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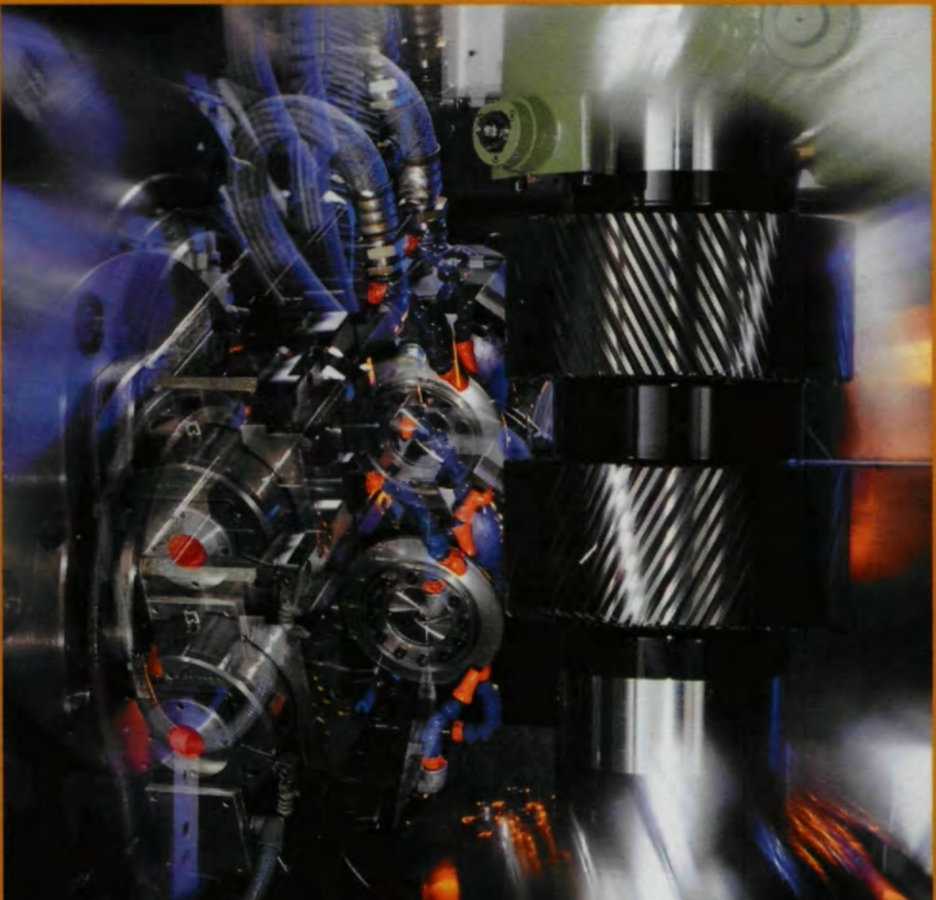
THE JOURNAL OF GEAR MANUFACTURING

MAY/JUNE 1991



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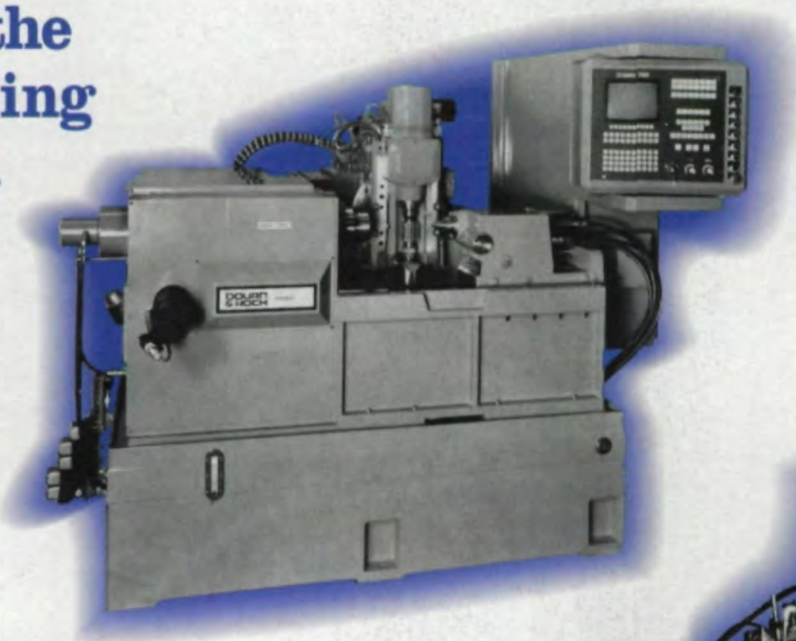

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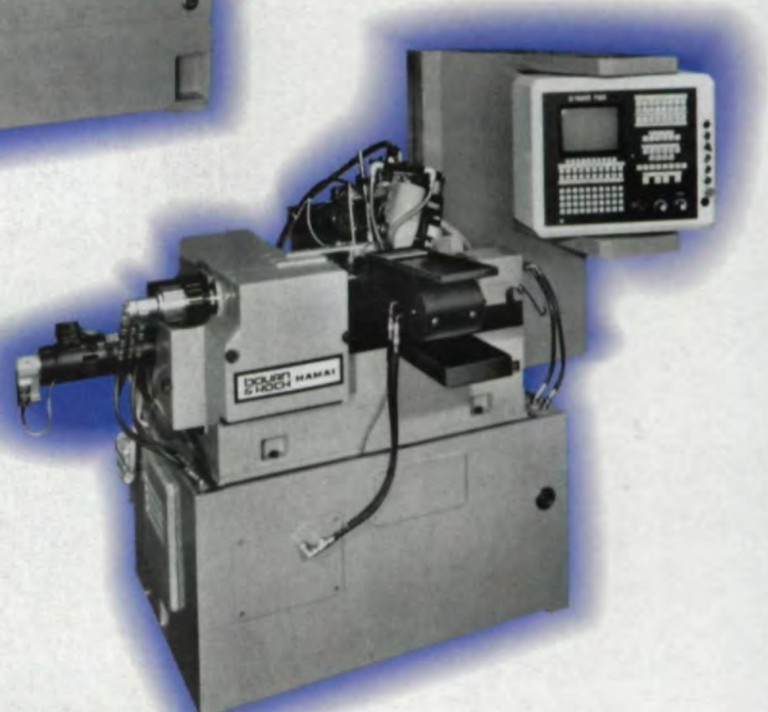

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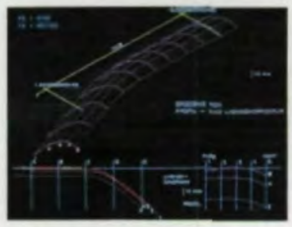

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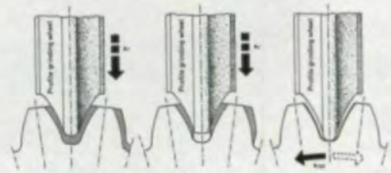

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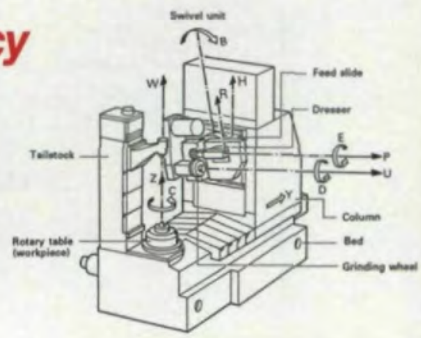

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A United Europe Will Be A Long Time Coming

PUBLISHER'S PAGE

Pride. Awe. Relief. Admiration. These were some of the emotions with which I, like most Americans, greeted the end of the Persian Gulf War. I was proud of our country for saying it would do a job and then doing it with a minimum of loss and a maximum of effectiveness; I was awed by the terrifying efficiency of our weapons and relieved that our casualties were so light; and I was filled with admiration at the skill with which one of the most complex logistical military operations of the century was carried out.

Over the last twenty-five years our military had developed a reputation for foul-ups that became defining examples of the word "snafu," but it seems the lessons of past disasters have been well-learned. Most important this time, there was one clearly designated leader, as well as one clearly stated goal, and one basic rule: once the balloon went up, the special interests and agendas of individual service branches and differing methods of carrying out the plan were put aside. Everyone had the same goals and was listening to only one commander. The results were nothing short of spectacular.

The response of our neighbors in the European Community was less clear cut. By all accounts, the troops sent from the various European countries performed admirably; it was the tardy, vacillating response of the countries themselves that has been the subject of criticism and anxiety on both sides of the Atlantic. People who had seen the events of the last two years as the harbinger of a united Europe that would speak with one voice politically and militarily were sorely disappointed. When push came to shove, even over an issue as economically important to Europe as securing the major source of its petroleum supply, the twelve members of the EEC could not find a common voice or a common will, much less a single leader. They had similar goals, but no unified agenda for reaching them.

Whether that is a tragedy or not is a matter of opinion. The fact remains that a united Europe is not here now, nor will it be any time soon; and, perhaps, neither we nor our European neighbors should be too anxious over the fact.

Building an economically and politically united Europe may be an admirable goal, but it will take time. American history could be instructive here. Starting from scratch, with a common language and without generations of past conflict to live down, it took the United States eighty years, culminating in five years of civil war, to decide that we would be one nation speaking with one voice, not just a collection of individual states and regional interests. Real unity will not come any easier in Europe.

The fact is that a united Europe will probably not be built until some event of cataclysmic proportions forces all the nations of the continent to put aside their individual differences and act together. If the threat to its basic energy supply didn't do it, one shudders to think what will.

At present, the individual nations of Europe apparently lack the motivation or the will to unite completely. What is possible is some kind of economic cooperation. More unity than that will be a long time coming. Being anxious or frustrated with one another over it will not change things.

For the foreseeable future, the international community will have to deal with the complexities inherent in a multinational Europe and remember that nothing is ever gained by forcing a good idea before its time.

Michael Goldstein

Michael Goldstein,
Publisher/Editor-in-Chief

Misalignment No Beauty in Gearsets

Don McVittie

SHOP FLOOR

"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, with more than 120 years of gear experience among them, help. 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.

When we have problems with gearset failure, a common diagnosis is misalignment." What exactly is that and how do we prevent it?

The second most common "killer" of good gear sets is misalignment (dirt, or abrasive wear, is first). Gear teeth simply won't carry the load if they don't touch, and the portion that does touch has to carry an overload to make up for the missing contact area.

Fig. 1 shows the effect of one kind of misalignment - deflection of an overhung pinion. This example could be from any drive where both pinion bearings are on the same side of the gear. When power is applied, the pinion shaft bends and its bearings deflect, so the pinion teeth aren't parallel to the gear teeth. (See Fig. 2.) The deflected pinion is heavily loaded at the unsupported end. The load may not even cover the whole pinion face.

The heavily loaded end usually can't take the abuse, so it wears, pits, scuffs, cold-flows, or breaks. By the time it wears enough to allow the other end to carry some load, the gear is badly damaged. Wear progresses until teeth break, because there's no self-healing effect.

Some things that will help are:

•Tightening up the pinion bearings if they're adjustable.

•Reinforcing the pinion bearing support structure by adding extra bracing.

•Re-aligning the pinion to tip the free end toward the gear, so it's misaligned under light loads and bends into full-face contact as the load is applied. This works best if the load is always applied in one direction.

•Checking backlash at all positions of

Fig. 1 - Uneven wear on an overhung pinion. The stub of the broken shaft is still in the bore.

Don McVittie
*is President of Gear Engineers, Inc., Seattle, WA. He is a past president of AGMA and Chairman of the U.S. Technical Advisory Group for International Gear Standards. He is a licensed professional engineer in the State of Washington. If you have questions for Mr. McVittie, please circle **Reader Service No. 36***

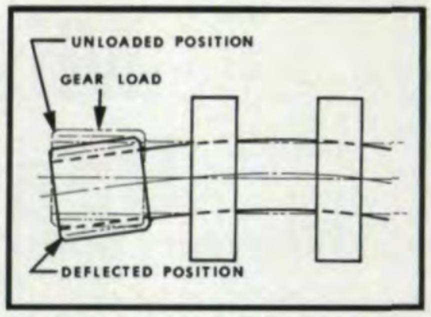

***Fig. 2** - Deflection caused by gear misalignment.*

the bull gear. Sometimes the bull gear runs out enough to bottom out with the backlash. This causes severe overload and aggravates the misalignment problem; it can even break the pinion shaft.

•Re-working the drive to mount the pinion on a loose spline, so it can find its own alignment even when the shaft bends.

•Making a tapered pinion, carefully calculated to match the slope of the deflected shaft under load. It should probably be crowned too, because it can only be a perfect match at one load level.

Misalignment of an overhung pinion is easy to understand, but sometimes we see signs of misalignment even though the gear and the pinion are both supported between bearings. Fig. 4 shows a pitting failure extending only half way across the tooth face - a sure sign of misalignment under load. What caused

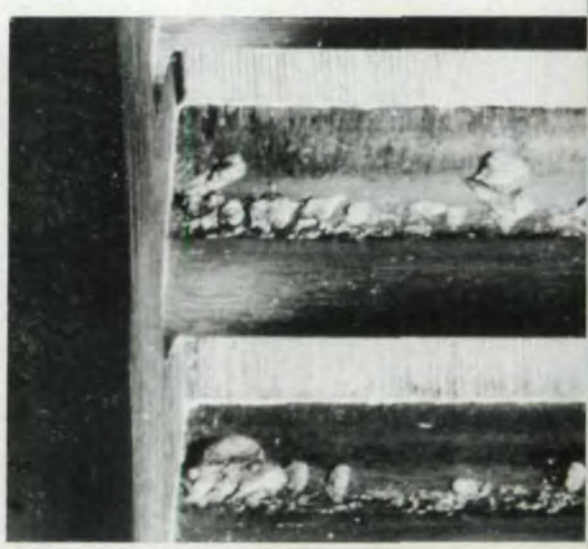

***Fig. 4** - Pitting failure caused by misalignment under load.*

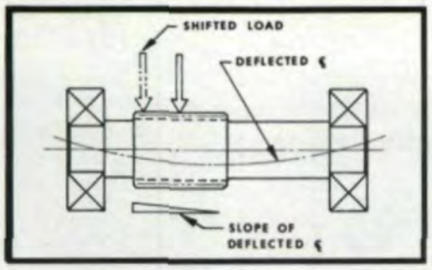

Fig. 3 - Bending deflection of the pinion shaft.

the misalignment? The key lies in the words "under load."

