GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

MAY/JUNE 1994

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Getting With The Program

Getting and keeping a work force capable of meeting the demands of the 21st century is one of the key challenges most U.S. manufacturers face today. That's not even news anymore. I — and others — have been talking about it in editorials and speeches for ten years now. It's also not news that the job is a tough one and that industry-wide response often has not been particularly effective.

But this is not a call to despair. It has taken time, but now some exciting responses are in the works. At the AGMA Annual Meeting this year, one particularly creative approach was outlined.

An organization called the National Tech Prep Network is addressing the challenge of creating a work force for the next century. The Tech Prep idea is a simple one: Employers and schools must work together to change the way we educate most of our students today.

The goal of Tech Prep is nothing less than changing the way Americans think and act about training people for jobs. We've always known that manufacturing jobs were not "dead ends" or just for "losers." Tech Prep wants to prove that to the people who will fill those jobs in the coming years, the students of today, and to those responsible for getting them started in the right direction, their teachers, parents and potential employers.

It's difficult to describe a typical "Tech Prep" program, because one of the beauties of the system is that each group that buys into the idea develops it in ways that meet its own particular needs.

For example, not too far from our editorial offices, over in northwestern Illinois, a group including local high schools, the regional community college and six area companies (including a major gear tool manufacturer) have established a Tech Prep Youth Apprenticeship Program. The program is a combination of rigorous academic study, on-the-job training, mentoring and counseling designed to provide students with "real world" experience and the necessary background to fill jobs in the high tech workplace.

It's important to note that the "school" part of this program is not a "remedial" one. Students get first-rate training that prepares them to work in the more demanding work environment of today. Employers who contribute get the kind of trained force they desperately need. It's a win/win scenario for all concerned. (See page 14 for more information on Tech Prep.)

Of course, a program like Tech Prep requires an enormous commitment on the part of all involved. Educators have to be willing to make curriculum and program changes to meet the requirements of the industry sponsors. Students have to be committed to a lot of hard work to succeed in the program. Participating companies have to devote time, personnel, equipment, space and money to make the system work.

PUBLISHER'S PAGE

A tall order? No doubt. But talking to people involved in this experiment convinces me that this program and others like it have a good chance for success. Tech Prep is a realistic and effective response to a problem that needs addressing now. Some gear companies are already exploring these programs. The rest should follow suit.

Training a work force for the next century requires rethinking cherished assumptions, readjusting attitudes, and putting our time, money, capital and human resources where our mouths are. The trained work force we need is not going to arrive on our doorsteps gift-wrapped; we are going to have to go out and get it. We as employers hold a key piece of the trained work force puzzle — the specific knowledge of the kinds of skills we need and the ability to influence schools to provide it. Without our contribution through programs like Tech Prep, we remain part of the problem, not part of the solution.

Michael Goldstein

Michael Goldstein,
Editor-in-Chief

Choosing An ISO 9000 Consultant: Why, When & How

Amy Zuckerman

One of the key questions confronting any company considering ISO 9000 certification is, how much is this going to cost? The up-front fees are only the beginning. Dissect the ISO 9000 certification procedure with an eye for hidden costs, and two segments of the process will leap out — the cost of consultants and the cost of making in-house improvements for the sake of passing certification. Most of of these costs can be controlled by careful selection of the right consultant in the first place.

Because there is no regulating body in the United States setting rates for consulting fees, those fees will vary. The lack of regulation also means that there is no organization to fall back upon when and if the decision is made to hire a consultant. The American Society for Quality Control will release a list of consultants who work in the quality field. But unlike the British National Registration Scheme, the ASQC does not screen consultants or offer certification to those who meet its standards.

The lack of similar regulation and standards in the U.S. bothers officials of the U.S. Registrar Accreditation Board. The American RAB, based in Wisconsin, advises U.S. companies thinking of hiring a consultant to learn as much about potential consultants as they can ahead of time, and not just from brochures and sales pitches. Get references from companies that have employed them in the past and check for their experience outside of ISO 9000.

> **Hire a consultant as you would a vendor: Do your homework; demand references; talk to other customers.**

Do I Have To Hire A Consultant At All?

Can you avoid hiring a consultant? The answer depends on your company, its makeup and the circumstances surrounding your decision to seek ISO 9000 certification. Many large corporations have sufficient staff levels to free up several employees, send them to seminars on ISO 9000 certification and work through the implementation process on their own. Small and mid-size companies may not have this option and may very well need some assistance from a qualified expert.

And how does one go

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Amy Zuckerman
is co-principal in IN/EX Information Export, a marketing consulting firm in Pelham, MA. She is the author of ISO 9000 Made Easy: A Self-Help Guide to Certification.

about determining who is a qualified expert or deciding whether his/her services are necessary?

Before making this all-important decision, it's crucial to understand how quality ISO 9000 consultants can operate to assist you. There is no single way or method consultants should adopt, nor can they assist every company in the exact same way. But it is possible to determine when a consultant is taking advantage of you through over-advising and increasing the workload or by under-advising or neglecting your concerns.

What Can A Consultant Do For Me?

Consultants should act as facilitators of an in-company process and encourage clients to help themselves as much as possible. A good consultant will not attempt to take over the total pre-certification process to earn higher fees. A quality consultant will realize that quality standards and procedures must be generated from within the organization and become a working part of it.

Consultants can help set direction, determine which form of ISO certification to pursue, provide guidance, encouragement and structure. But it's up to each company to design its own procedures, determine how to implement quality standards and then continue to live up to those standards during the post-audit and post-certification years. These standards should make sense for the company's long-term operation, not to simply pass ISO 9000 certification.

