledgement: *Our thanks to Mr. Yefim Kotlyar of American Pfauter Limited Partnership for his help with the technical editing of this article.*

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Gear Fundamentals
Reverse Engineering

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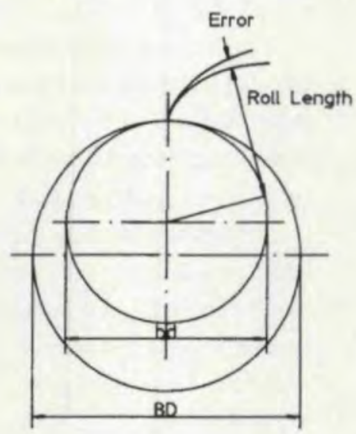
Introduction

Whether gear engineers have to replace an old gear which is worn out, find out what a gear's geometry is after heat treatment distortion, or just find out parameters of gears made by a competitor, sometimes they are challenged with a need to determine the geometry of unknown gears. Depending on the degree of accuracy required, a variety of techniques are available for determining the accuracy of an unknown gear. If a high degree of precision is important, a gear inspection device has to be used to verify the results. Frequently, several trial-and-error attempts are made before the results reach the degree of precision required.

The concept of the reverse engineering method described below is employed by some CNC gear measuring centers for automatic determination of unknown gear geometry. This article is an attempt to systematize a method for accurate determination of unknown gear geometry with or without the use of CNC inspection machinery and sophisticated software.

Concept and Requirements

The result of a gear inspection provides enough information for determining actual gear geometry characteristics. For instance, the result of a gear involute inspection can be used for determining the actual base circle diameter (Fig. 1). The actual base circle diameter is a function of involute error, length of roll relevant to the involute error, and the base circle diameter which was assumed in order to conduct the involute inspection. Similarly, the results of a gear lead inspection provide sufficient data for calculating the actual lead (Fig. 2). Thus, by making an



BD - Actual base circle diameter
bd - Assumed base circle diameter
Error - Involute error detected
Roll Length - Length of roll relevant to detected error

BD = bd * (1 + Error/Roll Length)

Fig. 1

assumption for a gear base circle diameter and lead and using the results of the inspection based on this assumption, it is possible to determine the actual base circle and lead of an unknown gear.

Every gear has a unique base circle diameter and a unique lead. However, the same gear can have numerous combinations of Normal Pressure Angle (NPA), Normal Diametral Pitch (NDP), Pitch Diameter (PD), and Helix Angle (HA), because these parameters are relevant to gear mating qualities or cutting tools. Some gear drawings contain various combinations of NPA, NDP, PD, and HA. It is important to realize that only base circle and lead are unique and can be verified. The pitch diameter at which the gear is going to be meshed with another gear and related characteristics like NDP, NPA, and HA can only be guessed, since the same gear can mesh with various gears at different pitch diameters.

The main concept of this method is that the unique gear characteristics like base circle diameter and lead are indirectly measured first by trial-and-error technique with a specified degree of precision. One or several combinations of the rest of the parameters can be computed afterwards.

The task is divided into three steps. The first step is determining gear base circle diameter. The second step is determining lead - required only for helical gears. The third step is computing of other commonly used gear characteristics based on the base circle and lead.

The first and second steps require a lead and involute measuring machine with variable base circle setting capabilities. It does not have to be a modern CNC gear checker. However, utilizing a CNC gear checker, especially one which can follow the tooth form material, makes the determination of a gear's base circle and lead easier, more accurate, and requires fewer iterations.