Here are some possibilities:

•Bending of the pinion shaft. If the center of the pinion is not in the center of the bearing span, any bending deflection

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of the pinion shaft will cause a slope, forcing the load toward the end of the pinion closest to a bearing. Fig. 3 illustrates this.

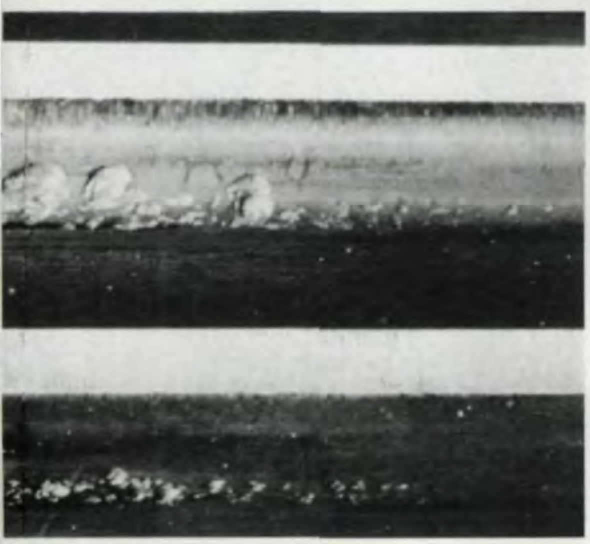
•Twisting of the pinion shaft. Heavily loaded, slender pinions "wind up" under load, forcing the load toward the end of the pinion closest to the input.

•Poor adjustment of tapered roller bearings.

•Bearing wear.

•Case deflection due to poor foundation or heavy load.

•Manufacturing error in the case or gears.



OOPS! In last issue's article, "Hard Finishing by Conventional Generating and Form Grinding," the name of Niles Grinders was eliminated from the list of formed-wheel gear grinding machines. These machines are most suitable for one off to medium batch production runs. They offer the capability of grinding gears with different pressure angles, addendum and modifications, tip and root relief as well as lead crowning without the requirement of additional equipment such as index plates, pitch blocks or special grinding wheels. The machines cover a working range with maximum outside diameters of 24" to 157" and maximum table loads of 880 lbs. to 88,000 lbs. Today, more than 5,000 Niles gear generating grinding machines have been installed throughout the world. We apologize for omitting Niles from the list.

***Fig. 5** - Bending failure caused by a misaligned gear set.*

In any case, the misalignment must be fixed. If it is allowed to persist, the teeth will fail. Fig. 5 shows a typical bending failure which resulted from failure to correct a misaligned gear set.

We should be aware that even parts supported between bearings can be misaligned under load. Fig. 6 shows a classic case. This new gear box developed problems during an overload test before going into service, but it could have happened at any time.

The diagram (Fig. 7) shows the bearing arrangement. Two tapered roller bearings, one at each side of the box, were expected to carry the radial load and the thrust load induced by the helical gear teeth.

***Fig. 6** - This new gear box failed because of misalignment even though it was supported between bearings.*

Unfortunately, a careless assembler didn't get the cone of one bearing seated all the way against its shoulder. A simple check with a feeder gauge between the back of the cone and the shoulder would have shown a gap, but the check was overlooked. The press fit held the cone in place for a while, but an overload moved it to the shoulder, creating a gap between cup and cone. A loose adjustment would have had the same effect.

Once the bearings were loose, the

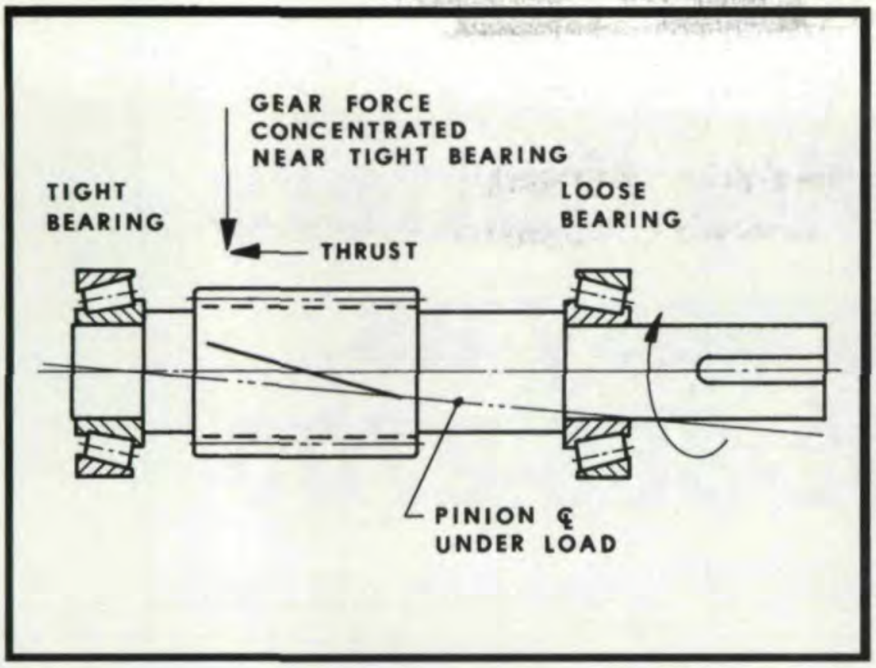


Fig. 7 - The bearing arrangement of the gearbox in Fig. 6., showing the cause of the failure.

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thrust from the helical teeth pushed the pinion tightly into one cup, as shown in Fig. 7. The other bearing was then loose, so its cone shifted laterally away from the load, leaving the pinion misaligned. The pinion teeth at the end with the tight bearing took all the load - until they began to pit.

Careful assembly will prevent this problem, if the gear case is stiff enough to support the bearings. Here are some easy tips to help you get it right:

•Be sure the cones are seated against the shoulders on the shaft; use a .001" feeler gauge between the back of the cone and the shoulder to check. When possible, press the bearing into place instead of heating it. It can't shrink back from the shoulder if it's pressed against it.

Fig. 8 - The pitting pattern on this helical pinion is the result of a heavy load at one end caused by misalignment.

•Be sure cups are seated too. A stray burr can keep the cup from seating and leave room for movement when the load is applied.

•Be extra careful shimming gear box bearings. They really should run at no clearance, but that may be too tight to run cool. One to three thousandths is usually enough clearance.

•Check bearing temperatures and gear tooth contact patterns after a few hours under load to be sure everything is as you left it.

Sometimes there simply is no easy fix for a misalignment problem. Maybe the box is bored out of line, the gears weren't matched originally, or the parts are so slender that they bend under load. In any case, the load is so heavy at one end of the teeth that wear is unacceptable. Fig. 8 shows this kind of pitting pattern on a helical pinion.

If it can be removed, the whole gear box should be sent to a good gear shop. There the cause of the bad contact can be found and the part(s) causing the misalignment reworked.

The gears will first be checked on true center distance in a test fixture as illustrated in Fig. 9. If the contact pattern in the fixture looks good, but the contact under load is bad, the box will be checked to make sure that the bores are parallel. The theoretical bending and twisting deflections are calculated to estimate the amount of deflection under load.

With this information, a correction can be made to the helix angle of the gear or pinion, creating an intentional mismatch at no load. When the gears wind up under load, they match and their capacity is increased. If they have to run at a variety of loads, the required tooth form can be curved or "crowned."

Sometimes the gear box can't be removed, so only the gears are available for checking. The repair is still possible, using the test fixture to simulate the gear box, but it's difficult to know if the fault lies in the gears, the box, the deflection, or all three. If only the gears are available for a modification, it may take more than one try to get it right.

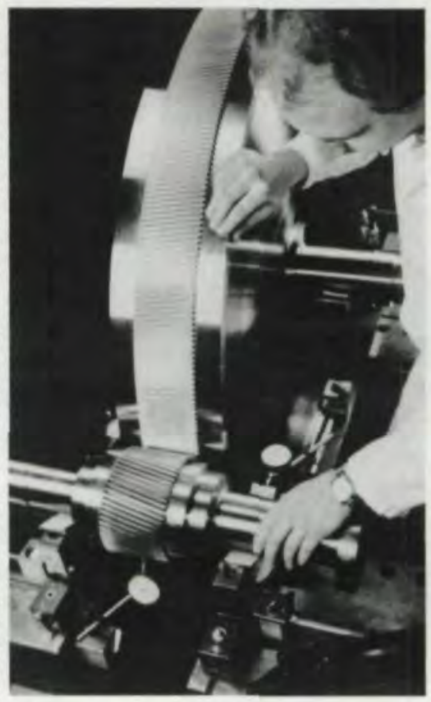

Fig. 9 - A contact check in a test fixture is a crucial step in correcting misalignment.

This article originally appeared in consecutive issues of the "Tooth Tips" column of Pitch Lines, a newsletter of The Gear Works, Seattle, WA. Thanks to Mr. Roland Ramberg, president of Gear Works, for allowing us to reprint them.

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PART DATA

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Pitch ☐	Module ☐		Meas. over/between Pins
	Diametral		Pin Diameter
	Normal		Arc Tooth Width
	Pressure Angle		Arc Space Width
Diametral ☐	Normal ☐		Machine Model No.
	Helix Angle (Part)		Serial No.
Hand Right ☐	Left ☐	Taper Shank ☐ Disc ☐	Deep C'Bore ☐
	Inside/Outside Dia.		Approx. P.D. Cutter
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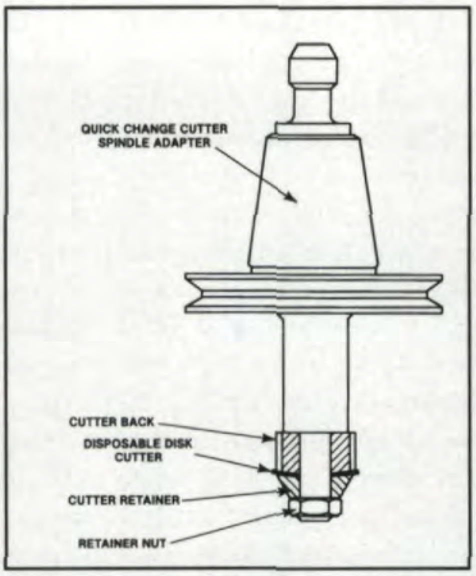
Shank Type Disposable Disk Cutters



In the past few years, gear manufacturers have begun to see the advantages in the disposable disk gear shaper cutting tools. The most apparent time and money saver, is the time no longer spent on machine setups. Disposable disks are manufactured perfectly flat and are designed to eliminate the re-sharpening process. The consistency in cutter dimensions is translated to consistent part uniformity. As the cutting tools are changed and no sharpening has taken place, machine setups can be left unchanged from cutter to cutter.

Fellows Corp. has introduced an additional advantage to this concept. By deflecting the thin disposable disk into a negative rake angle, both sides of the disk become identical cutting surfaces. The negative deflection provides the cutting disk with its own top rake and side clearance angles. When one side becomes dull, the disk can be turned over and a new cutting surface is available.

The majority of the disposable disk cutters presently in the field have been in the 3" PD and up range. Fellows has introduced a shank type version of the disposable cutting tool, which includes pitch diameters down to approximately .75 inches. The example illustrated here is a shank type disposable cutter assembly which is integrated with the quick change cutter spindle adapter. The disposable disk is designed for a 16/32 DP, 30° pressure angle spline and designed specifically to cut to minimum root diameter and maximum tooth thickness. The consistent cutter geometry allows this to be possible. Not only are specific dimensions being cut, but also the inside diameter of the spline is being topped by the disk cutter as well. This process has saved the manufacturer considerable time and expense by eliminating one complete operation.

Production for this part, cutting cast iron, has been between 200 and 400 parts per cutting surface. Double that production for an amazing 400 to 800 parts *cut with a .030 inch thick cutting* disk.

The Lubrication of Gears - Part II

Robert Errichello
GEARTECH, Albany, CA

Introduction

What follows is Part 2 of a three-part article covering the principles of gear lubrication. Part 2 gives an equation for calculating the lubricant film thickness, which determines whether the gears operate in the boundary, elastohydrodynamic, or full-film lubrication regime. An equation for Blok's flash temperature, which is used for predicting the risk of scuffing, is also given.

Elastohydrodynamic Lubrication

Gear teeth are subjected to enormous contact pressures on the order of the ultimate tensile strength of hardened steel, yet they are quite successfully lubricated with oil films that are less than one micrometer thick. This is possible because a fortuitous property of lubricants causes their viscosity to increase dramatically with increased pressure. Fig. 1 depicts the region of contact between mating gear teeth. It shows the shape of the elastically deformed teeth and the pressure distribution developed within the contact zone. The molecular adsorption of the lubricant onto the gear tooth surfaces causes it to be dragged into the inlet region of the contact, where its pressure is increased due to the convergence of the tooth surfaces. The viscosity increase of the lubricant caused by the increasing pressure helps to entrain the lubricant into the contact zone. Once it is within the high pressure, Hertzian region of the contact, the lubricant cannot escape because its viscosity has increased to the extent where the lubricant is virtually a rigid solid.

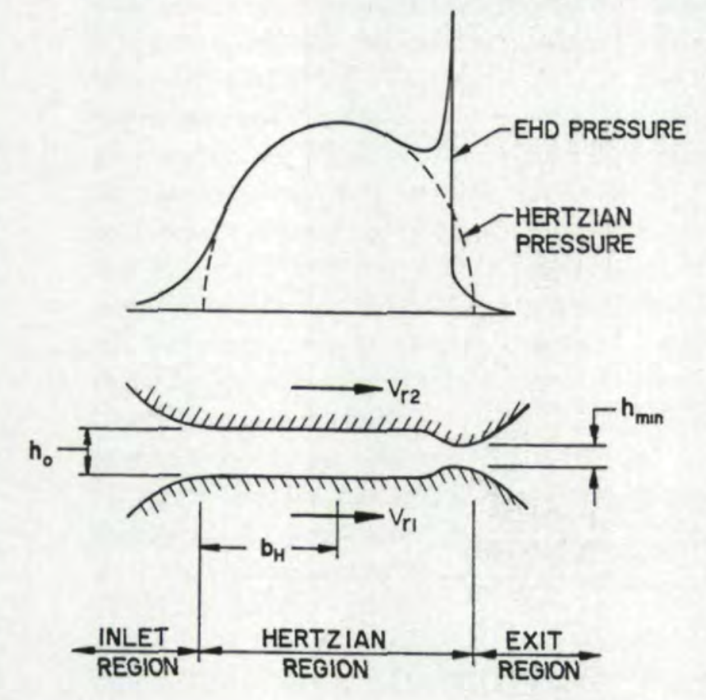


Fig. 1

The following equation, from Dowson and Higginson[1] gives the miniumum film thickness that occurs near the exit of the contact.