A good ISO 9000 consultant will constantly emphasize that quality emanates from within, not from the consultant. The company must create its own quality procedures while the consultant can merely guide the process. For this reason, be wary of consultants trying to oversell the certification process. Like any honest salesperson, a good ISO 9000 consultant should want to truly assist a client, not push to earn an unwarranted fee.

Here are some ways you can best utilize a consultant's services:

✔ Consultants may be brought in-house to assess whether a company is an appropriate candidate for ISO 9000 certification and how well this company would perform against ISO 9000 quality standards.

✔ A good consultant can translate the difference between your current quality system and the ISO 9000 standards — should there be one — into an action plan of items to be accomplished before the real ISO 9000 registrar is brought in for the certification audit.

✔ The consultant can work together with company personnel on implementing the action plan. This may mean training personnel to maintain new quality standards, establishing new operating procedures, advising on equipment purchases, recalibration of measuring equipment or any other change that will bring the operation up to acceptable levels under ISO 9000.

✔ A consultant may set up a checks-and-balances system within the organization, thus allowing employees to monitor quality from within.

✔ The consultant may assist the company in applying to the RAB for certification and in hiring a registrar to conduct the certification audit.

✔ Some consultants will act as an "internal" inspector, conducting a pre-audit to insure that all necessary changes have been implemented and that the operation will pass its ISO 9000 certification.

The absolute first step any consultant should take should be to assess whether certification is necessary or even desirable for a particular company. Companies should be advised to look at ISO 9000 like any investment: How much can you afford to pay, and will the ends warrant the expenditure? Be wary of consultants pressuring you to seek certification regardless of your circumstances. Instead, like any good advisor, a quality ISO 9000 consultant should assist you to sort out an appropriate direction.

The kinds of questions he/she should pose to insure that ISO 9000 certification is appropriate for your company appear in the sidebar on the next page.

What Happens Next?

If a consultant, upon posing these questions, finds that a company is operating at top ISO 9000 levels, he or she should encourage the company to seek an early audit and bow out of the process. The consultant should also bow out if he or she discovers that an alternate form of certification would be preferable to ISO 9000.

To conduct an audit, the consultant must conduct interviews with crucial members of the company's staff. A good rule of thumb for determining the time needed for this preliminary process is to divide the

If your company is operating at ISO 9000 levels, your consultant should encourage an early audit and then bow out of the picture.

The ISO 9000 Quiz

Your ISO 9000 consultant should be asking the following kinds of questions:

1. What is your company's quality policy?
2. How do you determine your customers' needs?
3. Do you have a structured means of retrieving information from your customers?
4. Who is responsible for maintaining quality control in your organization?
5. How do you assure quality?
6. Are all the people in your employ aware of quality and the need for maintaining strict quality control throughout the entire operation?
7. Are all the people in your employ aware of your company's quality procedures?
8. Are your employees well-trained or educated for their current positions?
9. How do you currently measure the performance of your suppliers?
10. How do you keep up with the quality standards within your industry?
11. How do you measure accuracy, whether you deal in a product or a service?
12. How pervasive are your accuracy procedures?
13. How are the results registered?
14. Who is in charge of non-conforming products?
15. Are customer complaints being registered in a structured manner?
16. How is corrective action taken?
17. How well are you controlling these processes (Nos. 12–16)?
18. What kind of measurement equipment is being used (manufacturer)? How do you measure customer satisfaction (service industry)?
19. How often do you calibrate your measuring equipment?
20. Are all of the above (Nos. 12-19) formalized through procedures?
21. Who is responsible for maintaining them?
22. Who initiates improvements?
23. How are suggestions for improvement handled?

number of employees by six (for six interviews a day). So it would take about four days to complete an audit with a company of 24 employees.

The consultant must then draw up a report of findings, which would add another six or so hours to the 24 already spent with the client. This report should include a list of actions required for the company to meet ISO 9000 standards. At this point the consultant has worked about 30 hours. Depending on the client's needs and staff levels, a consultant can be maintained in-house for the duration of the certification process or to act as an occasional advisor.

A Case In Point

A Massachusetts precision machine company with 35 employees is a good example of how a consultant can and should help a company through the ISO 9000 process. A consultant was brought into assess this company's quality levels and determined that the shop was operating at close to ISO 9000 levels. To pass certification, however, this company was advised to better document its quality procedures. The consultant in question offered the shop owner the choice of appointing an in-house ISO 9000 expert — thus going it alone — or hiring him on for additional hours. The owner chose to hire the consultant to assist with changes in his quality manual on a one-time basis. Cost savings in consulting hours were insured by this approach.

The machine shop owner did recognize, however, that the consultant was able to speed up the certification process by acting as a pre-auditor and by serving as an advisor in the creation of the ISO 9000 company implementation plan required for certification.

The Final Analysis

To hire or not to hire an outside consultant to help you with ISO 9000 decision-making remains the question. The answer must be based on an honest assessment of your company's capabilities. Do you have the time and resources to go it alone?

Once you have made a decision to seek outside help, evaluate potential candidates for the job with the same care you give to any other vendor. Ask for references and recommendations. Don't be snowed by ISO 9000 hype. Scrutinize proposals with care. Ask whether a particular part of the process is really necessary in your circumstances. Look for the consultant who is interested in helping you achieve your ISO 9000 goals, rather than in just making a big profit. Investment in such a consultant can be money well spent. ■

NEWS BRIEF

The Tech Prep Approach to Worker Training

Industry and Education Combine Forces to Close the Training Gap.