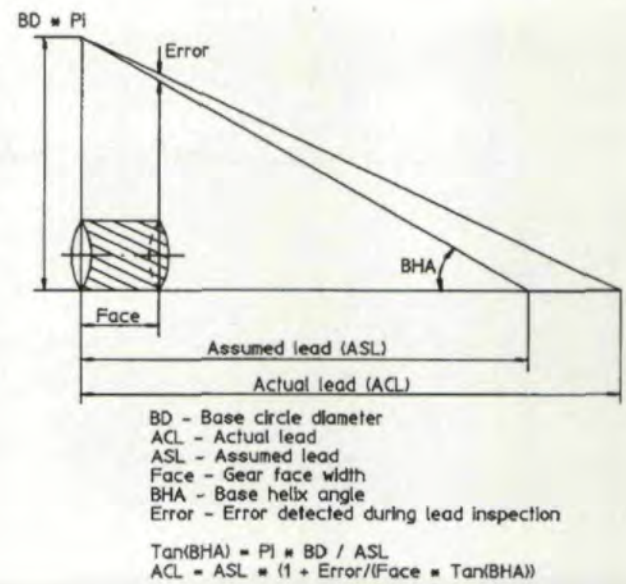


Fig. 2

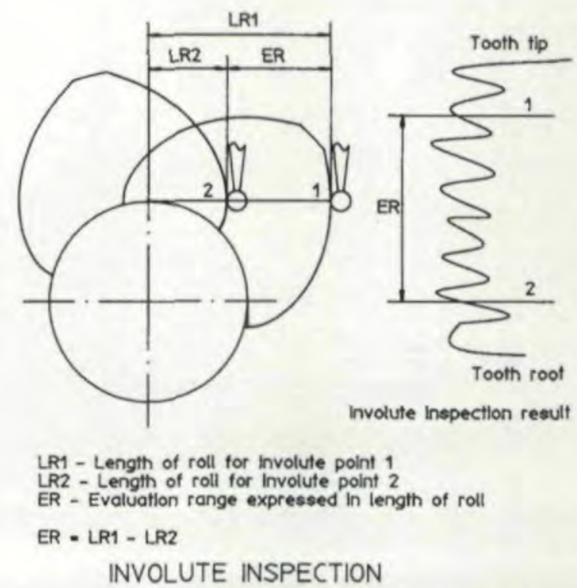


Fig. 3

Step 1 - Determination of Base Circle Diameter

Repeat Sub-steps 1.1 and 1.2 shown below until the slope error becomes smaller than the required gear accuracy. Generally, the accuracy does not have to be smaller than one micron (0.00004"). At any rate, it would be superfluous to use a number which is smaller than the accuracy limitation of the inspection machine. Because of imperfect gear surface conditions and possible inspection inaccuracies for gears with large errors, it is necessary to repeat Sub-steps 1.1 and 1.2, using a more and more accurate base circle diameter for each iteration.

The accuracy of the first assumption is not critical because every following iteration is a giant leap closer to the actual base circle diameter. Usually, no more than three iterations should be required, regardless of the first assumption. However, the base circle diameter assumption for the first iteration should be smaller or equal to the gear root diameter. Otherwise, since an involute does not exist below the base diameter, it might be impossible to make an involute inspection.

Sub-step 1.1. Assume a gear base circle (PBD) for the first iteration or use the result calculated in Sub-step 1.2 for setting up an inspection machine. Inspect the gear involute and record slope error (Slope) and evaluation range (ER). The evaluation range is the length of roll for which slope error is observed (Fig. 3). Most inspection machines provide inspection results in a scale proportional to the length of roll, and this is the unit system that should be used. For instance, if the base circle diameter and evaluation range are in inches, then the slope error should be in inches as well. Depending on whether a material plus condition is closer to the root or tip

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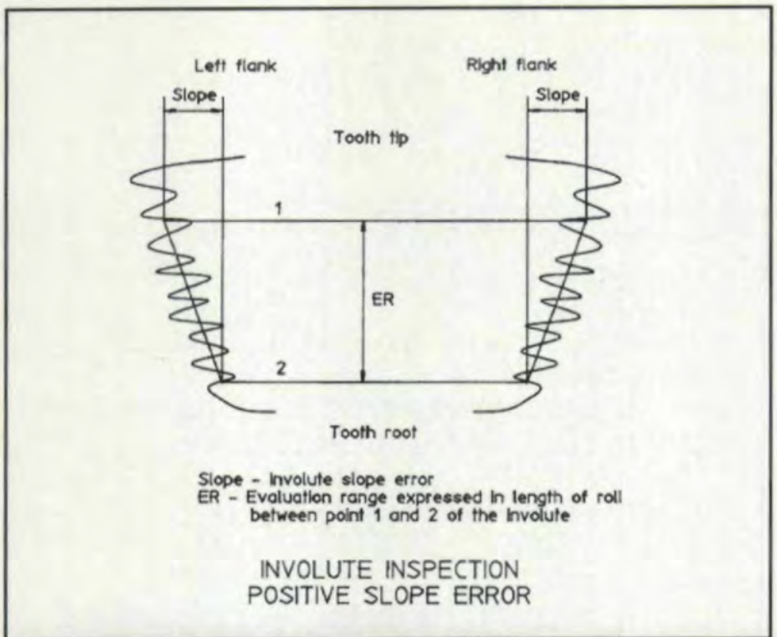