Minimum film thickness:

$$hmin = \frac{1.63\alpha^{0.54}(\mu_o V_e)^{0.7}\rho_n^{0.43}}{(X_r w_{Nr})^{0.13} Er^{0.03}}$$

The specific film thickness is given by

$$\lambda = \frac{hmin}{\sigma}$$

where

σ = composite surface roughness

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

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

μ_0 = absolute viscosity, Reyns (lb sec/ in^2) Fig. 2 gives average values of viscosity versus temperature for typical mineral gear lubricants with viscosity index of 95.

α = pressure-viscosity coefficient, (in^2/ lb). The pressure-viscosity coefficient ranges from $\alpha = 0.5 \times 10^{-4}$ to $\alpha = 2 \times 10^{-4}$ in^2/lb for typical gear lubricants. Data for pressure-viscosity coefficients versus temperture for typical gear lubricants are given in Fig. 3.

Er = reduced modulus of elasticity given by

$$Er = 2\left(\frac{1-\nu_1^{\,2}}{E_1} + \frac{1-\nu_2^{\,2}}{E_2}\right)^{-1}$$

where

ν_1, ν_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 (pinion, gear)

ψ_b = base helix angle

Ve = entraining velocity given by

$Ve = Vr_1 + Vr_2$

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

$$W_{Nr} = \frac{W_{Nr}}{Lmin}$$

where

W_{Nr} = normal operating load

Lmin = minimum contact length

Load Sharing Factor, X_r

The load sharing factor accounts for load sharing between succeeding pairs of teeth as influenced by profile modification (tip and/or

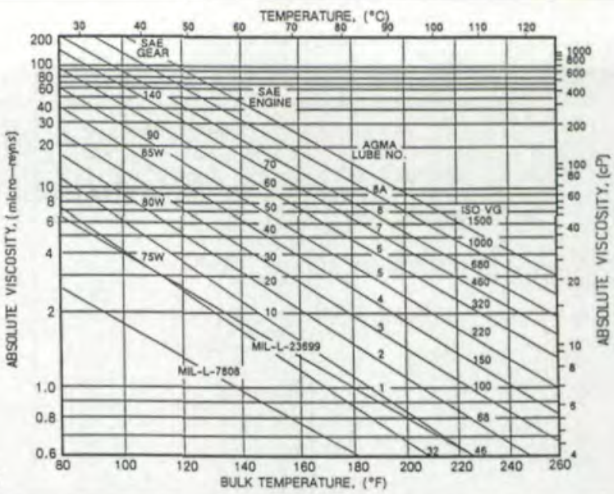


Fig. 2

root relief) and whether the pinion or gear is the driver. Fig. 4 gives plots of the load sharing factors for unmodified and modified tooth profiles.

As shown by the exponents in the Dowson and Higginson equation, the film thickness is essentially determined by the entraining velocity, lubricant viscosity, and pressure-viscosity coefficient, while the elastic properties of the gear teeth and the load have relatively small influences. In effect, the relatively high stiffness of the oil film makes it insensitive to load, and an increase in load simply increases the elastic deformation of the tooth surfaces and widens the contact area, rather than decreasing the film thickness.

Blok's Contact Temperature

Blok's[2] contact temperature theory states that scuffing will occur in gear teeth that are sliding under boundary-lubricated conditions when the maximum contact temperature of the gear teeth reaches a critical magnitude. The contact temperature is the sum of two components, the bulk

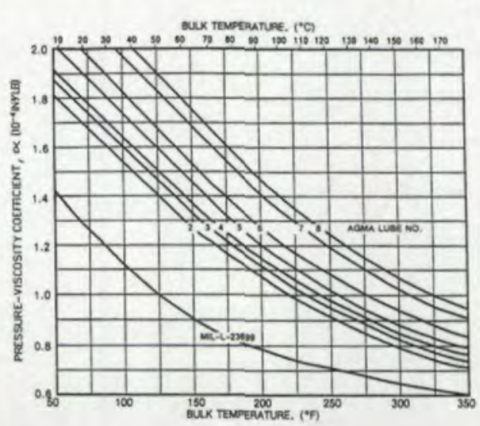


Fig. 3

Robert Errichello

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Nomenclature Table

Symbol	Description	Units	Symbol	Description	Units
B_M	-thermal contact coefficient	1bf/[ins$^{0.5}$°F]	Ve	-entraining velocity	in/s
b_H	-semi-width of Hertzian contact band	in	Vr_1,Vr_2	-rolling velocity (pinion,gear)	in/s
c	-constant (See Table 3.)	hp/gpm	W_{Nr}	-normal operating load	lbf
c_M	-specific heat per unit mass	1bf in/[lb °F]	w_{Nr}	-normal unit load	lbf/in
d	-operating pitch diameter of pinion	in	Xw	-welding factor	--
E_1,E_2	-modulus of elasticity (pinion, gear)	lbf/in^2	X_r	-load sharing factor	--
Er	-reduced modulus of elasticity	lbf/in^2	α	-pressure-viscosity coefficient	in^2/1b^2
h_{min}	-minimum film thickness	in	λ	-specific film thickness	--
L_{min}	-minimum contact length	in	λ_M	-heat conductivity	lbf/[s °F]
n	-pinion speed	rpm	μ_m	-mean coefficient of friction	--
P	-transmitted power	hp	μ_o	-absolute viscosity	Reyns (lbs/in^2)
q	-oil flow rate	gpm	$\nu_1\nu_2$	-Poisson's ratio (pinion, gear)	--
S	-average surface roughness, rms	μin	ν_{40}	-kinematic viscosity of 40°C	cSt
Tb	-bulk temperature	°F	$\rho_1\rho_2$	-transverse radius of curvature (pinion, gear)	in
Tb,test	-bulk temperature of test gears	°F	ρ_M	-density	lb/in^3
Tc	-contact temperature	°F	ρ_n	-normal relative radius of curvature	in
Tf	-flash temperature	°F	σ	-composite surface roughness, rms	μin
Tf, test	-maximum flash temperature of test gears	°F	σ_1,σ_2	-surface roughness, rms (pinion, gear)	μin
Ts	-scuffing temperature	°F	ψ_b	-base helix angle	deg
V	-operating pitch line velocity	ft/min	ω_1,ω_2	-angular velocity (pinion, gear)	rad/s

perature of the gear teeth reaches a critical magnitude. The contact temperature is the sum of two components, the bulk temperature and the flash temperature; i.e., Tc = Tb + Tf.

Blok's flash temperature equation as formulated in AGMA 2001-B88, Appendix A[3] for spur and helical gears is

$$Tf = \frac{0.8\mu_m X_r w_{Nr} \left| (V_{r1})^{0.5} - (V_{r2})^{0.5} \right|}{B_M (b_H)^{0.5}}$$

where

μ_m = mean coefficient of friction
X_r = load sharing factor
w_{Nr} = normal unit load
V_{r1} = rolling velocity of the pinion
V_{r2} = rolling velocity of the gear
B_M = thermal contact coefficient
b_H = semi-width of Hertzian contact band

Mean Coefficient of Friction, μ_m

The following equation gives a typical value of $0.06 < \mu_m < 0.18$ for the mean coefficient of friction for gears operating the the partial EHD regime ($\lambda < 1$). It may give values too low for boundary-lubricated gears where μ_m may be greater than 0.2, or too high for gears in the full-film regime ($\lambda > 2$), where μ_m may be less than 0.01.

$$\mu_m = 0.06 \left(\frac{50}{50 - S} \right)$$

where

$$\left(\frac{50}{50 - S} \right) \leq 3.0$$

S = average surface roughness, rms

$$S = \frac{\sigma_1 + \sigma_2}{2}$$

Thermal Contact Coefficient, B_M

The thermal contact coefficient is given by

$$B_M = (\lambda_M \rho_M C_M)^{0.5}$$

where

λ_M = heat conductivity
ρ_M = density
C_M = specific heat per unit mass
For typical gear steels, $B_M \approx 43$ Lbf/[in $s^{0.5}$ °F]

Table 1[5] - Welding Factor X_w

Material	X_w
Through-hardened steel	1.00
Phosphated steel	1.25
Copper-plated steel	1.50
Nitrided steel	1.50
Carburized steel	
Content of austenite < average	1.15
Content of austenite average	1.00
Content of austenite > average	0.85
Stainless steel	0.45

Semi-Width of Hertzian Contact Band, b_H

$$b_H = \left(\frac{8X_r w_{Nr} \rho_n}{\pi E_r} \right)^{0.5}$$

Bulk Temperature, T_b

The gear bulk temperature is the equilibrium bulk temperature of the gear teeth before they enter the meshing zone. In some cases, the bulk temperature may be significantly higher than the temperature of the oil supplied to the gear mesh. In a test with ultra high-speed gears [4], the pinion bulk temperature was 275°F (171°F hotter than the oil inlet temperature). For turbine gears at lower speeds, the bulk temperature rise of the gear teeth over the inlet oil temperature may range from 20F at 12,000 fpm pitch line velocity to 40°F at 16,000 fpm. At similar speeds, the bulk temperature rise of aircraft gears with less oil flow may range from 40°F to 60°F.

Scuffing Temperature, Ts

The scuffing temperature is the contact temperature at which scuffing is likely to occur with the chosen combination of lubricant and gear materials.

For mineral oils without anti-scuff additives or for mineral oils with low concentrations of anti-scuff additives, the scuffing temperature is independent of the operating conditions for a fairly wide range. For these oils, the scuffing temperature may be correlated with the composition of the oil. The viscosity grade is a convenient index of the composition and, thus, of the scuffing temperature.

For non-anti-scuff mineral oils, the mean scuffing temperature (50% chance of scuffing) is given by

Table 2[6]
Synthetic Lubricant Mean Scuffing Temperature, Ts

Lubricant	Mean Scuffing Temp. Ts (°F)
MIL-L-6081 (grade 1005)	264
MIL-L-7808	400
MIL-L-23699	425
DERD2487	440
DERD2497	465
DOD-L-85734	500
MOBIL SHC624	540
DEXRON II	550

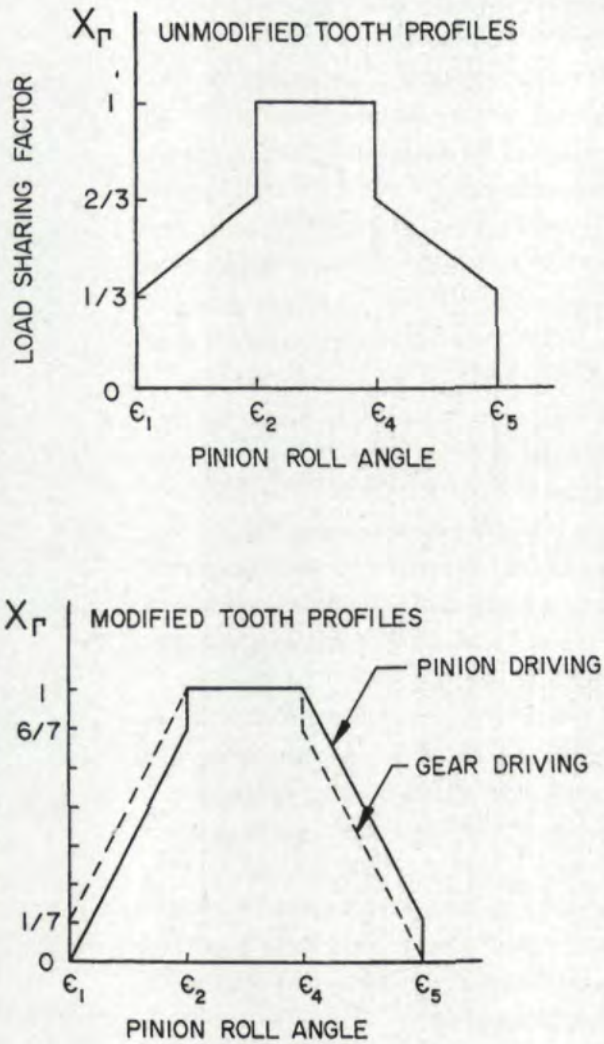

Fig. 4

$$Ts = 146 + 59\ln\nu_{40}\ ^\circ F$$

For mineral oils with low concentrations of anti-scuff additives, the mean scuffing temperature is given by

$$Ts = 245 + 59\ln\nu_{40}\ ^\circ F$$

where

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

The scuffing temperature determined from FZG test gears for mineral oils without anti-scuff additives or with low concentrations of anti-scuff additives may be extended to different gear steels, heat treatments, or surface treatments by introducing an empirical welding factor:

$$Ts = T_{b'test} + X_w T_{f'test}$$

where

X_w = welding factor (See Table 1.)
$T_{b'test}$ = bulk temperature of test gears
$T_{f'test}$ = maximum flash temperature of test gears.

For synthetic lubricants and carburized gears typical of the aerospace industry, the scuffing temperatures are shown in Table 2.

For mineral oils with high concentrations of anti-scuff additives, such as hypoid gear oils, research is still needed to determine whether the scuffing temperature is dependent on the materials and/or operating conditions. Special attention has to be paid to the correlation between test conditions and actual or design conditions.

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References:

1. DOWSON, D. "Elastohydrodynamics." Paper No. 10, Proc. Inst. Mech. Engrs. Vol. 182, PT3A, 1967, pp. 151-167.
2. BLOK, H. "Les Temperatures de Surface dans les Conditions de Graissage sons Pressions Extreme." Second World Petroleum Congress, Paris, June, 1937.
3. AGMA 2001-B88. "Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth." 1988.
4. AKAZAWA, M., TEJIMA, T. & NARITA, T. "Full Scale Test of High Speed, High Powered Gear Unit - Helical Gears of 25,000 PS at 200 m/s PLV." ASME Paper No. 80-C2/DET-4, 1980.
5. AGMA. op.cit.
6. DRAGO, R. J. "Comparative Load Capacity Evaluation of CBN-Finished Gears." AGMA Paper No. 88 FTM 8. Oct, 1988.