Nancy Bartels

"More than half our young people leave school without the knowledge or foundation required to find and hold a job."according to a 1991 report from the U.S. Dept. of Labor. A huge gap exists between the needs of employers (especially in manufacturing) and the training received by most high school students.

This is not good news for employers, especially in the manufacturing sector, for their managers, who need skilled workers to keep their departments running at peak efficiency, or for the high school graduates who find that their diplomas have not prepared them to earn a decent living. However, just in the last few years, more than a half a million young people and the employers and managers for whom they work have found a bridge across this training gap with the help of a program called Tech Prep.

The Tech Prep Idea

The idea behind the program is a simple one: Employers and schools must work together to change the way we educate most of our students today. Eighty percent of the jobs in this country do not require a college degree, but they do require some kind of advanced trianing; however, at present, students in "general" curricula, (those not bound for college) are not trained for any of these jobs. The goal of present high school programs seems to be to qualify students for a diploma, not to train a work force.

Tech Prep addresses this issue through the democratic notion of "grass roots" efforts on the part of the people involved — in this case educators, employers, students and parents. In Tech Prep, local employers discuss their particular training needs with the high schools and community colleges in their area and then develop a program to meet those needs.

One such program is now in the works in north central Illinois. There, Rock Valley College, several Rockford area high schools and six local employers (including a gear tool manufacturer) have combined their resources to start a Youth Apprenticeship Program. This program recruits students after their tenth grade year and places them in an intensive work/study program that combines a rigorous academic schedule with on-the-job training at the facilities of one of the six sponsoring companies. The program carries students through their high school years and continues as they pursue studies at Rock Valley College and into full-time apprenticeships.

Youth Apprenticeship Training Program

This is how it works. The summer after their junior year, the Rock Valley program recruits begin full-time (40 hours a week for at least six weeks) work/training at all the sponsor sites. During their senior year, they work four hours a day with an individual sponsor while maintaining a regular academic schedule. After graduation, qualifying students move to full-time apprenticeships, with credit being given for their Youth Apprenticeship hours. They also have the option of continuing their education toward an associate's degree (or a baccalaureate) while working as full-time apprentices.

Students in the program get paid for the work they do while learning. They also explore the variety of opportunities available in manufacturing and acquire skills essential in the workplace now. At the same time, employers get a supply of skilled, capable employees, trained for present jobs in their factories and ready to take on new challenges.

The Basic Ingredients

Other programs in other places are tailored to the particular needs of their developers, but whatever the externals, each one incorporates the following characteristics:

• Contextual learning. The emphasis is on applied academics — how math, science and communication skills are usable in "real world" settings. Content is academically rigorous, but also connected to actual workplace experiences. The principles of productivity, teamwork, and flexibility that are so important to the modern workplace are also emphasized.

• Local Partnerships. Employers, labor and community leaders, parents and educators are all part of the planning and implementation of local programs. Broad-based local support is essential to the success of Tech Prep programs.

• Career exploration and counseling. One of the goals of Tech Prep programs is to help students to make intelligent career choices and explore options that best suit their aptitudes.

• Advanced degree potential. Foundational skills are solid enough to prepare students for either associate or baccalaureate degrees if they wish.

• A "bridge" program. Participating groups have also developed a variety of internship and other work/study and "bridge" programs addressed to the needs of older workers who have already left school and may need some academic "refresher" courses to succeed in the advanced associate degree programs required to upgrade their skills.

Flexibility and Options

One of the virtues of the Tech Prep program is its versatility. Programs are developed by the people who will use them to address their specific needs. All the programs begin with extensive conversations between local employers, educators, labor and community leaders, and students and parents to assess needs and plan systems that best fit the requirements of particular communities. Extensive community support is needed in order to make the program work, and development of "ownership" on the part of all participants is crucial to program success. Company size is no deterrent to participation. In fact, smaller companies without the resources to provide elaborate training of their own may find a Tech Prep alliance particularly useful.

Beginnings

The "father" of Tech Prep is Dale Parnell, an educator and president of the American Association of Community and Junior Colleges. He first outlined the idea in his book *The Neglected Majority*. Parnell and engineer Dan Hull further refined the idea of building alliances between high schools, community colleges and employers, and in the late 1980s Parnell established the Center for Occupational Research and Development (CORD), a non-profit organization with headquarters in Waco, TX, devoted to fostering the Tech Prep idea.

The program was given more impetus by the passage in 1990 of the Carl D. Perkins Vocational and Applied Technology Education Act, which provides funds for programs addressing worker education, including Tech Prep.

In 1991, CORD organized the National Tech Prep Network to support local Tech Prep programs through information-sharing, networking, conferences, publications, and publicity. The NTPN also has a data base of information on various local programs and will arrange tours of model pilot programs in various parts of the country for groups interested in developing their own Tech Prep programs. A call to NTPN is frequently the first step in getting a local Tech Prep program off the ground.

For a $95.00 membership fee individuals in industry and business, education, community organizations and local, state and national government, can become part of the network and make use of all its programs and services.

A Drop In The Bucket

One estimate of employment trends suggests that by the year 2000, 15,000,000 manufacturing jobs will require advanced technical skills. At the same time 15,000,000 service jobs will disappear from the economy. These numbers suggest that Tech Prep, and programs like it, have a long way to go if they are to meet current and future demands for skilled labor.

What they are attempting is nothing less than a reform of general high school education and worker training nationwide. While the undertaking may seem visionary, it is also essential if the country is to maintain its competitive edge in the future. It is a chance for manufacturers of all types and sizes to take a pro-active stance in developing the skilled work force they need to remain competitive. ■

Call the NTPN at 1-800-972-2766 for more information about Tech Prep.