Fig. 4

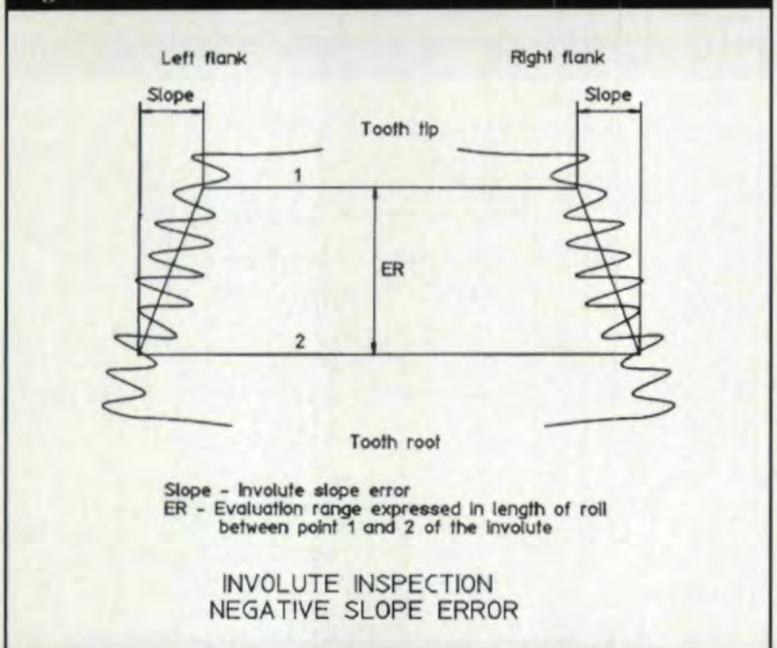

Fig. 5

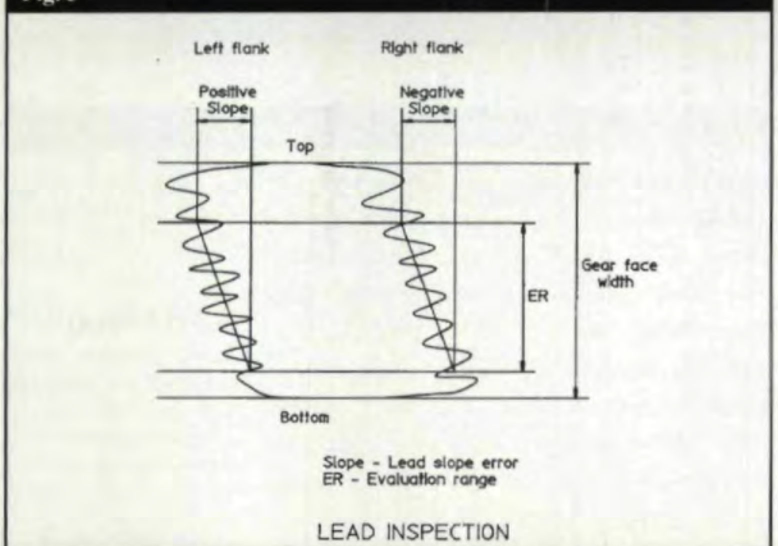

Fig. 6

of the tooth, the slope error should be positive or negative. (Figs. 4 and 5)

Sub-step 1.2. Determine a more accurate base circle diameter.

NBD = PBD * (1 + Slope/ER)

Where:

Slope = slope error recorded during inspection

ER = evaluation range

PBD = previous base demeter

NBD = new base diameter

If the slope error exceeds the required accuracy, Sub-steps 1.1 and 1.2 should be repeated using the new base circle diameter (NBD) for setting up an inspection machine.

Step 2 - Determination of Lead

(This step is not required when dealing with spur gears.)