Surface Pitting Fatigue Life of Noninvolute Low-Contact-Ratio Gears

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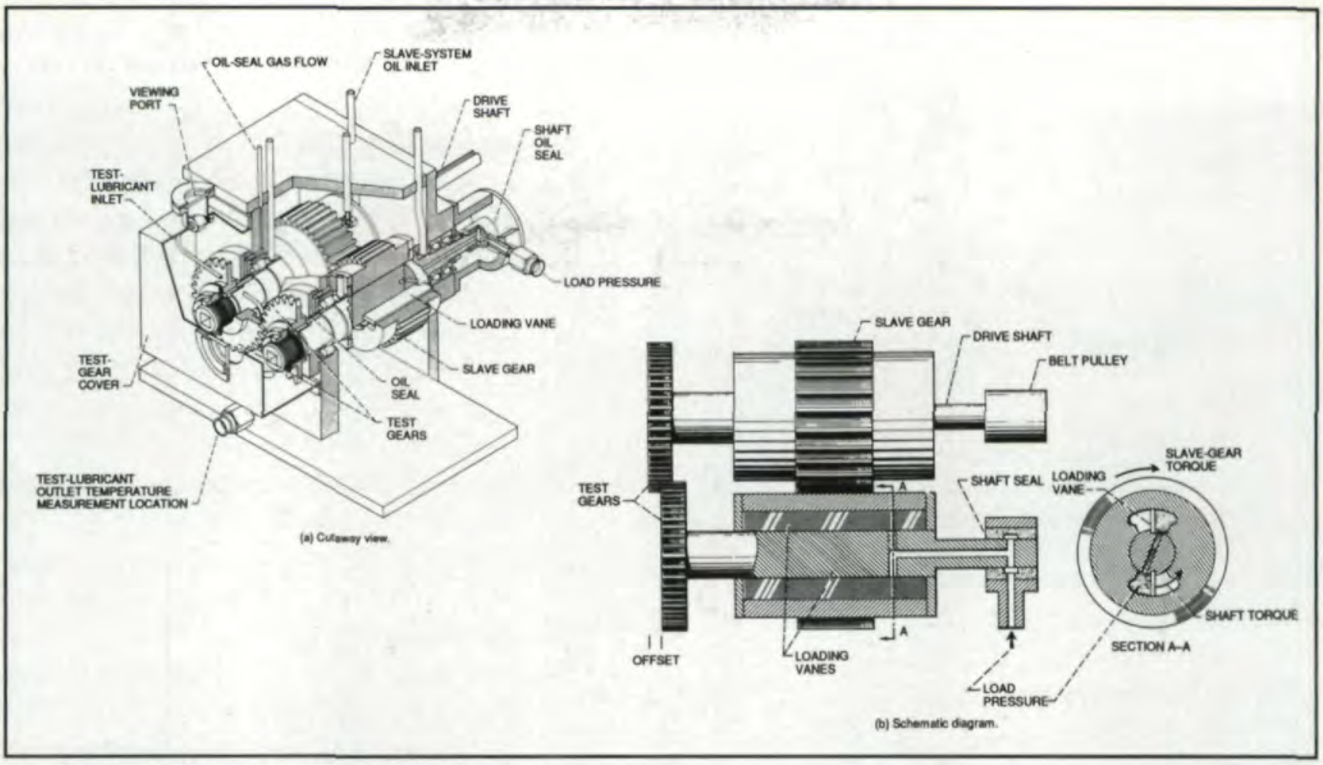
Abstract: Spur gear endurance tests were conducted to investigate the surface pitting fatigue life of noninvolute gears with low numbers of teeth and low contact ratios for use in advanced applications. The results were compared with those for a standard involute design with a low number of teeth. The gear pitch diameter was 8.89 cm (3.50 in.) with 12 teeth on both gear designs. Test conditions were an oil inlet temperature of 320 K (116° F), an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The following results were obtained: The noninvolute gear had a surface pitting fatigue life approximately 1.6 times that of the standard involute gear of a similar design. The surface pitting fatigue life of the 3.43-pitch AISI 8620 noninvolute gear was approximately equal to the surface pitting fatigue life of an 8-pitch, 28-tooth AISI 9310 gear at the same load, but at a considerably higher maximum Hertz stress.

Introduction

Many gears used in aircraft and other transmissions have size limitations based on the minimum number of teeth that can be cut on a pinion without undercutting the teeth.[1] If the number of teeth is made fewer than this minimum, a weaker tooth will be produced because of the undercutting. One method currently used to allow smaller minimum numbers of teeth on a pinion is to change the involute radius at both the dedendum and the addendum. Smaller numbers of teeth can be manufactured on a given pinion with a standard addendum by increasing the involute radius in the dedendum region, where it normally becomes very short, and decreasing the radius in the addendum region, where it normally increases rapidly. This can be seen on an involute chart as a positive modification in the dedendum region and a negative modification, similar to a large tip relief, in the addendum region. In addition to allowing smaller numbers of teeth without undercutting, this method, sometimes called new tooth form,[2] also reduces the maximum Hertz stress in the dedendum region, where the very short involute radius has been increased. This increased involute radius may also improve the gear tooth's surface fatigue life and possibly improve its scoring resistance. The new tooth form can be used for most spur or helical gears with either normal or high contact ratios to reduce the effect of undercutting on gears with fewer than the minimum number of teeth.

The objectives of the research reported herein were (1) to investigate the noninvolute modifications for use as a design method for gears with small numbers of teeth, (2) to determine the surface endurance characteristics of a spur gear with the new tooth form, and (3) to compare the results with those for a standard involute gear of similar design parameters. In order to accomplish these objectives, tests were conducted with one



(a) Cutaway view.

(b) Schematic diagram.

Fig. 1 NASA Lewis Research Center's gear fatigue test apparatus.

lot each of spur gears made from a single heat of AISI 8620 AMS 6274 material in the noninvolute design and in the standard involute design. The gear pitch diameter was 8.89 cm (3.50 in.). Test conditions included an oil inlet temperature of 320 K (116° F), which resulted in an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a shaft speed of 10,000 rpm.

The work reported herein was conducted as a joint effort of NASA Lewis Research Center, General Electric Co. Ordinance Systems Division, and ITW Spiroid Division.

Apparatus and Procedures

Gear Test Apparatus. The gear fatigue tests were performed in the NASA Lewis gear fatigue test apparatus (Fig. 1). This test rig uses the four-square principle (recirculating power) of applying the test gear load so that the input drive needs to overcome only the frictional losses in the system. A schematic of the test rig is shown in Fig. 1b. Oil pressure and leakage flow are supplied to the loading vanes through a shaft seal. As the oil pressure is increased on the loading vanes inside the slave gear, torque is applied to the shaft. This torque is transmitted through the test gears back to the slave gear, where an equal, but opposite torque is maintained by the oil pressure. This torque on the test gears, which depends on the hydraulic pressure applied to the loading vanes, loads the gear teeth to the desired stress level. The two identical test gears can be started under no load, and the load can be applied gradually without changing the running track on the gear teeth.

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

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

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

Test Gears. The test gears are shown in Fig. 2. Their dimensions are given in Table 1. All the gears had a nominal surface finish on the tooth face of 0.82 μm (32 μin.) rms. The baseline gears had a standard involute profile; the noninvolute gear had a profile that deviated from a standard

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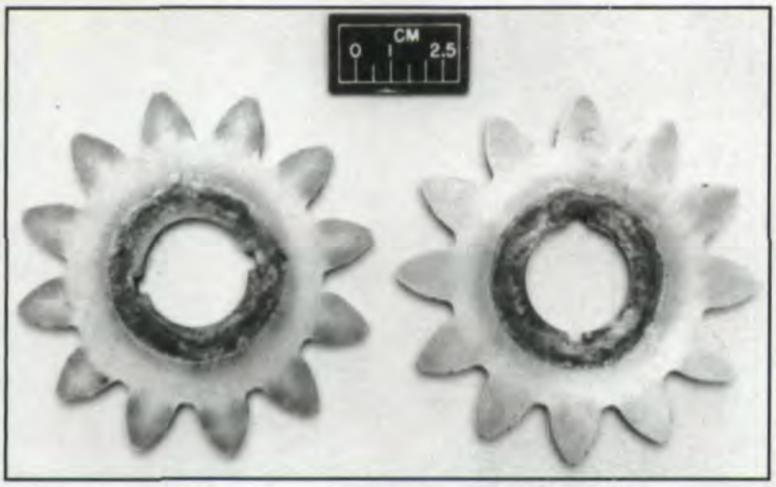


Fig. 2 Test gear configurations. (a) Standard involute. (b) Noninvolute.

Table 1 Description of Test Gears*

Pitch diameter, cm (in.)	8.89 (3.50)
Number of teeth	12
Module (diametral pitch)	7.4083 (3.4286)
Pressure angle (deg)	20°
Face width, cm (in.)	0.635 (0.250)
Outside diameter cm (in.)	10.2558 (4.0377)
Root diameter, cm (in.)	7.248 (2.854)
Tooth thickness (arc), cm (in)	1.3528 (0.5326)
Fillet radius, cm (in.)	0.198 (0.078)
Surface finish (min.), m (in.)	0.8 (32)

*Gears were identical except for the tooth form, which was involute for the standard gear and noninvolute for the other gear.

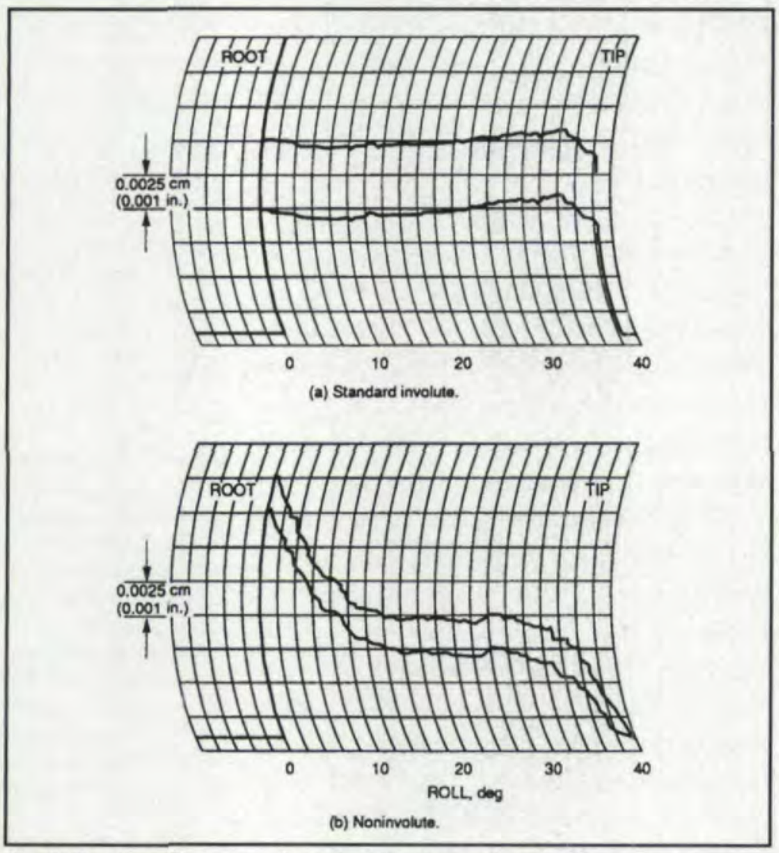


Fig. 3 Tooth profile traces of standard involute and noninvolute gears.

involute profile as shown in Fig. 3. The pressure angle for all the gears was 20° and the contact ratio was 1.15.

Test Materials. The test gears were manufactured from one lot of AISI E8620 AQHR AMS 6274 material. The chemical composition of the gear material is shown in Table 2. The heat treatment for the test gears is described in Table 3. The case hardness was R_c 60 with a case depth of 0.147 cm (0.058 in.); the core hardness was R_c 40. Photomicrographs of the case and core regions of the gear material are shown in Figs. 4a and b.

Lubricant. All the gears were lubricated with a single batch of synthetic paraffinic oil. The physical properties of this lubricant are summarized in Table 4. Five volume percent of an extreme-pressure additive, designated Lubrizol 5002 (partial chemical analysis given in Table 4), was added to the lubricant.

Test Procedure. After the test gears were cleaned to remove their protective coating, they were assembled on the test rig. The test gears ran in an offset condition with a 0.30 cm (0.120 in.) tooth-surface overlap to give a 0.28 cm (0.110 in.) load surface on the gear face after allowing for the edge radius on the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each set of gears. All tests were run-in at a load per unit length of 1230 N/cm (700 lb/in.) for one hour. The load was then increased to 5800 N/cm (3300 lb/in.), which resulted in a 1.49 GPa (216 ksi) pitch-line maximum Hertz stress. The tooth bending stress at the worst load point was calculated to be 0.10 GPa (15 ksi).

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

The pitch-line elastohydrodynamic (EHD) film

Table 2 Chemical Composition of AISI 8620 Gear Material

Element	Content, wt %
Carbon (core)	0.22
Manganese	.82
Phosphorus	.013
Sulfur	.01
Silicon	.27
Copper	.16
Chromium	.49
Molybdenum	.16
Nickel	.54
Iron	Balance

Table 3 Heat Treatment Procedure (Material, AISI 8620)

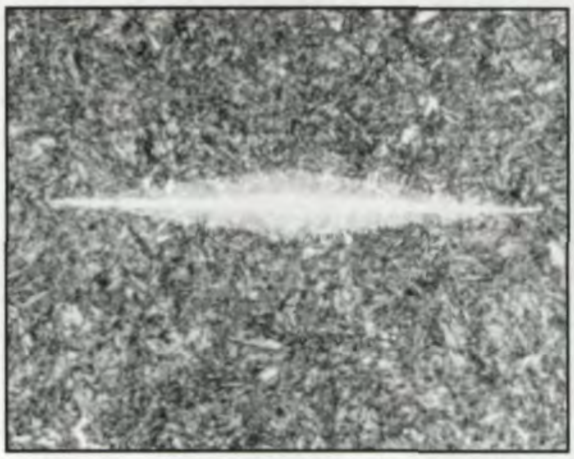
Step	Process
1	Carburize at 1200 K (1700° F) for 8 hrs.
2	Temper at 922 K (1200° F) for 1 hr.
3	Austenize or harden at 1118 K (1550° F) for 2.5 hrs.
4	Oil quench
5	Deep freeze at 190 K (-120° F) for 3.5 hrs.
6	Temper at 436 K (325° F) for 2 hrs.

thickness was calculated by the method of Dowson and Higginson.[3] It was assumed, for this calculation, that the gear temperature at the pitch line was equal to the oil outlet temperature and that the oil inlet temperature to the contact zone was equal to the gear temperature, even though the oil inlet temperature was considerably lower. It is possible that the gear surface temperature was even higher than the oil outlet temperature, especially at the end points of sliding contact. The EHD film thickness for these conditions was computed to be 0.94 μm (37 μin.), which gave an initial ratio of film thickness to composite surface roughness h/σ of 0.82 at the 1.49 GPa (216 ksi) pitch-line maximum Hertz stress.