Nancy Bartels
is Gear Technology's Senior Editor. She has written many articles on issues of importance to business and industry.

Carbide Rehobbing — A New Technology That Works!

Fred Young
Forest City Gear
Roscoe, IL

Many people in the gear industry have heard of skiving, a process wherein solid carbide or inserted carbide blade hobs with 15 – 60° of negative rake are used to recut gears to 62 Rc. The topic of this article is the use of neutral (zero) rake solid carbide hobs to remove heat treat distortion, achieving accuracies of AGMA 8 to AGMA 14, DIN 10-5 and improving surface finish on gears from 8 DP – 96DP (.3 module – .26 m.).

Early technology developed with Azumi skiving hobs yielded encouraging results. However, few people seemed to adopt this process in lieu of gear grinding. Among the drawbacks were the necessity of having extremely rigid hobbing machines and work fixturing, expensive tooth timing, centering devices, expensive cutters, and the difficulty of re-sharpening the cutters properly to maintain the correct involute.

Some of these problems have been eliminated by using zero-rake solid carbide hobs which are available from many of the major hob manufacturers. This eliminates the need to offset the diamond wheel when resharpening on a hob sharpener, reduces the initial cost of producing the cutter and increases cutter life. Another improvement is the common use of non-contact, electro-magnetic sensors which automatically divide the stock on the tooth flanks with extreme accuracy. If the quality of the gear is excellent initially and heat treat distortion is minimal, it is possible to recut gears with as little as .0015 – .003" over wires as a salvage method for gears which grew unexpectedly during heat treat.

Ideally gears should be roughed with protuberance hobs which have any desired profile modification built in. If the part is crowned, it may be beneficial to crown during roughing so that an even amount of stock is removed from the flanks, leaving a uniform case hardness depth. Helical gears often unwind during heat treat, and accommodation may be made to increase the helix angle. For fine pitch, we start with approximately 0°15' as an offset. Remember that if a protuberance rougher is used, it generally only produces the correct involute in a sometimes narrow range of teeth.

Our experience is limited to the carbide

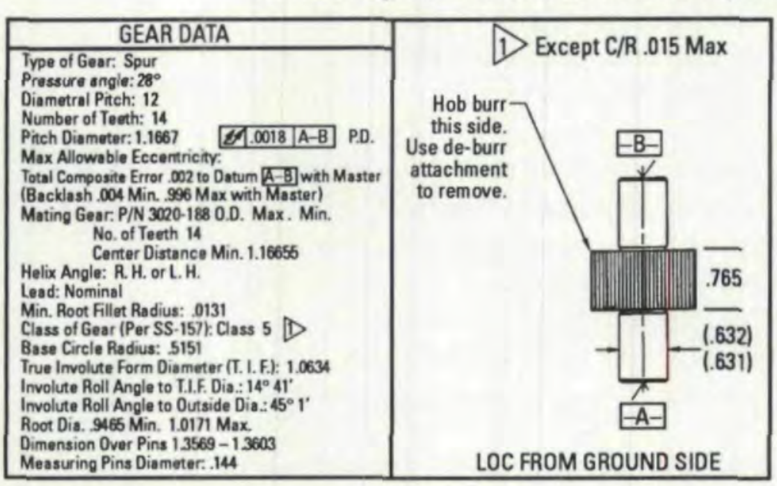


Fig.1 – Case 1

rehobbing of gears ranging from 8DP – 96DP and less than 4" (100 mm.) in diameter, although many people have skive-hobbed much coarser, larger gears to 40 mm module. In fact a number of manufacturers skive coarse pitch gears to reduce finish grinding times.

Remember that proper gear blank finishing is crucial for achieving high quality levels for carbide rehobbing, just as it is vital to gear grinding. Bore-type blanks should be honed or I.D.-ground, and faces should be parallel and perpendicular to the bore. Similarly, shafts should be cylindrically ground to provide true locating diameters to the centers for cutting and later inspection.

Since carbide can be susceptible to cracking, later vintage hobbing machines with backlash-eliminating features, power tailstock clamping and CNC controls multiply dramatically achievable production rates and accuracies as well as prevent hob damage. As reported in a January, 1993, paper, "Skive Hobbing Hardened Gears" by Hans Glatzeder, rigid machines can carbide rehob to 2/3 their capacity diameter and module (Ref. 1).

Cutting tools should be A or AA quality and preferably titanium carbo-nitride (TiCN) coated to improve lubricity. Either face or bore keys are suggested to increase drive rigidity. Typical carbides are ISO K10, K05, M15, M10 and P20 (Tables I – IV). Depending on pitch, stock removal, hardness and whether protuberance hobs were used to rough, surface speeds range from 90 – 340 m/min. Recommended stock removal is .002 – .006" (.05 – .15 mm.). Required quality may dictate a finishing pass. In this case it is beneficial to climb-cut on one pass and conventionally cut the other to reduce cutter wear and improve quality. Experimentation on helical gears is warranted to determine if opposite-hand hobbing; i.e., left-hand hob, right-hand gear or vice versa, will yield better results. Close observation of the cutter and gear quality will help determine shift and sharpening intervals. On fine and medium pitch gears, we expect at least 2000 pieces per hob or more, depending on hob size and part hardness, unless suffering a crash. Microfinish is determined by the number of flutes in the cutter versus the number of teeth in the gear, the pressure angle, feed rates,