As in Step 1, Sub-steps 2.1, and 2.2 should be repeated until the slope error becomes smaller then the required accuracy.

Sub-step 2.1. Assume a gear lead (for the first iteration) or use the result calculated in Sub-step 2.2 for setting up an inspection machine. Inspect the gear lead and record slope error (Slope) and evaluation range (ER).

The slope error is positive or negative, depending on whether a material plus condition exists on the inspected flank closer to the top or the bottom of the gear. (Fig. 6) Also note that some inspection machines require a different sign for lead depending on whether the right or left hand gear is checked.

Sub-Step 2.2. Determine a more accurate lead.

Tan (BHA) = Pi * BD / PL

NL = PL * (1 + (slope/ER)/Tan(BHA))

Where: BD = Base diameter determined in the Step 1

BHA = Base helix angle

ER = Evaluation range

NL = New lead

PL = Previous lead

If the slope error exceeds the required accuracy, repeat Sub-steps 2.1 and 2.2 using a new lead (NL) for setting up an inspection machine.

Notes for Step 1 and Step 2

If the accuracy requirements are very high, then a more precise filtering out of the surface irregularities created by enveloping cuts or feed marks can be beneficial. In this case the assis-

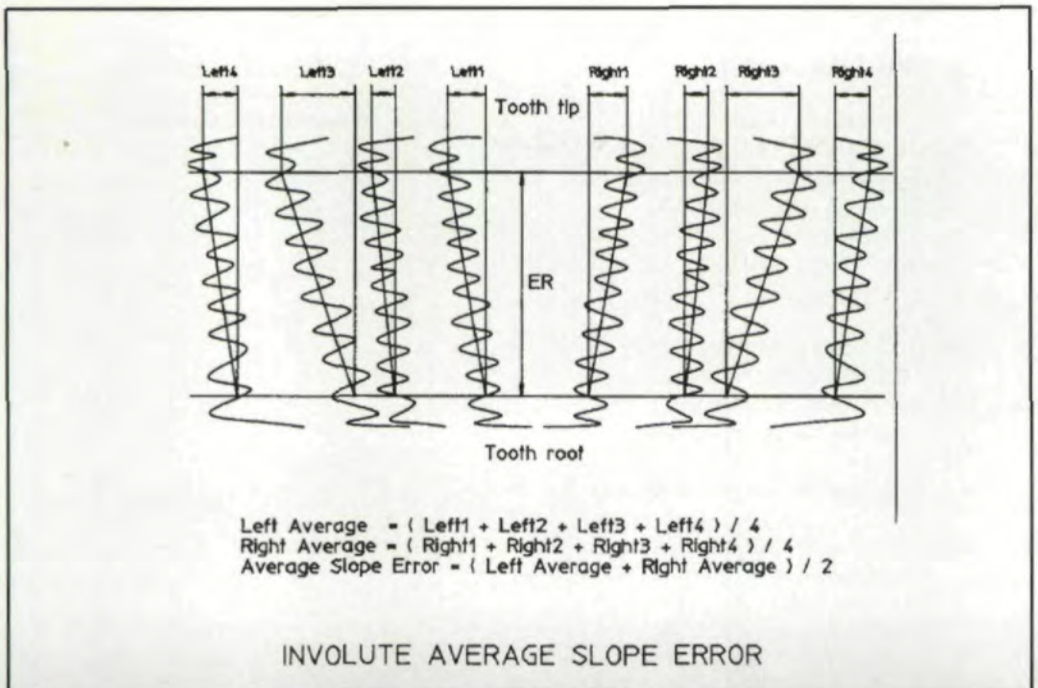

Fig. 7

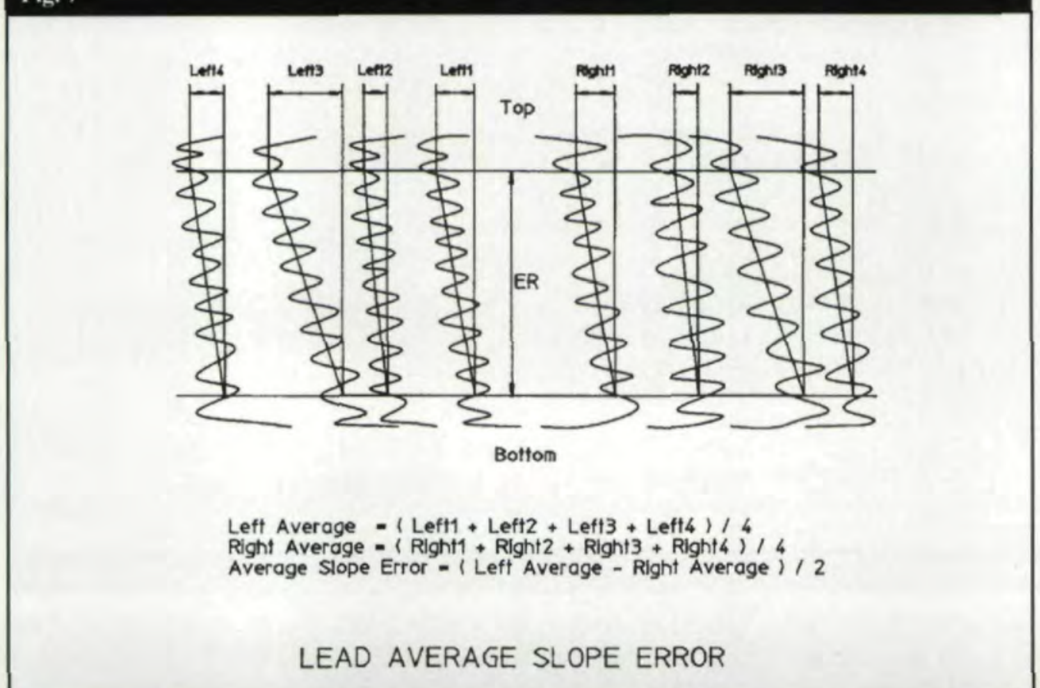

Fig. 8

tance of a computerized "least square, best fit" slope calculation can be used.

The undesirable effects of gear run-out, wobble, and taper on the results can be decreased. Involute and lead slope errors can be averaged out by checking right and left flanks as well as several equally spaced teeth. (Figs. 7 and 8) The resulting average slope error can then be applied in the Steps 1.2 and 2.2. Obviously, this would prolong the inspection time, but would also provide more accurate results.