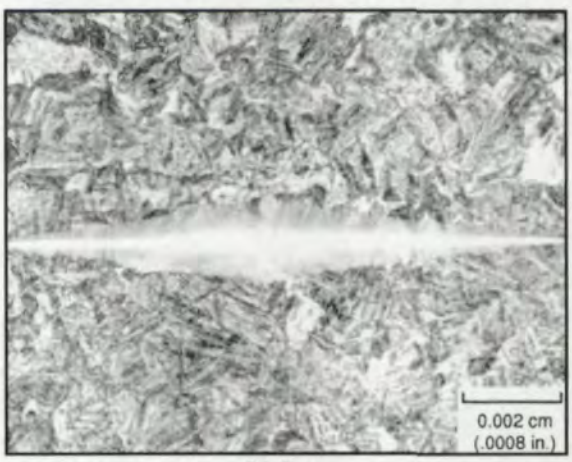
Each test conducted with a pair of gears was considered as a system and, hence, a single test. A maximum of four tests were conducted with each pair of gears. Test results were evaluated by using Weibull plots calculated by the method of Johnson.[4] (A Weibull plot is the number of stress cycles versus the statistical percentage of gear systems failed.)

Results and Discussion

One lot each of AISI 8620 standard involute gears and noninvolute gears with the modified involute profile was endurance tested. Test conditions included a tangential tooth load of 5800 N/cm (3300 lb/in.), which produced a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The gears failed by classical subsurface pitting fatigue. The surface pitting fatigue life results of these tests are shown in the Weibull plots of Fig. 5 and are summarized in Table 5. Surface pitting fatigue life results for the standard involute gears are shown in Fig. 5a. The 10% and



(a) Case



(b) Core.

Fig. 4 Photomicrographs of case and core of AISI 8620 test gear material.

Table 4 Lubricant Properties
(Lubricant, synthetic paraffinic oil plus additive.*)

Kinematic viscosity, cm^2/sec (cS) at:		
244 K (-20° F)	2500×10^{-2}	(2500)
311 K (100° F)	31.6×10^{-2}	(31.6)
372 K (210° F)	5.5×10^{-2}	(5.5)
477 K (400° F)	2.0×10^{-2}	(2.0)
Flash point, K (°F)	508	(455)
Fire point, K (°F)	533	(500)
Pour point, K (°F)	219	(-65)
Specific gravity	0.8285	
Vapor pressure at 311 K (100° F). torr	0.1	
Specific heat at 311 K (100° F), J/kg K (Btu/lb °F)	2190 (0.523)	

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

50% lives were 14.6×10^6 and 45.8×10^6 stress cycles (24.3 and 76.3 hr), respectively. The failure index (i.e., the number of fatigue failures out of the number of sets tested) was 20 out of 20. A typical fatigue spall that occurs near the pitch line on a standard involute gear is shown in Fig. 6a. This spall is similar to those observed in rolling-element fatigue tests. Pitch-line pitting is the result of a high subsurface shearing stress, which develops subsurface cracks. The subsurface cracks propagate into a crack network that results in a fatigue spall slightly below the pitch line, where the sliding condition is more severe.

Surface pitting fatigue life results for the noninvolute gear systems are shown in Fig. 5b. The 10% and 50% surface pitting fatigue lives were 23.2×10^6 and 62.5×10^6 stress cycles (38.1 and 104.2 hr), respectively. The failure index was 18 out of 18. Fig. 6b shows a typical fatigue spall for a noninvolute gear. The fatigue spalls and tooth wear were very similar for both types of gears. The 10% life of the noninvolute gear was approximately 1.6 times that of the standard involute gears. The confidence number was 77%, which indicates that the difference in surface fatigue life is statistically significant. (The confidence number indicates the percentage of time that the relative lives of the two types of gears will occur in the same order.)

The gear life data are summarized in Fig. 5c. The surface pitting fatigue test data show the noninvolute gear to be superior in surface pitting fatigue life to the standard involute gear for the gear sets tested. It is not clear why there was an improvement in surface pitting fatigue life for the noninvolute gear, since the fatigue failures occurred near the pitch line, where the load, Hertz stress, and involute radius are the same for both types of gears. Since the gears had a very low contact ratio on only 1.15 because of the low number of teeth, it is possible that the dynamic load for the noninvolute gear was less than that for the standard involute gear. Data from Lin et al.[5] indicate that certain types (or length) of profile modification give reduced dynamic loads. Since the noninvolute gear is a special form of profile modification, it may have a reduced dynamic load.

The 10% pitting fatigue life of the noninvolute

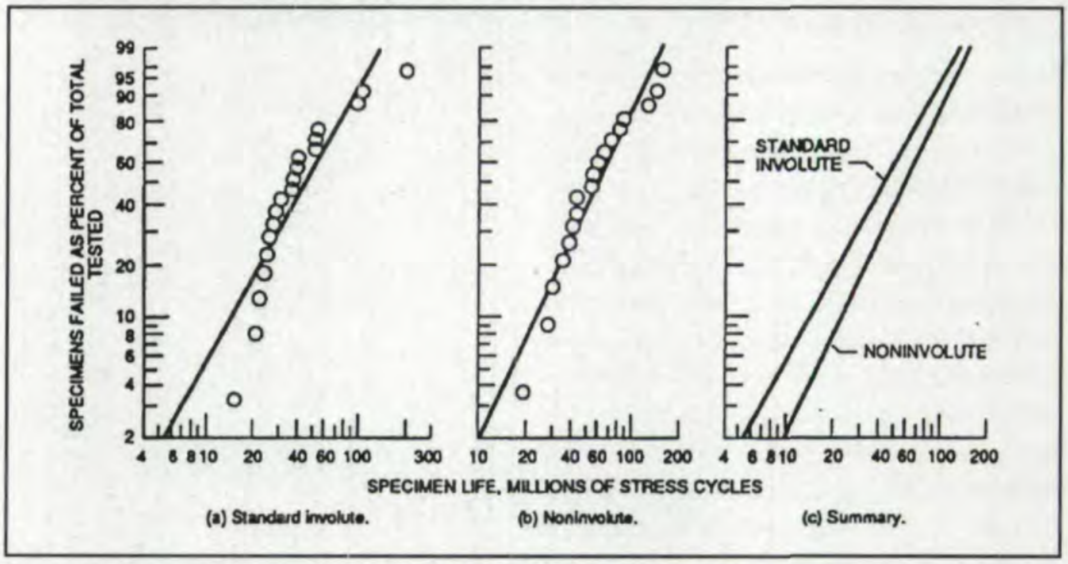


Fig. 5 Pitting fatigue lives of carburized and hardened AISI 8620 AMS 6274 standard involute and noninvolute gears. Speed, 10,000 rpm; lubricant, synthetic paraffinic oil with 5 vol % extreme-pressure additive; maximum Hertz stress, 1.49 GPa (216 ksi); temperature, 350 K (170° F).

Table 5 Results of Spur Gear Fatigue Life Tests

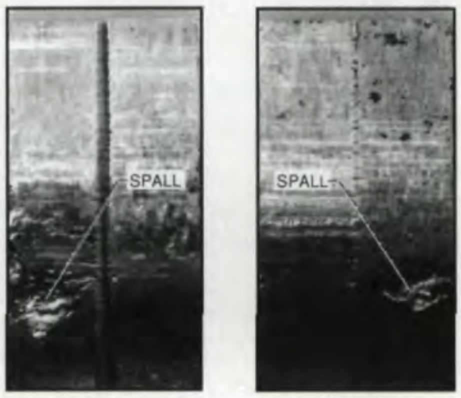
Pitch diameter, 8.2542 cm (3.2497 in.); maximum Hertz stress, 1.49 GPa (216 ksi); speed, 10,000 rpm; lubricant, synthetic paraffinic oil; gear temperature, 350 K (170 °F.)

Tooth form	10% life	50% life	Weibull slope	Failure index*	Confidence no. at 10% level#
	Gear system life, revelutions				
Involute	14.6×10^{6}	45.8×10^{6}	1.64	20 out of 20	–
Noninvolute	23.2	62.5	1.9	18 out of 18	77

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

Percentage of time that 10% life obtained with involute gears will have the same relation to 10% life obtained with noninvolute gears.

gear (23.3 million cycles) at the 1.49 GPa (216-ksi) maximum Hertz stress was approximately equivalent to that of a standard AISI 9310 8-pitch gear (19 million cycles) at the same load, but with a much higher maximum Hertz stress of 1.71 GPa (248 ksi).[6] The 8-pitch gears had a contact ratio of 1.638, in contrast to 1.15 for the 3.43-diametral-pitch gears used in these tests. Normally, the gear life is inversely proportional to the stress to the ninth power.[7] In these tests the low contact ratio may have resulted in higher dynamic loads on the teeth and therefore in a higher dynamic maximum Hertz stress and a reduced life. Results form the NASA gear dynamic analysis program show the 12-tooth gear to have a dynamic load factor of 1.5, in contrast to 1.04 for the 8-pitch, 28-tooth gear. In addition, the AISI 8620 material may have a lower fatigue life at the same stress than the AISI 9310 material.



(a) Standard gear. (b) Noninvolute gear.

Fig. 6 Typical fatigue spall.

Summary of Results

Spur gear endurance tests were conducted to investigate the effect of the noninvolute tooth form on the surface pitting fatigue life of gears with low numbers of teeth. The results were compared with those for a standard involute design with the same number of teeth. The gear pitch diameter was 8.89 cm (3.50 in.) with 12 teeth on both gear designs. Test conditions were on oil inlet temperature of 320 K (116° F), an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The following results were obtained:

1. The noninvolute gear had a surface pitting fatigue life approximately 1.6 times that of a standard involute gear of similar design.

2. The surface pitting fatigue life of the 3.43-pitch, 12-tooth AISI 8620 noninvolute gear was approximately equal to the surface pitting fatigue life of an 8-pitch, 28-tooth AISI 9310 gear at the same load, but at a considerably higher maximum Hertz stress.

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References:

1. DUDLEY, D.W. *Handbook of Practical Gear Design.* McGraw Hill, New York, 1984.

2. TOWNSEND, D.P., BABER, B.B., and NAGY, A. "Evaluation of High Contact Ratio Spur Gears With Profile Modification." NASA TP-1458. 1979.

3. DOWSON, D. & HIGGINSON, G.R. *Elastohydrodynamic Lubrication.* Pergamon Press, New York. 1966.

4. JOHNSON, L.G. *The Statistical Treatment of Fatigue Experiments.* Elsevier, New York. 1964.

5. LIN, H.H., TOWNSEND, D.P. & OSWALD, F.B. "Profile Modification to Minimize Spur Gear Dynamic Loading." NASA TM-89901, 1987.

6. TOWNSEND, D.P. "Surface Fatigue Life and Failure Characteristics of EX-53, CBS 1000M and AISI 9310 Gear Materials," NASA TP-2513, 1985.

7. TOWNSEND, D.P., COY, J.J., AND ZARETSKY, E.V. "Experimental and Analytical Load-Life Relation for AISI 9310 Steel Spur Gears." *Journal of Mechanical Design.* Vol 100. Jan. 1978, pp. 54-60.

Carbide Hobs

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Introduction

The following article is a collection of data intended to give the reader a general overview of information related to a relatively new subject within the gear cutting industry. Although carbide hobbing itself is not necessarily new, some of the methods and types of application are. While the subject content of this article may be quite broad, it should not be considered all-inclusive. The actual results obtained and the speeds, feeds, and tool life used in carbide hobbing applications can vary significantly.

History of Carbide Cutting Tools

The use of carbide has been accepted in the metal cutting tool industry for many years. Generally, when we talk about carbide cutting tools, we direct our attention to carbide inserts used mainly for machining centers and lathes. Carbide lends itself particularly to these types of tools for several reasons. One is that the insert itself has a relatively simple geometry (compared to a hob, for example). This geometry makes maintaining dimensional control of the tool somewhat easier during manufacturing process.

These types of tools take advantage of the benefits of carbide and at the same time tolerate some of its disadvantages. In most applications, the tool was used in high-speed, uninterrupted cuts that required the high heat qualities of the carbide material, yet was not hindered by the brittleness and lack of ability to withstand shock loads encountered during interrupted cuts. The stability of carbide at elevated temperatures also allowed insert manufacturers to take advantage of the properties of CVD titanium carbide and other CVD coatings to enhance the tool performance.

Early Carbide Hobbing

The benefits realized in the successful application of carbide in the insert industry made the ability to apply the same technology to the hobbing industry look quite attractive. Initially, the use of carbide here was somewhat limited to those applications with special needs that could not be met by the use of more conventional materials, such as high-speed steel.

Examples of these types of applications include hobbing gear materials, such as plastics, phenolics, or cast iron. These materials have a tendency to be very abrasive and can be very difficult to hob using high-speed steels. The high abrasion resistance of carbide in these cases offset the early manufacturing problems to be discussed in greater detail later.

Another example of the early use of carbide in the hobbing process is in hard finishing or skiving hobs. (See Fig. 1.) Here, the tool was used to finish hob gears after hardening. In many such cases, the part to be hobbed reached hardnesses in the range of 60Rc. High-speed steel simply would not hold up because of high part hardness.