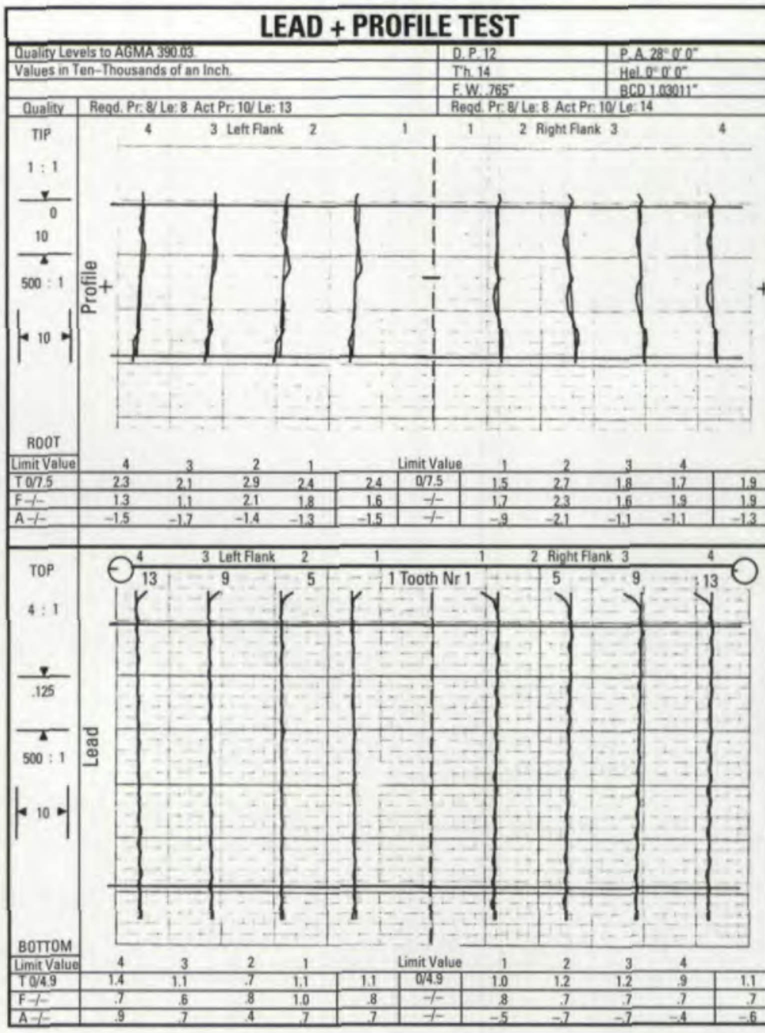


Fig.2 – Case 1

material cut, hardness and coolant. Surprisingly, carbide-hobbed finishes approach ground finshes, often around 16 microinches. Flank wear of .002 – .006" (.05 – .15 mm.) per hob flank is a suggested starting point in fine and medium pitches before shifting and sharpening. Because TiCN does not lend itself to recoating, and regrinding the hob profile is not easily done, we have not used recoating.

So far only recutting has been mentioned. It should be noted that many manufacturers are carbide hobbing from solid. In fine to medium pitches, this is practical in the 40 – 50 Rc range, especially where small quantities are involved, and hob spindle rpms may be reduced to under 100. Hobbing from the solid is generally more costly than carbide rehobbing rough hobbed gears due to tooling and machine time costs.

Most of our carbide rehobbing has been done on mechanical hobbers which realign the hob tangentially after the machine has been initialized with the home position for hob and

Fred Young

is the president of Forest City Gear Co, a fine and medium pitch gear job shop. He was the Chairman of AGMA's Small Business Committee and the author of several papers on gearing subjects.