Since most inspection machines measure lead and involute in the transverse plane (plane of gear rotation), these formulae are valid for transverse plane inspection. If an inspection machine checks a gear in the normal plane, then the slope error in the transverse plane could be calculated approximately as follows:

Transverse slope error =

Normal slope error/Cos(Helix Angle)

Another difficulty may be encountered during gear inspection. Unless an inspection machine can follow the material, it is likely that the machine will run out of probe deflection range before completion of the profile or lead inspection. However, one

Assumed Parameter	Calculated Parameters: Helical Gear	Calculated Parameters: Spur Gear
PD	HA = ArcTan (Pi * PD/Lead) NDP = Z/(PD * Cos(HA)) TPA = ArcCos (BD/PD) NPA = ArcTan (Tan(TPA) * Cos(HA)) BHA = ArcTan (Pi*BD/Lead)	NDP = Z/PD NPA = ArcCos (BD/PD)
HA	PD = Lead * Tan(HA)/Pi NDP = Z (PD * Cos(HA)) TPA = ArcCos (BD/PD) NPA = ArcTan (Tan(TPA) * Cos(HA)) BHA = ArcTan (Pi * BD/Lead)	
NPA	BHA = ArcTan (PI * BD/Lead) HA = ArcSin (Sin(BHA)/Cos(NPA) PD = Lead * Tan (HA)/Pi TPA = ArcCos (BD/PD) NDP = Z / (PD * Cos(HA))	PD = BD/ Cos (NPA) NDP = Z/PD
NDP	BHA = ArcTan (Pi * BD/Lead) HA = ArcSin (Z * Tan(BHA) (BD * NDP)) PD = Lead * Tan(HA)/Pi TPA = ArcCos (BD/PD) NPA = ArcTan (Tan(TPA) * Cos(HA))	PD = Z/NDP NPA = ArcCos (BD/PD)

Where: BD = Base Diameter determined in the Step 1
BHA = Base Helix Angle
HA = Helix Angle
Lead = Gear Lead determined in the Step 2
NPA = Normal Pressure Angle
PD = Pitch Diameter
Pi = 3.141592654
TPA = Transverse Pressure Angle
Z = Number of teeth

Table 1

(HA), and pitch diameter (PD) can be calculated. Since the same gear can have numerous combinations of these characteristics, one of them should be assumed in order to calculate the rest. Depending on whether pitch diameter, helix angle, normal pressure angle, or normal diametral pitch is assumed, one of the sets of formulae shown in Table 1 could be used in order to calculate other traditional gear characteristics.