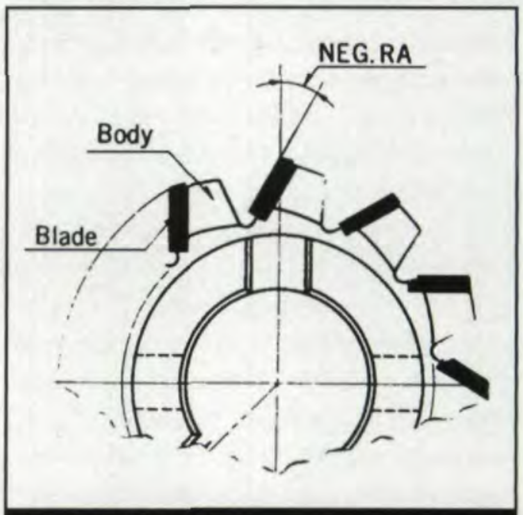

Fig. 1 - Skiving hob.

Hobbing Steel From Solid

More recently, a great deal of effort has been applied to using carbide in cases where normally high-speed steel hobs would be used. The main reason for this is the desire to take advantage of the high production rates that are possible with carbide. The gear hobbing industry has realized that in many cases the relatively high tool cost of a carbide hob can be more than offset by the reduction of machining costs. We are now using carbide in applications such as soft hobbing gears from the solid.

Transfering the technology learned from many successful years of carbide insert applications to the hobbing industry was very desirable. However, a number of problems needed to be overcome before carbide hobbing could be considered a viable alternative to high-speed steel hobbing.

One of the first obstacles was the inability of the older hobbing machines to provide the right conditions to take advantage of all that carbide hobs had to offer. Two key factors had to be addressed. The first was the rigidity of the machine being used. Because of its extreme hardness, carbide has an inherent tendency to chip, so every attempt had to be made to minimize looseness, vibration, and chatter. In many cases the older machines were not capable of withstanding the conditions created when attempting to use carbide hobs.

The second item that was critical to the success of carbide hobbing was the hob head speed capabilities. To take full advantage of carbide materials, hob speeds that had never before been realistically possible (because of the high-speed steel hobs being used) would have to now be made available. In many cases, hob speeds in the area of 900 sfm would have to be accommodated.

The hobbing machines were not the only early obstacles to applying carbide to the hobbing industry. One of the major problems that needed to be addressed was the ability of carbide to withstand the severe cutting conditions that are present during the hobbing process. The carbide would have to be tough enough to allow the cutting edges (individual hob teeth) to continuously enter and exit the cut during the generating process. The impact and shock resistance of the carbide would have to be increased if it were to be used successfully. Fortunately, over the last few years, a great deal of progress has been made with new grades of carbide, as well as grain size control, so that now numerous selections of grades for a given application are available.

Carbide Hob Manufacturing

Besides the machine and the material itself, the problem of manufacturing the carbide tool still remained. In most cases, the hob manufacturers were not tooled for or experienced in manufacturing tools made out of carbide. One attempt to overcome this was the "bladed" or "tipped" design. (See Fig. 2.) With this design, a steel hob was manufactured, the teeth were removed by gashing, and carbide blades were inserted and finished by grinding the teeth in the carbide from solid. This allowed the manufacturer to use many

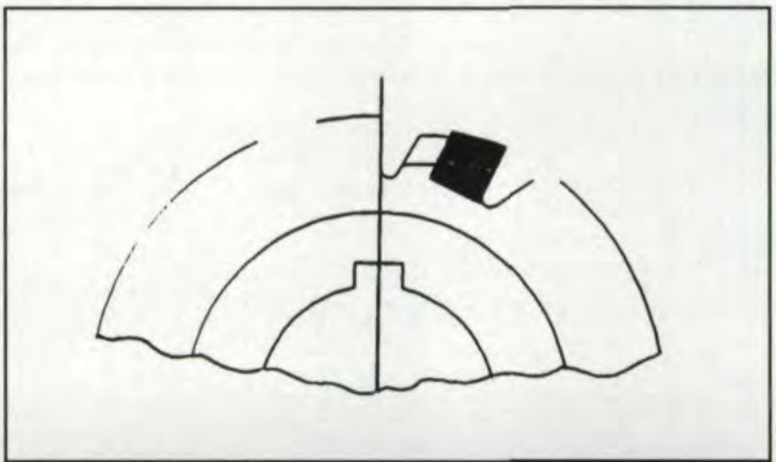

Fig. 2 - Carbide "tipped" hob.

of the same processes used on standard high-speed steel hobs.

This method, however, did present some problems. By using a composite design, the manufacturer had to develop an effective means of holding the carbide blade in the steel body. Early on the most widely used method was brazing. This method was successful, but did have drawbacks because of the heat required to braze. The high heat made it too difficult to accurately locate the carbide tip in its proper position. In some cases, the heat also caused the carbide to crack because of the difference in thermal expansion properties between the steel body and the carbide blade. The introduction of new adhesives in recent years allows the blade to be held effectively without the need for high heat. Positioning of the blades can now be done very accurately, and cracking has been virtually eliminated. Using this method does require a more accurate interface between the blade and body and, in most cases, requires ground surfaces to assure proper adherence.

To help minimize the amount of form grinding required to produce the teeth in the blade, one of two different approaches can be used. The first is to use preformed tips supplied with the teeth formed in each blade by the carbide manufacturer. The second is to use wire EDM to cut the teeth in each blade. (See Fig. 3.) The use of blades with teeth make accurate location of the blades in

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the body even more critical, so here, normally the adhesive bond technique is used.

The ability of carbide manufacturers to accurately produce preformed blanks has solved another carbide hob manufacturing problem. Through different methods, ranging from preform pressing to CNC machining before sintering, manufacturers are now able to deliver much more complex blanks than in the past. It is now possible to receive preformed hob blanks that have the bore, hubs, and gashes roughed into them. (See Fig. 4.) The hob manufacturer is then required to complete the grinding operations to the hole, hubs, sharpening, and tooth form to finish the hob.

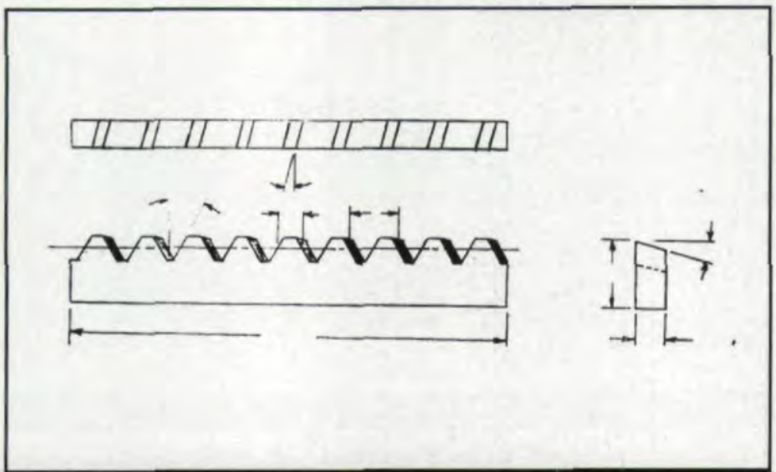

Fig. 3 - Pre-formed carbide tips.

Solid carbide blanks have an advantage over the composite design in that it is possible to enhance the performance of the tool with coatings without the threat of contaminating the furnace with the braze or adhesive bond material.

Design

Certain common practices should be followed to assure an acceptable design for carbide hobs. An example of one of these considerations is the avoidance of sharp corners. An internal sharp corner will result in a stress riser that could cause cracks to propagate. An external sharp corner creates a structurally less sound edge that could lead to premature failure due to chipping.

With high-speed steel hobs, it is possible to grind with an assortment of different types of grinding wheels (aluminum oxide, silicon carbide, Borazon,® etc.). When grinding with carbide, the choice is limited to diamond. Although work is currently being done to develop a dressable, vitrified diamond wheel, the majority of carbide form grinding is done with either plated diamond or resin bond diamond grinding wheels. The inability to readily form the diamond wheels has, in the past, limited the profile modifications that are permissible on the tooth flanks.

Maintenance

When discussing the proper maintenance of carbide hobs, a number of areas must be addressed. The first one - often overlooked - is the special care that must be taken when handling carbide hobs. Although precautions must be taken when handling any cutting tool, carbide is somewhat more brittle than high-speed steel and more susceptible to chipping and breakage. When dealing with carbide hobs, special cases for transporting and storage should be investigated.

Tooling is another area that should be examined. Some tooling that may be considered standard when using high-speed steel hobs, simply must be avoided when using carbide materials. One example is any kind of "press type" arbor. Obviously, given some of the material characteristics of carbide, any excessive force used with this type of arbor could lead to cracking sooner than with steel hobs.

The only maintenance with which the final user of a carbide hob must be concerned is the proper sharpening of the cutting face of the hob

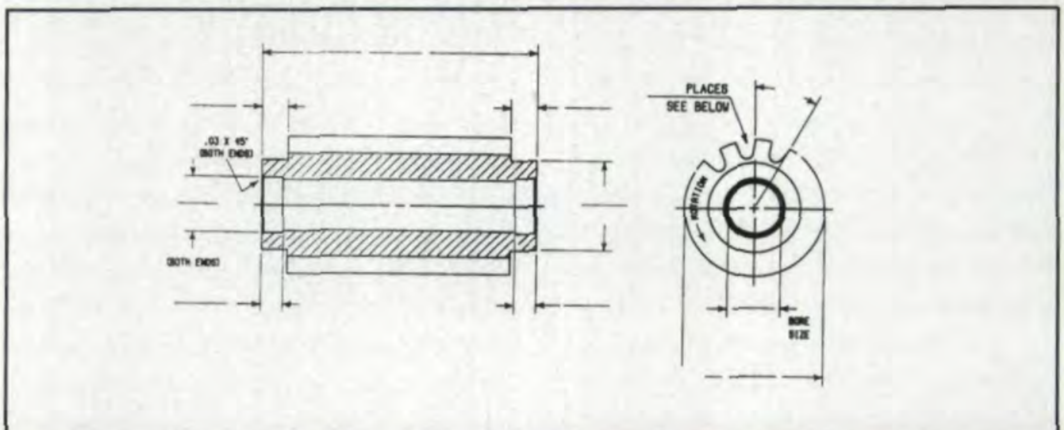

Fig. 4 - Solid carbide hob blank as manufactured by carbide vendor.

when the tool is worn. In practice, the same process is used to sharpen a carbide hob as a steel hob. The differences are the type of grinding wheel used, the feeds and speeds, and possibly the coolant used. Diamond grinding wheels should be used when sharpening carbide hobs. We have had success using both plated diamond and resin bonded diamond grinding wheels. The actual wheel specification can vary according to the machine being used, and the surface finish and amount of stock removal required. The wheels we have found successful are:

- Plated - Universal Super Abrasive (Elgin Diamond) 180 Grit
- Resin Bond - Universal Super Abrasive (Elgin Diamond) 180 Grit 100 concentration

Both wheels are capable of producing surface finishes of 16µ or better (resin as low as as 6-8mm) if applied correctly.

Grinding wheel speeds will vary according to the type of wheel being used, but, typically, to provide the proper cutting conditions, the speed will be in the range of 6500/6700 sfm. The table feed rate should stay in the area of four inches per minute. The stock removal rate is very critical when sharpening carbide hobs. The rate of removal is much less than with steel hobs. Normally .001 stock removal per pass of the grinding wheel should not be exceeded. Stock removal rates much higher than this can lead to heat generation, causing sharpening cracks. Note that using magnaflux to detect cracks in carbide tools (as might be done with steel hobs) is not recommended. A more readily accepted method for crack detection in carbide is a chemical means, such as Zyglo.

One final topic to cover in the proper maintenance of carbide hobs is the coolants being used, not only in the sharpening operation, but also in the hobbing application. Cutting oils should be evaluated to assure that no elements in the chemical make-up of the oil may be detrimental to the carbide material itself. Certain sulfur and chlorine additives can tend to leach the carbide binders, leaving an extremely brittle cutting edge that will lead to premature failure of the tool.

Carbide Properties

Of the many different types of carbide materials available today, the one based on tungsten carbide and cobalt has the widest industrial use. In the case of machining steel, the chips are much different from those produced in the early carbide hob applications discussed previously. The chip in this case can be relatively long and stringy. The properties of the newer grades of carbide for these applications have been tailored to meet the needs present here.

The strength and hardness of a cobalt-bonded tungsten carbide section depends primarily on the uniformity and the thickness achieved in the cobalt film surrounding the carbide particles. This is controllable by adjusting the proportion of cobalt to tungsten carbide and, to a lesser degree, by varying the particle size of the tungsten carbide. Smaller amounts of cobalt will result in the structure assuming properties more like tungsten carbide itself; namely, higher hardness and increasing tendency toward brittleness and lower strength. The effect of cobalt on hardness and strength is represented in Figs. 5 and 6.

Aside from its tremendous compressive strength and hardness, the best known characteristic of carbide

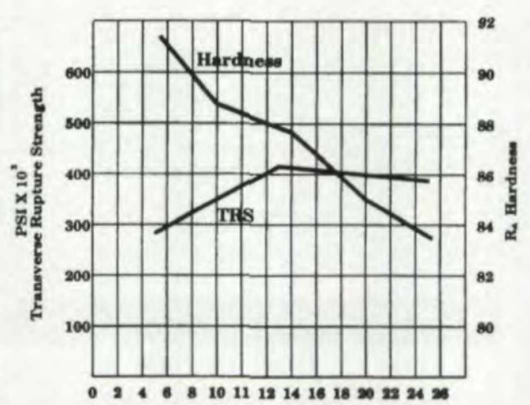

Fig. 5 - Hardness and strength vs. cobalt content.

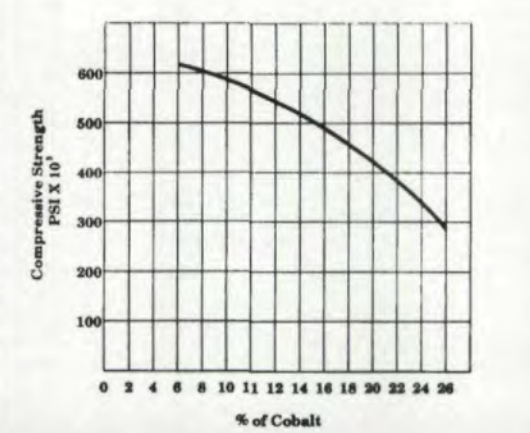

Fig. 6 - Effect of cobalt on compressive strength.

is its abrasion resistance. (See Fig. 7.) However, a direct relationship exists between these two characteristics and the transverse rupture strength. This is a measure of shock resistance. (See Fig. 8.)