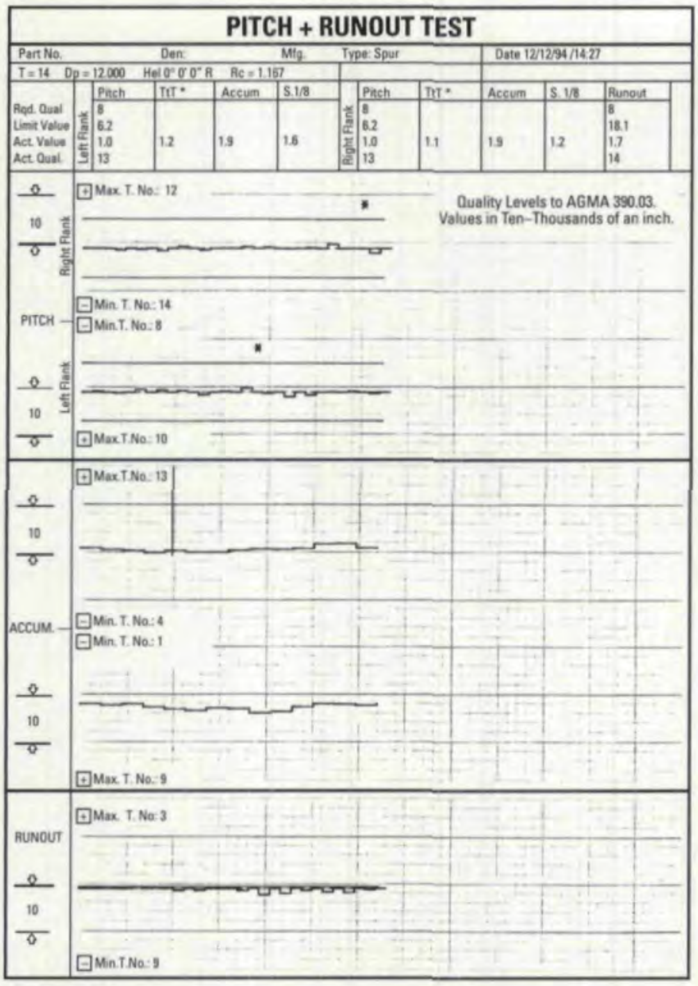


Fig.3 – Case 1

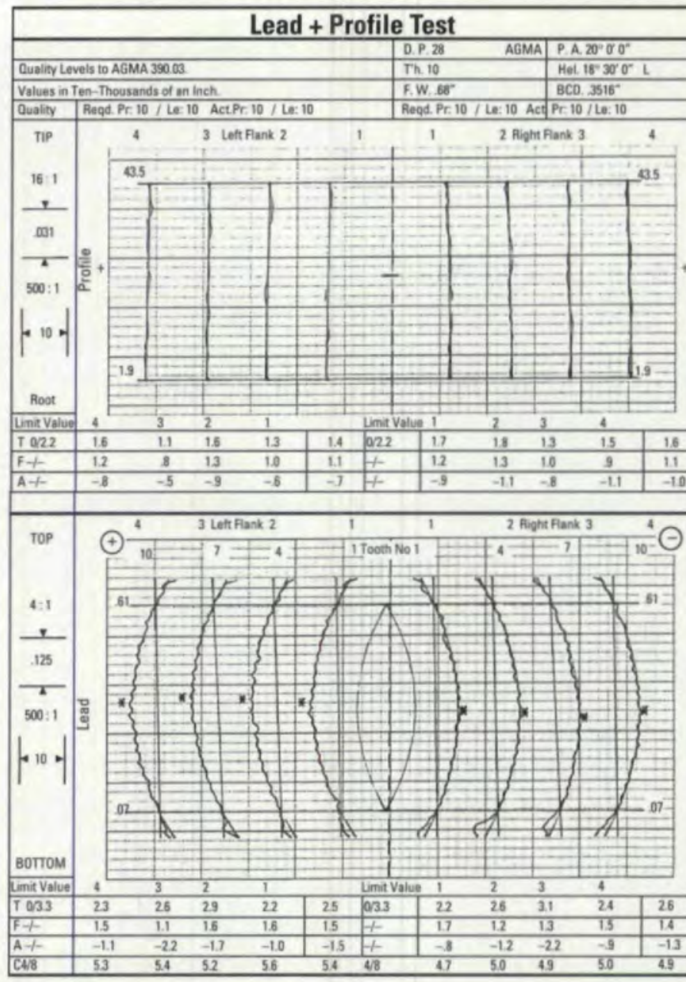


Fig.4 – Case 2

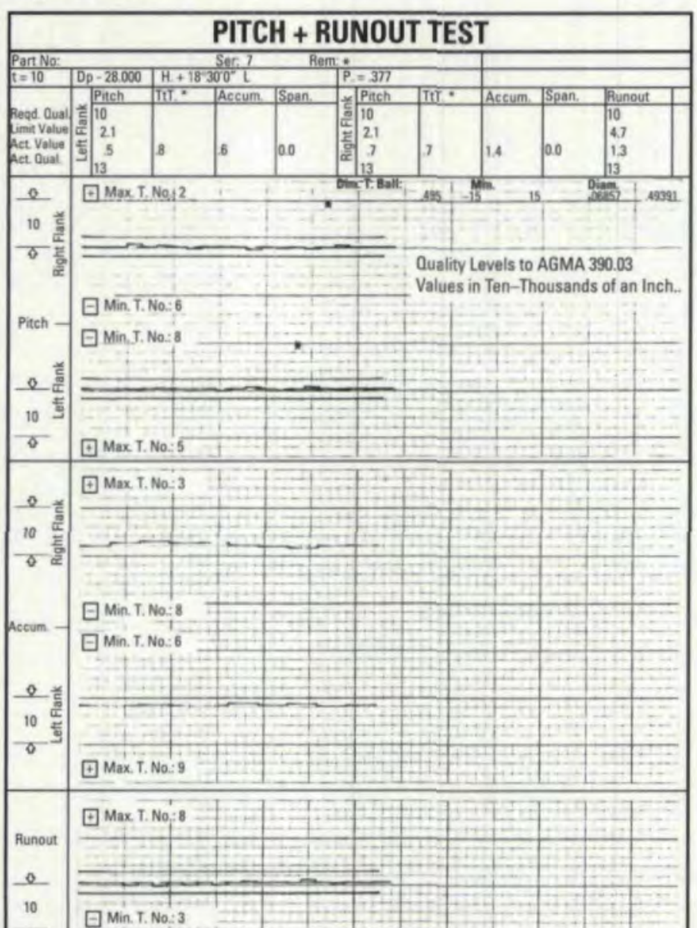
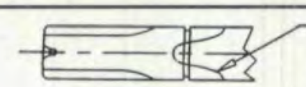


Fig.5 – Case 2



Item	Spur or Helical Gear Data	Specification	Insp.
1	Type	Helical	X
2	Normal diametral pitch	28	
3	Number of teeth	10	X
4	Normal pressure angle	20°	
5	Helix angle	18° 30'00"	
6	Hand of helix	Left	
7	Standard pitch diameter	.3766	
8	Whole depth	.0803	X
9	AGMA quality number	10	
10	Total composite tolerance	.001031	X
11	Tooth-to-tooth composite tolerance	.000712	X
12	Master part number		
13	Meas. over .06857 dia. (2) wires	.4965 – .4935	X
14	Lead	3.536026	
15	Lead tolerance	.000323	X
16	Profile tolerance	.000216	X
17	Roll angle (functional profile)	1.8943°	X
19	Outside dia. of gear 5	.4715 ± .000 .006	X
20	Center distance (functional)	.96034	
21	Circular tooth thickness	.0677 – .0658	
22	Backlash	.003 – .0003	
23	Tooth form (see detail " ")		
25	Modification	.0115	
26	Base dia.	.3516	
27	Mating gear	699607	
28	Min root dia.	.305	
29	Pinion must have .0004 to .0008 crown		X