Conclusions

The described technique assures accurate determination of a gear's actual base circle diameter and lead. Nevertheless, the results might slightly differ from the base circle and lead specified on the print because gears are manufactured imperfectly. If measuring machine inaccuracy is disregarded, the difference between the actual base circle and lead and a drawing specifications depends on how accurately the gear was manufactured.

It is also important to reiterate that various combinations of normal diametral pitch (NDP), normal pressure angle (NPA), helix angle (HA), and pitch diameter (PD) can be specified for the same gear. Thus, depending on need, selection of proper assumptions in Step 3 is important. Suppose one needs to reproduce an unknown gear and he has a stock of 20° pressure angle hobs. In such a case, it is reasonable in Step 3 to make an assumption for normal pressure angle of 20° and then calculate the rest, hoping that one of the available hobs could be utilized.

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Pineapples, Corncobs & Other Hobbing Matters

William L Janninck

Question: I have worked in the gear manufacturing department for over 15 years, and just recently someone told me about a pineapple hob. I wonder just what kind of tool this is and what it does.

"Pineapple hob" is the rather picturesque name the shop people have given to what really should be called a tapered-end, tangential-feed, worm gear hob. It is used to produce throated worm gears on a hobber equipped with a special feed slide which moves the hob tangentially to the gear, rather than in the usual axial feed motion. Standing on its big end, this multiple-start, high-lead-angle hob with its helical flutes and long slow taper, immediately reminds one of a pineapple. Hence, the name.

A lot of the names given to hobs, such as gear hob, involute spline hob, parallel key spline hob, herringbone gear hob, camshaft hob, ring gear hob, ratchet hob, or skiving hob, are properly indicative of their purpose and use. But many have other pictorially descriptive second names. For example, the term "pancake hob" usually means any narrow-faced hob where the width is much less than the diameter. A camshaft hob, which is used to cut the integral gear on an engine camshaft, can also be called a pancake hob.

What about a fly cutter? It is a tool used on a tangential feed hobber to cut or fly-out worm gear teeth. Basically, it is a hob with all but one tooth removed. Usually this cutter is composed of a body with an adjustable blade clamped in place. If it is multiple-start and has only one tooth left per start, it may also be called a pancake hob. If the hob has five or six starts and, thus, has only five or six teeth or points, it may be called a star hob. An earlier name for carbide-tipped fly cutters with only one tooth is the snail back cutter, an allusion to the long, rounded-off tooth used to back up the carbide.

The tapered end hob, the tapered hob, and the tapered root hob have similar names, but very different functions. The first is a hob with a short, tapered end used to cut helical gears. This feature is used to spread the chip load over a broadened area to reduce the danger of hob tooth overload, wear, and failure. The next is a hob tapered over its entire form and used on a hobber with oblique or diagonal feed capability to cut tapered forms, such as tapered serrations or splines. A tapered root hob has a tapered outside diameter, but the form itself is not tapered. It is used on an oblique hobber to generate parallel or involute splines where the root of the spline is conical instead of cylindrical.

"Corncob hob" is another coined name for extra long, but rather small diameter gear hobs. These were developed as part of a plan to increase gear productivity in high-speed hobbing by increasing hob rotational and indexing speed and by having a long useable hob face for long in-machine time. They are usually coated with bright yellow titanium nitride. A quick

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"Shop Floor" is your column. When you have questions you can't answer or need information you can't seem to find, let our panel of experts help. They have a total of more than 120 years of gearing experience. Address your questions to them, care of Shop Floor, Gear Technology, P.O. Box 1426, Elk Grove Village, IL 60009, or call our editorial staff at (708) 437-6604.