Carbide Grades

Each carbide vendor has a wide selection of carbide grades available for a given application. The primary bases for selection of a specific grade of carbide are the material being cut and the type of cutting application being performed. A chart showing a number of different carbide vendors and their specific grade designations is shown in Fig. 9. In many cases, within a specific grade, there may be variations as far as grain structure and size to help tailor the performance to an application. The carbide vendor should be included in discussions regarding grade selection.

Application Results

As stated in the introduction, the actual results obtained in specific applications of carbide hobbing can vary greatly. The following examples are included for reference only, but can help give an indication as to the possible benefits obtainable with the proper application of carbide hobs.

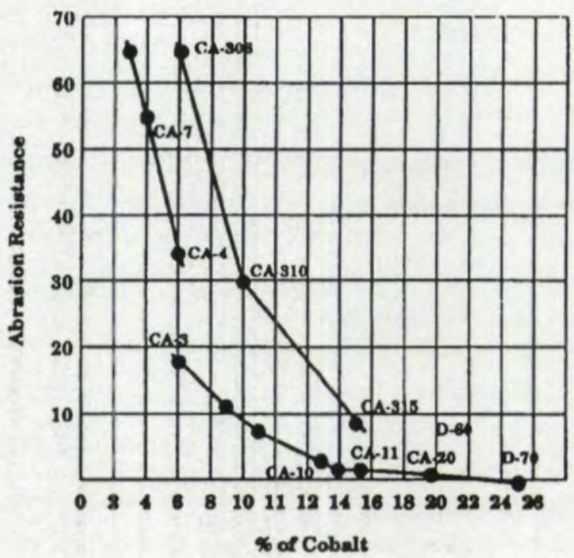

Fig. 7 - Relative resistance to abrasion. (Carmet grades.)

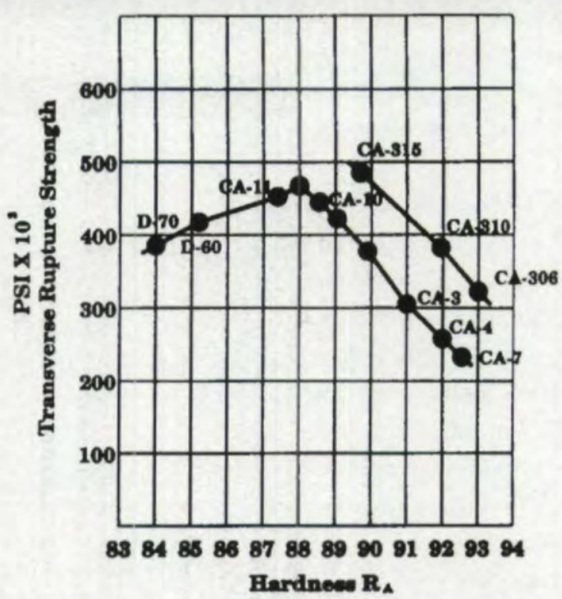

Fig. 8 - Transverse rupture strength vs. hardness. (Carmet grades.)

Application	Material	C Code	ISO Code	Working Conditions	ADAMAS	Carboloy	Firth Sterling	Kennametal	Newcomer	Sandvik	Valenite	VR-Wesson	Wendt Sonis
CHIP REMOVAL APPLICATIONS	CAST IRON, NON-FERROUS METALS AND NON-METALLIC MATERIALS	C-1	K30	Roughing Cuts	B	**523 44A 883	**HN +4 HN + H17 H8	**KC 250 K8735 K1	N10	H10F	**VN8 **VIN **V88 VC1 VC28	**650 **653 **660 2A68 RAMET 1 VR 54	**CQ22T **U222 **722 CQ2
		C-2	K20	General Purpose	#ADAMAX 6000 †ROXIDE †264 †241 A PWX AM	†570 (560) **523 875 883	**HN+ †CC46 H6	†KC910 (KC 950) **KC210 K8735 K6	**NT25 **NT2 N25 N22	**GC1025 **GC315 †GC015 †GC320 †GC310 (GC415) HIP H10F H13A H20 HM HML IRB	**VN8 **VIN **V88 †V01 †V05 VC2 VC28	**630 **633 **650 **660 (680) 2A5 VR54 VR 82	**CQ22T †918 **U222 **Q27 CQ2 CQ23
		C-3	K10	Light Finishing	#ADAMAX 3000 †ROXIDE †241 PWX	†570 (560) **523 895 883	**HN+ †CC46 H21 HA HTA	†KC910 (KC950) **KC210 K68	**NT 25 N30 N20	**GC1025 **GC315 †GC015 †GC320 †GC310 (GC415) HIP H10 HM IRB	**VN8 **V88 †V01 †V05 VC2	**630 **633 (680) (690) 2A7 VR52 VR82	**CQ23T **U222 **723 CQ23
		C-4	K05	Precision Boring and Finish Turning	†241 GU-2	†545 895	†CC46	†KC910 (KC950) **KC210 K11 K8	**NT25 N20	**GC1025 **GC315 †GC015 †GC310 †GC315 HIP	**V88 †V01 †V05 VC3	(680) (690) 2A7 VR52 *VR65 *VR97 *VR100	**CQ23T **037 CQ3
	STEELS AND ALLOY STEELS	C-5	P40	Rough and Interrupted Cutting at Slow Speeds	434	**515 **518 **538 **550 370 390	**TC+4 **HN+4 TO4	**KC850 KM K420 K21 K2S	**NT5 N55 N52	**GC135 **GC235 (GC435) S6 S30T S35 SM30 SMA	**VIN VC56	**643 **650 **653 †655 VR77 VR79 WS	**CY17T **717 CY17
		C-5	P30	Rough and Interrupted Cutting at Moderate Speeds	*T50 **764 **CNC †264 499 THERMILL #ADAMAX 6000	†570 (560) **550 **538 **536 **518 **515 375 370	**TC+4 **TC+ **HN+4 **HN+ NTA T12 T14	**KC850 (KC950) **KC810 K2885 K2884 K21 K420 K2S	**NN55 †NA02 N50	**GC1035 **GC135 GC225 **GC120 †GC015 (GC415) (GC435) S4 S30T S35 SM SM30 SMA	**VN8 **V88 **VIN †V01 †V05 VC56 VC35M VC5	**650 **653 (655) †680 **660 **663 VR75 VR77	**CY15T **U225 **715 **716 **717 **755 CY16
		C-6	P-20	General Purpose	*T60 **CNC **764 †ROXIDE †264 499 THERMILL	†570 (560) **550 **536 **515 375 370	**TC+ **HN+ †CC46 TXH T22	**KC850 (KC950) **KC810 K2885 K2884 K29 K4H	**NN55 **NN6 †NA02 N60	**GC1025 **GC120 **GC225 †GC015 (GC415) S2 S30T SM SM30 SMA	**VN8 **VN5 **V88 †V01 †V05 VC5 VC35M	**660 **663 (655) (680) VR 73	**CY15T †918 **U225 **U227 **714 **715 **716 **755 CY14 CY16
		C-7	P10	Finish Cuts	#ADAMAX 3000 **CNC †ROXIDE 548 *T80	†570 (560) †545 350	**TC+ **HN+ †CC46 T24 T25	†KC910 (KC950) K45 K5H	**NN55 **NN6 †NA02 N70	**GC1025 **GC120 **GC225 (GC415) †GC015 SIP SIOT	**VN8 **VN5 **V88 †V01 †V05 VC165 *VC67 VC7	**660 **663 (680) (690) *VR65 VR73	**CY15T **U227 **714 **715 CY14 CY31
		C-8	P05	Precision Turning and Boring	*T80 **CNC MIRROCUT MIRROMILL	†570 †545	†CC46 *SD3	†KC910 (KC950) K7H	N95 N80	†GC015 (GC415) SIP SIOT	**V88 *VC67 *VC83 †V01 †V05 VC7	(680) (690) *VR65 *VR97 *VR100	**CY31T **U227 **714 **731 *T18
WEAR APPLICA-TIONS		C-9	–	Wear Surface No Shock	PWX A	895 883	HA	K68 K96 K6	N20	CS05 CS10 CS20	VC9	2A5 2A68	CQ12
		C-10	–	Wear Surface Light Shock	B	44A 779	H	K95	N10	CS20 CG35	VC10	2A68 2A6	CQ12
		C-11	–	Wear Surface Heavy Shock	BB	55A 55B	HC	K94 K92 3109	–	CG35 CG40 CG60	VC11	2A3 2A1	CQ14
IMPACT APPLICA-TIONS		C-12	–	Impact-Light	BB	779 115 55A	DC1 DC2	K94 3109	–	CT45 CT50	VC12	2A3 VR13	U50 CQ14
		C-13	–	Impact-Medium	HD15	55A 55B	DCX	K92	–	CT60	VC13	VR14 VR15	CQ13

NOTES: *Grades containing more than 50% titanium carbide.
**Coated Grades
#Cermet Grades
†Al_2O_3 Coatings.
Multiple Coated Grades including Al_2O_3( )

The above chart is not a grade comparsion chart. The information listed herein is approximate, and instructions regarding the specific use and application of any competitive grade should be obtained direct from the manufacturer.

Fig. 9 - Grade selections.

Steel Hob vs. Carbide Hob Comparison

Example 1

Part Data:

Number of teeth	11	Helix angle	0
Pitch	30	Face width	0.53
Outside diameter	0.46	Hardness Rc	35

Hob Data:

# threads	1	Outside diameter	1.25
Material Steel (M3+TiN) vs Carbide			

Machining Data:

	Steel	Carbide
Speed RPM	500	2500
Speed SFM	163	820
Feed rate	.030	.035
Shift Amount	.009	.009
# pieces per shift	1	2

Total Cost Per Part Analysis

	Steel	Carbide
I. Tool cost per part		
Cost of tool	$300	$1300
No. of pieces per sharpening	140	280
Amt. of stock removed per sharpening	.010	.005
No. of sharpenings per hob	9	18
Total pieces per tool	1260	5040
Total cost per piece	$0.23	$0.26
II. Hobbing cost per part		
Feed	.030	.035
Hob travel - inches	.530	.530
No. of teeth in part	11	11
No. of threads in hob	1	1
Hob speed - rpm	500	2500
Helix angle - degrees	0	0
Hobbing time per piece (min.)	.390	.067
Shop labor rate per hour	$50	$50
Hobbing cost per piece	$.325	$.056
III. Total cost per part		
Tool cost + Hobbing cost	$.555	$.316

Steel Hob Vs. Carbide Hob Comparison

Example 2

Part Data:

Number of teeth	42	Helix angle	21
Pitch	17.8	Face width	1.00
Outside diameter	2.585	Hardness Bhn	180

Hob Data:

\# threads Steel-4 Carbide-1 Outside diameter 2.75

Material Steel (M3+TiN) vs. Carbide

Machining data:

	Steel	Carbide
Speed RPM	285	1000
Speed SFM	205	720
Feed rate	.060	.200
Shift amount	.0036	.0009
# pieces per shift	1	1

TOTAL COST PER ANALYSIS

		Steel	Carbide
I. Tool cost per part			
	Cost of tool	$1190	$3680
	No. of pieces per sharpening	1500	6000
	Amt. of stock removal per sharpening	.015	.008
	No. of sharpenings per hob	16	30
	Total pieces per tool	24000	180000
	Tool cost per piece	$0.05	$0.02
II.	Hobbing cost per part		
	Feed	.060	.200
	Hob travel - inches	1.00	1.00
	Number of teeth in part	42	42
	Number of threads in hob	4	1
	Hob speed - RPM	285	1000
	Helix angle - degrees	21	21
	Hobbing time per piece (min.)	.657	.225
	Shop labor rate per hour	$50	$50
	Hobbing cost per piece	$.548	$.188
III. Total cost per part			
	Tool cost + hobbing cost	$.598	$.208

Acknowledgement: *This article was presented at the Society of Manufacturing Engineers Gear Clinic in Nashville, TN, Oct., 1990. Reprinted with permission.*

When a Customer Files Bankruptcy: Write-Off is Not Your Only Remedy

Gary D. Boyn

MANAGEMENT MATTERS

Bankruptcy filings have not noticeably declined despite the economic recovery of the Reagan years. Businesses continue to receive notices that their customers have filed bankruptcy. Many of them are writing off significant losses each year as a result. Despite the frequent use of bankruptcy by debtors, many business owners and managers have little or no idea of the post-bankruptcy remedies available to them.

Once bankruptcy is filed, you are subject to the automatic stay. You no longer have a right to prosecute any lawsuits anticipated or pending against the debtor. You no longer have the right to repossess collateral in the possession of the debtor without prior court order. You no longer have the right to offset accounts, without prior court order. All actions against the debtor must stop.

The remedies available to you differ in a Chapter 7 liquidation from those in Chapter 11 reorganization. A Chapter 7 contemplates turnover of all the debtor's nonexempt assets to an independent trustee to liquidate. The only recovery the creditors will receive will be derived from the value of the assets administered by the trustee. The debtor will be forever discharged from any obligation to repay his debts incurred prior to bankruptcy. Under Chapter 11, the debtor is proposing to continue to operate his business under the protective arm of the court. He will usually act as his own trustee as debtor-in-possession. He will hold and control his assets, continue to do business, and ultimately file a plan of reorganization. In most cases, he will propose to pay only part of his debts and will be discharged from the balance.

File a Claim

If a Chapter 7 is filed, creditors should file proofs of claim with the bankruptcy clerk. These are simple, one-page forms that can be obtained from your attorney or the clerk. In most cases, you can fill out the forms yourself. Be sure to attach any written evidence of debt you have, such as invoices and promissory notes. If you have any knowledge of fraudulent conveyances by the debtor prior to bankruptcy, payments of large sums to family members or preferred creditors, unusual gifts or transfers of property prior to bankruptcy, or other misdealings by the debtor, notify the appointed trustee. The trustee has only the information contained in the debtor's schedules, and without creditors' assistance, he will rarely discover hidden assets.

Object to Discharge

Creditors should also determine whether they were induced to extend credit to the debtor by any fraudulent or false representation. You have the right to object to discharge of the debtor if you can prove he is hiding assets from the estate, has been guilty of fraudulent conveyances within one year before filing, has committed fraud in obtaining money or property, or is failing to comply with orders of the court. If you think you have grounds for such objection, discuss them with your attorney, as you must act before the discharge is granted.