Fig.6 – Case 2

work piece stored from a setup part. Hob shift limits are established over a circular pitch range, shifting the hob in and out to realign from the timed position established by the proximity device located over the top of the workpiece. When the hob is thoroughly worn in a given area, the hob is shifted a full circular pitch and continues across the hob face. Care must be taken to cut reasonably good parts in roughing and to insure that no severe damage occurs in heat treat. If a CNC hobber can be used, better profile results are possible because the workpiece is rotated rather than shifting the hob, which may degrade the profile.

Following are some histories which should make a strong case for the cost-effectiveness of carbide rehobbing as a substitute for gear grinding. In fact, many applications would be impractical to grind because of wheel diameter and cost. Keep in mind this hobbing was done on non-CNC, mechanical hobbers, costing \$250,000 – \$350,000, versus gear grinders, which usually cost \$1,000,000.

Case History 1. Aerospace high pressure pump gear (Figs. 1–3). Fourteen teeth, 12 DP, 28° PA, .765" face width, spur, hardened to 55 – 60 Rc.

A. Rough cut-double cut, .050/.030" (1.27 – .76 mm.) feed at 350 rpm with a 2" (50 mm.) diameter hob-33/hr. Normal tooth thickness, .130/.132 (3.3 mm.).

B. Finish-cut 400 rpm, feed .015" (.38 mm.), 2" (50 mm.) diameter cutter, TiCn coated, production 14/hr. auto-loading.

Case History 2. Power tool application (Figs 4–6). Ten teeth, 28 DP, 20° PA, 18.58°, .68" (17.3 mm.) helical face width, hardened to 48 – 53 Rc, material 41L50.

A. Rough hobbed 800 rpm, .045" (1.14 mm.) feed, .070/.072" (1.6 mm.) Normal tooth thickness, 100/hr.

B. Finish crown hobbed 500 rpm, .025" (.63 mm.) feed, .0642/.0625" (1.6mm.) normal tooth thickness, 60/hr.

Case History 3. Aerospace application (Figs. 7-9). Forty teeth, 32 DP, 20° PA, .325" face spur material 431 cres, heat treat 210/245,000 PSI, AGMA 10.