William L Janninck

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glance immediately reveals the corncob connection.

Some hob names, such as "convolute hob", are not exactly descriptive. This name is applied to a single convolution hob that is used for cutting face gears and face-serrated couplings.

The thread hob and rack hob are not hobs at all, but are form milling, non-generating cutters used for machining screw threads or rack gear segments. Their names are derived from their appearance. From a short distance they look just like gear hobs.

Question: When setting up a gear hobbing machine, we on the shop floor are told to be sure to set the hob swivel angle as exactly as we can. Does this setting really have to be so precise? What happens if it is off a bit, say some 15 or 20 minutes? We are sure we have run some hobs that were off on this setting, and we never seemed to have any subsequent problems.

The answer depends on whether you're cutting cylindrical gears or worm gears.

Cylindrical gears are much more common. These can be separated into two discrete types, spur and helical.

The hobs for gears will have a lead angle marked on the hub end or body for set-up purposes. When cutting a spur gear, the hob swivel is set to this angle, so the thread of the hob will line up with the gear tooth normal section. On a spur gear the teeth are parallel to the gear axis. On many manual machines, the best one can set the machine is plus or minus about five minutes, based on a vernier scale. Of course, on occasion the true zero setting on the machine should be verified for alignment by use of master arbors and swing indicators.

Regardless of the accuracy of the setting angle, the hob still performs well as a cutting tool, properly forming chips and producing gear teeth. Adjusting the swing angle will not change the profile produced; the involute cut will be correct even with an incorrect setting angle.

To prove this, we conducted tests, supported by mathematical computer analysis, using a 4DP 20 PA Class A hob, where we cut five test gears using the same hob. The hob swivel was first set at the proper angle, and a gear cut and then offset by 1-degree intervals up to 4-degree swivel angle error. All five gears checked correctly on the involute produced, with the exception that the fillet radius became larger as the set error went up, and the radius to the involute starting point also went up. Since we maintained the center distance between hob and gears (that is, we kept a constant gear root diameter) we found that the hob swept out a wider space and reduced the gear tooth thickness by .005 inches per one degree of swivel setting error. Backwards interpolation implies that a swivel setting error of five to ten minutes in general is sufficiently close.

When considering the cutting of helical gears, the same results apply. However, the correct swivel angle for like-hand hobs and gears is the gear helix minus the hob lead angle. But one must be careful if a short lead hob is used, since in this circumstance, the hob setting is the helix at the generating circle minus the hob lead angle. Since short lead hobs are usually single-purpose, the hob lead angle as well as the proper swivel setting angle are both marked on the hob.

On occasion a slight swivel angle adjustment is used on purpose to make a hob cut a slightly wider space than the hob was designed for and yet hold the over-pin measurement and root diameter relative.

When considering the cutting of throated worm gears, the swivel setting angle is of much higher importance, and this is especially so on the higher lead angle jobs, heavily loaded drives, or precision sets. Typically these sets will use hobs with smaller oversize, and the less the oversize, the more sensitive the setting angle. However, most single-start sets tolerate more oversize and are less sensitive than multiple starts. On many of the sensitive sets, five minutes of accuracy in the set angle may be insufficient, and a pin-and-dial indicator may be used to make positive small adjustments. Although a correct zero point is not necessary, because you can make moves relative to the last position, a correct zero point is recommended. On worm gears, where the contact pattern and the gear set axis angle is measured on a separate inspection machine, the information is fed back to the gear hobber as small swivel adjustment using the dial indicator.

Some time ago, we were assigned to

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The swivel setting angle is of much greater importance on high lead angle jobs, heavily loaded drives, or precision sets.

design the hobs necessary for a new style of very simple spur gear hobber. Besides cutting only spurs and moderate pitches, it was to have no hob swivel, and the table was to be fixed at a zero angle, thereby eliminating all the extra components involved. This complicated design procedure for the hob was required in order that a zero set angle would be maintained. We were successful on the tooling for a few parts which were tested, but the inability to design tools to meet certain other part specifications caused the project to be stopped. ■

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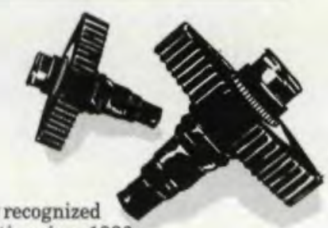

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CALENDAR

AGMA Events

For more information contact AGMA headquarters, 1500 King St., Suite 201, Alexandria, VA 22314. Phone: (703) 684-0211.