Reclaim Goods

If you have shipped prod

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Gary D. Boyn
*is a partner in the Elkhart, IN, law firm of Warrick, Weaver & Boyn. He advises businesses regarding corporate and commercial matters, including all aspects of bankruptcy law. If you have questions for Mr. Boyn, circle **Reader Service No. 37.***

uct to the debtor just prior to bankruptcy, you have the right to reclaim goods within 10 days after their receipt. This right applies when the goods were sold on credit while the debtor was insolvent. If you find this is the case, immediately inform your attorney, who will file an action in the bankruptcy court to obtain court approval of your retaking the property.

Obtain Relief

If you have a lien on the debtor's property, and you believe the value of the property is less than the amount you are owed, you may file an application for relief from the automatic stay and abandonment. Relief from stay will allow you to take action to recover the property. Abandonment is nothing more than a disclaimer of interest by the trustee. The trustee must conclude that there is no value to be recovered for unsecured creditors. If the debtor's schedules indicate there is no value in excess of your lien, the trustee will usually consent to abandonment. If the schedules indicate that there is equity, you should have the property appraised to prove the debtor was incorrect. Once the property is abandoned, you can exercise any remedy you would normally have under the law just as if no bankruptcy were pending.

Chapter 11

If a Chapter 11 is filed, you should consider some additional actions. You should always file a proof of claim. If you do not, the debt as listed by the debtor in his schedules will control. If he substantially devalued your claim in the schedules, his figure will govern if you have not filed a claim. In addition, if the case is ultimately converted to Chapter 7 liquidation, you must have a claim on file in order to share in the final settlement.

Joint Creditors' Committee

If you are an unsecured creditor and have a large claim against the debtor, you should consider joining the creditors' committee. The members of the creditors' committee act on behalf of all unsecured creditors and review the financial records of the debtor, negotiate with the debtor as to the contents of the proposed plan of arrangement, and recommend to creditors whether to accept or reject the plan. The debtor will attempt to negotiate as cheap a deal with creditors as he can. If you want your money, you should be prepared to fight for it.

Attend First Meeting

The unsecured creditors should have someone in attendance at the first meeting of creditors to interrogate the debtor. There is a great deal of information to be gathered at that meeting. It is normal for debtors to overvalue their property in the bankruptcy schedules in order to demonstrate that they have substantial equity and a reasonable likelihood of surviving if they can get their debts extended. Careful cross-examination as to the source and validity of the figures used in the schedules can help later in disputes over the value of secured claims and the ability of the debtor to finance a plan. It is also important to determine what caused the debtor to become bankrupt, and to determine what efforts the debtor is making to remedy those causes. It is important to determine the present financial structure of the company and the debtor's break-even point for operations. If the debtor is continuing to operate at less than break-even level during the Chapter 11, people supplying credit to him during the Chapter 11 are at risk, and the debtor is not generating cash with which to fund a plan. The court will not be inclined to allow a debtor to operate indefinitely under such conditions. The creditors have no way to monitor the debtor's performance unless they begin with some understanding of his financial position.

Requesting a Review

If the creditors believe that the debtor has committed fraudulent acts, is untrustworthy, or is dissipating the assets of the estate, the creditors have the right to seek appointment of an examiner to review the acts and financial condition of the debtor, and to seek appointment of an independent trustee to operate the debtor's business during Chapter 11.

Seek Adequate Protection

In Chapter 11, the debtor is entitled to use property that has been pledged to secured parties unless the secured parties object. If they object, they are entitled to adequate protection of their collateral. If that does not occur, the secured parties have the right to seek relief from stay and abandonment of their property from the estate. If you are a secured creditor, and your property is in the hands of the debtor, you should contact your attorney to negotiate an agreement that you will have a continuing lien on the same assets and any proceeds, products, and replacements thereof. If you do not do so, and the debtor sells your collateral and reinvests the money in the business, your lien will disappear. Adequate protection can also mean providing a form of insurance or surety to protect the value of the collateral or depositing money instead of physical assets as collateral.

The debtor has no right under the Bankruptcy code to use "cash collateral" without prior approval of the bankruptcy court. Cash collateral is cash, negotiable instruments, documents of title, securities, deposit accounts, or other cash equivalents. The debtor can use it only if each entity with an interest in that cash collateral consents to such use, or the court has held a hearing and thereafter permits such use. If your security is cash collateral, you should contact the debtor immediately upon the filing of bankruptcy and advise him that you do not consent to such use unless he grants you adequate protection. Any agreement reached should be reduced to writing and approved by the bankruptcy court. Although the Bankruptcy Code specifically prohibits the use of cash collateral, most debtors will dip into money on hand to finance operations. Secured creditors must move quickly to minimize their risks.

Obtain Information

Creditors can have a substantial voice in the operation of the Chapter 11 if they take an active role. The creditors' committee can demand to be informed of every material decision made by the debtor-in-possession that affects its operations. It can demand more frequent and detailed financial accountings and meetings with the officers of the debtor to review the progress of the company. If the creditors are concerned about mismanagement, they can ask the court to limit the debtor's ability to operate his business, limit the amount of executive compensation to be paid, limit capital expenditures, or change the debtor's management.

Review Disclosure Statement

Many financial disclosure statements actually provide little financial information. If the debtor does not give sufficient financial information prior to filing a plan, creditors may object. In most instances, if your objections are reasonable, the court will order the debtor to amend his disclosure.

File Creditors' Plan

The debtor has the exclusive right to file the plan of arrangement within the first four months of bankruptcy. Once the exclusive period has run, whether or not the debtor has filed his plan, creditors have the right to develop and propose their own plan of arrangement. The creditors' plan may simply propose to liquidate all of the assets and pay the claims as allowed by the court. In some cases, that plan may be more reasonable than the debtor's plan to retain his assets and stay in business. Creditors should consider filing their own plan.

Vote No

Even if the creditors do not file their own plan, they need not accept the debtor's. If the creditors are convinced that the plan offers too little, they should consider contacting other creditors and soliciting their rejections of the proposed plan. The court has the right to approve the plan if the court determines that the creditors are receiving as much under the plan as they would receive if the debtor were liquidate under Chapter 7. In most cases, the court does not force acceptance of the plan, however, and the creditors' rejection of the debtor's plan will force him to offer a better deal.

When your account debtors file bankruptcy, you have the right and power to do more than simply write off your debt and take a bad debt deduction on your tax return. You have a number of options that may reduce your losses. Discuss these options with your attorney. Ignoring the case is not the answer.

The material in this article is provided for general information and should not be relied upon as legal advice generally or in any particular situation. Readers should consult their own legal counsel for advice as to how the subject matter of this article may affect them.

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American Gear Industry Faces Major Challenges

Rick Norment
Executive Director, AGMA

GUEST EDITORIAL

Five years of effort byAGMA came to fruition in January with the publishing of a report from the Department of Commerce. This "National Security Assessment of the U.S. Gear Industry" indicates that if serious measures are not taken, the gear industry's future is in jeopardy. It also sets the tone for confronting major challenges now looming large in our industry.

The report states: "The continued viability of the domestic gear industry is critical to U.S. national security and economic competitiveness," and that the outlook for the 1990s is dim. Today the gear industry is not capable of meeting potential gear production demands in a national emergency. The situation is most acute in the aerospace and marine gear sectors.

The Report

Analyzing profitability, R&D, capital investment, and surge and mobilization capacity of the gear industry, the report portrays a significant decline during the last decade. It also points to major infrastructural problems that can only be addressed by broad reforms. Gear industry facts cited include:

•Employment has shrunk since 1980. By 1987, the total number of employees had fallen by 37%, with production workers down nearly 40%.

•Pre-tax profitability fell each year from 1984 to 1988 and was very unevenly distributed, reflecting the diversified nature of the industry.

•The largest U.S. gear producers (500+ employees) were disproportionately damaged by the industry's decline in the 1980s. They will continue to be hardest hit in the 1990s due to increasing pressure from importers.

•Sales declined during the 1980s as most gear end-markets experienced their worst contraction since World War II.

•Alternative technologies have further eroded market opportunities for the U.S. gear industry.

•The outlook for the 1990s is for continued decline in end-market industries (farm, construction, and oil field equipment). This translates into decline in the gear industry as well.

Commercial Sector Implications

The DOC report establishes a clear link between the defense and commercial sectors of the gear industry. Production of gears for defense purposes cannot survive without a viable commercial industry. Besides being vital to the performance and construction of nearly all weapon systems, gears are also basic components for transportation and industrial machinery. Many AGMA companies make industrial gears that they consider "non-defense." On closer scrutiny, these companies may discover that the gears they manufacture actually go to OEMs that perform work for government contracts. This commercial base is critical to maintain manufacturing technology, a skilled labor pool, and capacity that can be converted to defense use if necessary.

Insufficient gear industry capacity in any sector means the industry would be unable to meet potential gear production demands in a national security emergency. As it stands now, the defense-intensive aerospace and marine gear sectors are already unable to reach emergency surge and mobilization production targets.

DOC officials predict that over a third of our industry's companies as they exist today will disappear within the next ten years. Severe international competition threatens our industry leaders. Citing the underlying causes as erosion of our markets, older equipment, and under-investment in capital and research, the report states that international competition has been a major factor in the decline of the industry. Gear imports during 1988 were $435 million. In 1990 they were over $500 million, an increase of more than 19%.

Government Assistance?

As an initial step toward meeting this crisis, the report recommends direct government assistance in rebuilding the gear industrial base. The proposals represent a significant change in attitude and are an important first step.

The DOC report recommends increased availability of low-interest loans and use of the Department of Defense's (DOD) Industrial Modernization Incentive Program to assist in plant modernization. The Bureau of Export Administration (BXA) proposes that Census, Trade Administration, and AGMA officials meet to begin to solve current deficiencies in trade statistics to improve government monitoring of trade problems. Additionally, BXA will as

GUEST EDITORIAL

sist by spearheading a meeting with Technology Administration officials to create shared flexible manufacturing centers and R&D consortiums, allowing collaboration between gear producers and/or their suppliers. More assistance in the development of technology and modernization of manufacturing processes from the National Institute of Standards and Technology is also urged. The report further recommends expanding the scope of its INFAC program to include all gear sectors and urges broader DOD participation.

The report suggests that DOD and DOC begin monitoring the troubled firms in the gear industry. If one or more should fail, the report suggests that capabilities of other U.S. firms should be developed to meet defense-critical needs. Finally, DOC encourages our industry to consolidate into larger, more technologically efficient firms to obtain better access to capital for investment in the latest technologies. This all sounds good, but what's the catch? These are just "recommendations," not "directives" that would guarantee program funding. AGMA and you must ensure that proposed solutions are implemented.

Moving Ahead!

How do we move ahead? As a first step, AGMA's Board of Directors has decided to file a 232 Petition. A 232 action investigates the effect of imports on national security. Under the provisions of the Omnibus Trade and Competitiveness Act of 1988, the President must determine which actions are necessary to adjust imports that threaten to impair national security. These could include restrictions on federal procurement practices (e.g, "Buy America" provisions); funding for manpower training programs; assistance in plant modernization; federal funding of industry research; and the collection of better trade statistics. The President could also restrict imports by establishing quotas, imposing tariffs, or negotiating a voluntary restraint agreement.

Our international trading partners fully understand the significance of a 232 Petition. In the midst of the Iraq War, a delegation of European gear manufacturers traveled to Washington in February to discuss trade issues with the AGMA Board of Directors. In their words, they were trying to "... avert a trade war." European representatives are threatening retaliation even before the 232 filing has occurred!

AGMA and its members fully recognize that successful international trade competition requires a level playing field. Currently, unreciprocal trade relationships and disparate manufacturing environments stifle U.S. gear manufacturers' ability to thrive, rebuild, or modernize. The 232 may provide some "breathing room" to achieve this goal. If we cannot make headway voluntarily, then AGMA may have no recourse but to push for sanctions against unfair trading practitioners. A key avenue of redress is the 232 Petition.

This report goes a long way toward focusing public attention on the gear industry. However, for the government to act decisively, the active support of the entire gear industry will be required. Equally important is gaining the support of our political leaders at both the state and federal levels. To marshall our forces, AGMA will distribute a survey, "Do You Know Your Elected Officials?" By effectively using the personal ties AGMA members have with these elected officials, AGMA will act! We need AGMA member CEOs to lead this effort! With your help, the American gear industry will once again resume its strong position in both the domestic and global markets.

For a copy of the Report National Security Assessment of the U.S. Gear Industry, contact the Office of Industrial Resource Administration, United States Department of Commerce, 202/ 377-3984.

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CALENDAR

AGMA Programs/Meetings/Seminars. For more information on AGMA programs, contact AGMA headquarters, (703) 684-0211.

MAY 21-22

Technical Education Seminar. Tapered Roller Bearings. Indianapolis, IN.

JUNE 5-6

Small Business Manufacturer's Committee. The Breakers, West Palm Beach, FL.

JUNE 6-9

AGMA 75th Anniversary Annual Meeting. The Breakers, West Palm Beach, FL. Speakers include Secretary of Labor, Lynn Martin, and Harley-Davidson Board Chairman, Vaughn Beals. Dr. Michael Bradley will again address the group on the economy.

JUNE 19

Technical Education Seminar Worm Gear Design. Cincinnati, OH.

Following are Society of Manufacturing Engineers technical conferences and events. For more information, contact Mike Traicoff, SME, (313)271-1500. Fax: (313) 271-2861.

JUNE 5-6

Fundamentals of Gear Design and Manufacture. Embassy Suites Hotels, Livonia, MI.

JUNE 11-13

SME Superabrasives '91 Conference & Exposition. O'Hare Exposition Center/Hyatt Regency O'Hare, Rosemont (Chicago) IL.

MAY 15-17

University of Wisconsin at Milwaukee. Micro Computer Applications in Gear Design and Analysis. Call Richard G. Albers, UWM, (800) 222-4643, (800) 222-4643 (in Wisconsin) or fax (414) 227-3125.

CALL FOR PAPERS. ASME 6th International Power Transmission and Gearing Conference. Interested authors should submit 150-250 word abstracts to Allen G. Strandford, Jr., Conference Chairman, Dresser Rand, P.O. Box 560, Olean, NY, 14760. Phone : (716)375-3285 Fax: (716) 375-3715.