OCTOBER 21-23

Gear Expo '91. The World of Gearing.

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Fall Technical Meeting. Held with Gear Expo '91.

Society of Manufacturing Engineers Technical Conferences and Events

For more information, contact Mike Traicoff, SME, One SME Drive, P.O. Box 930, Dearborn, MI 48121-0930. Phone: (313) 271-1500. Fax: (313) 271-2861.

SEPTEMBER 17-19

Advanced Gear Processing and Manufacturing. Holiday Inn Airport. Indianapolis, IN.

OCTOBER 14-16

Grinding Technology. Detroit, MI.

Other Events

SEPTEMBER 10-12

Ohio State University. Three-day course on gear noise covers general noise measurement and analysis, causes of gear noise, gear noise reduction, dynamic modeling, signal analysis, and other gear noise issues. For more information, contact Miss Carol J. Bird, OSU, (614) 292-3204.

SEPTEMBER 13-16

ASME 6th International Power Transmission and Gearing Conference. Marriott Camelback Resort. Phoenix, AZ. Contact: Allen G. Strandford, Jr., Conference Chairman, Dresser Rand, P.O. Box 560, Olean, NY 14760. Phone: (716) 375-3285. Fax: (716) 375-3715.

OCTOBER 22-24

American Society for Metals 13th Heat Treating Conference and Exposition. Cincinnati Convention Center. Cincinnati, OH. For further details write to: ASM International, Member Activities Dept., P.O. Box 473, Novelty, OH 44072-9901.

CNC Wheel Profiling in Gear Grinding Applications

In 1984, Normac, Inc. introduced the FORMASTER CNC Grinding Wheel profiler, a CNC dressing device for retrofit to existing grinding machines. Because of its superior accuracy, ease of installation, and relatively low price, the FORMASTER gained rapid acceptance in a wide variety of industries and applications. Gear grinding is among the most successful and advantageous applications of the FORMASTER, including gear grinding machine builders that are incorporating Normac's FORMASTER in their new machines as well as retrofitting older equipment with new technology.

As a result of the gear industry's need to more accurately profile wheels on both new and older grinding equipment, Normac now offers a complete wheel dressing retrofit system for parallel axis gear grinding. This package can be installed on nearly any grinder, and will profile both aluminum oxide and dressable CBN grinding wheels. Included are the FORMASTER dresser, the computer numerical control system, and off-line program generation software for both external and internal involute gears.

The FORMASTER is capable of dressing virtually any form if it is provided with the proper data. Obtaining this data can be a tedious, if not nearly impossible, task. To enhance the FORMASTER'S value to gear producers, Normac has developed application software that runs on an IBM-PC compatible computer that completely removes the burden of complicated math calculations and NC programming. The user is prompted by the program to input various gear data (i.e. diametral pitch, number of teeth, etc.) in conventional gear manufacturing terminology. The software then converts these inputs into the data required by the FORMASTER in the form of an NC program. This program can be saved for future use or sent by wire connection to the FORMASTER'S control system for immediate use. In addition to computing a true involute profile, this software also enables flexible root zone and protuberance profiling. The user can also define in great detail the specialized geometries of root and tip modifications, true involute modifications, multiple break points for varying modification amounts, complete "barrel" shaped profiles, and exponential modification.

The application of CNC wheel profiling offers some major advantages to gear manufacturers including dramatically reduced grinding costs, increased productivity, and product quality improvements. The application of this technology is relatively simple and fast to accomplish; installation and training are usually completed in less than a week. Because of the major reduction in grinding costs realized with the FORMASTER, the payback period on the equipment investment is surprisingly short; in some cases as little as two or three months!

To get more information about the FORMASTER, or to arrange a demonstration, contact: Normac, Inc., P.O. Box 69, Arden, N.C. 28704; phone: (704) 684-1002; fax: (704) 684-1384 or Normac, Inc., P.O. Box 207, Northville, MI 48167; phone: (313) 349-2644; fax: (313) 349-1440.