A. Rough hob 900 rpm/.050 (1.27 mm.) feed, 1 7/8 diameter M42 hob TiN, production 100/hour.

B. Finish carbide TiCn, 530 RPM, .025"

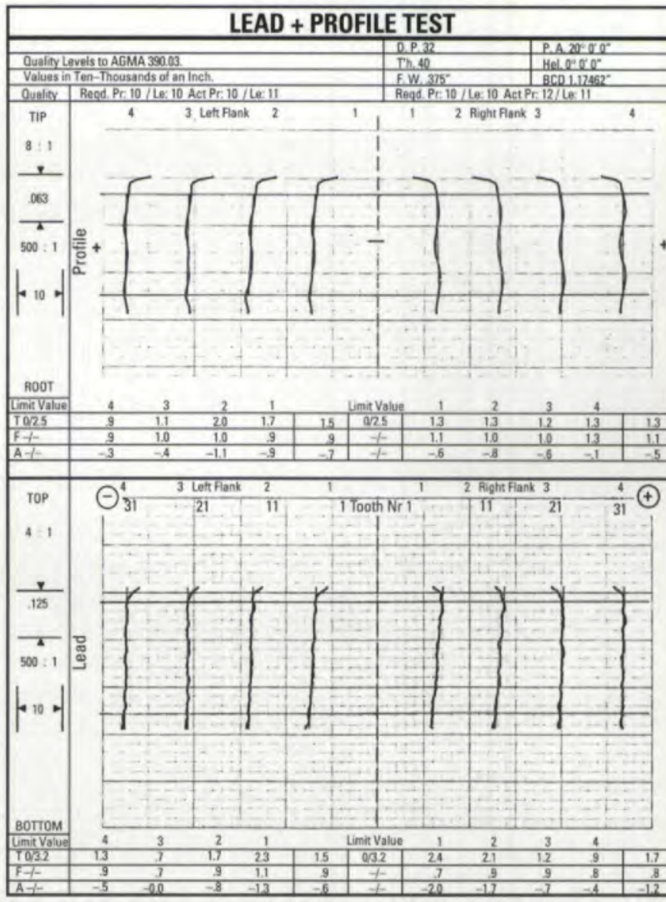


Fig.7 – Case 3

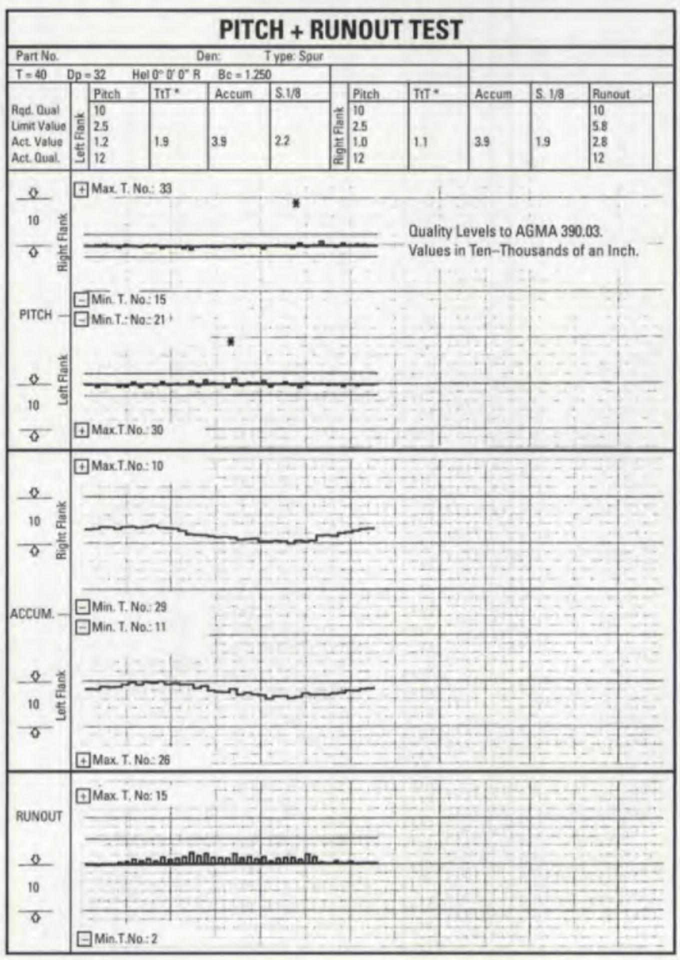


Fig.8 – Case 3

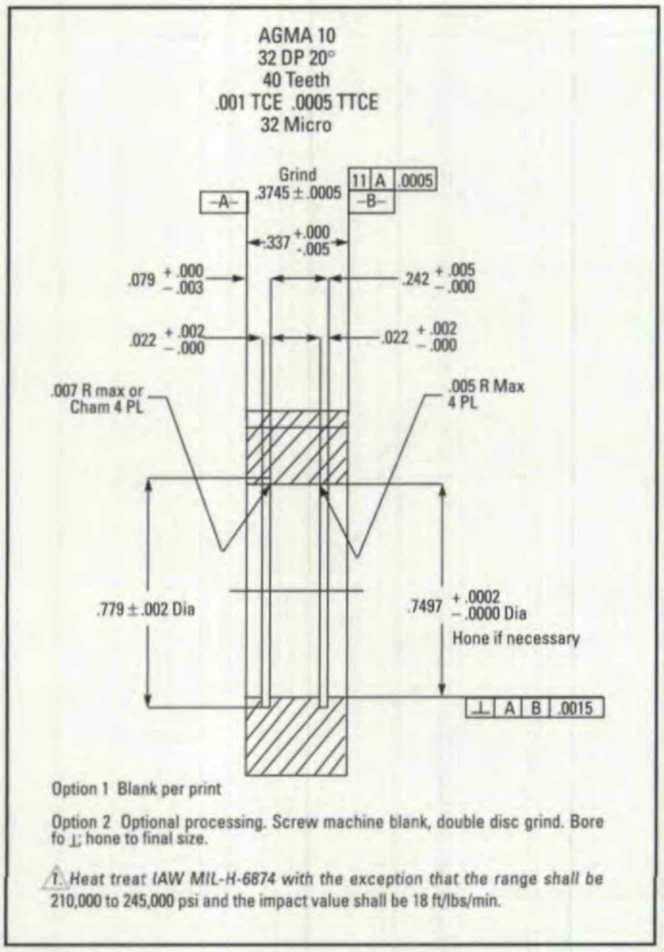


Fig.9 – Case 3

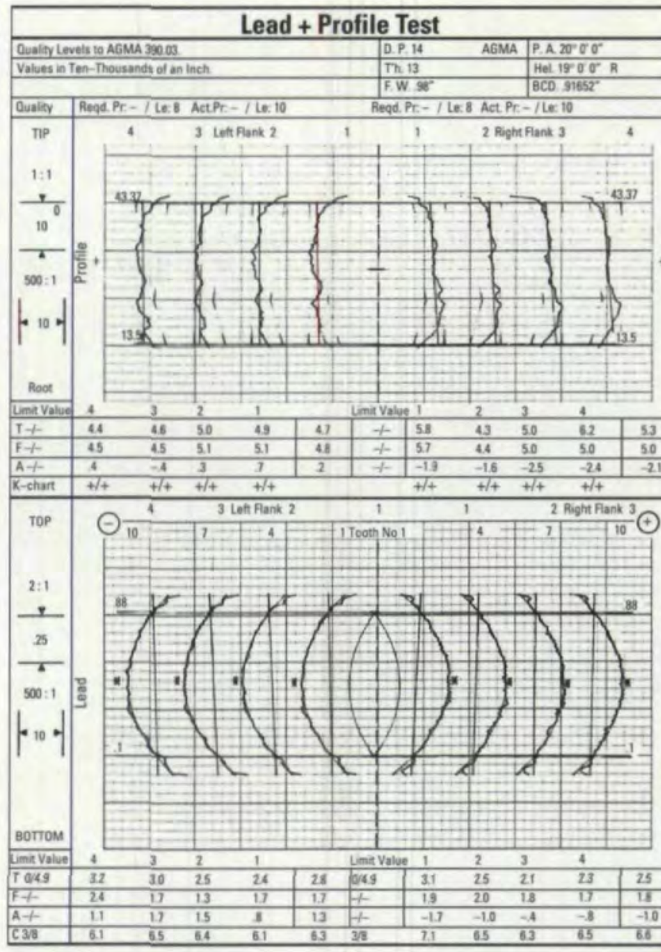


Fig.10 – Case 4

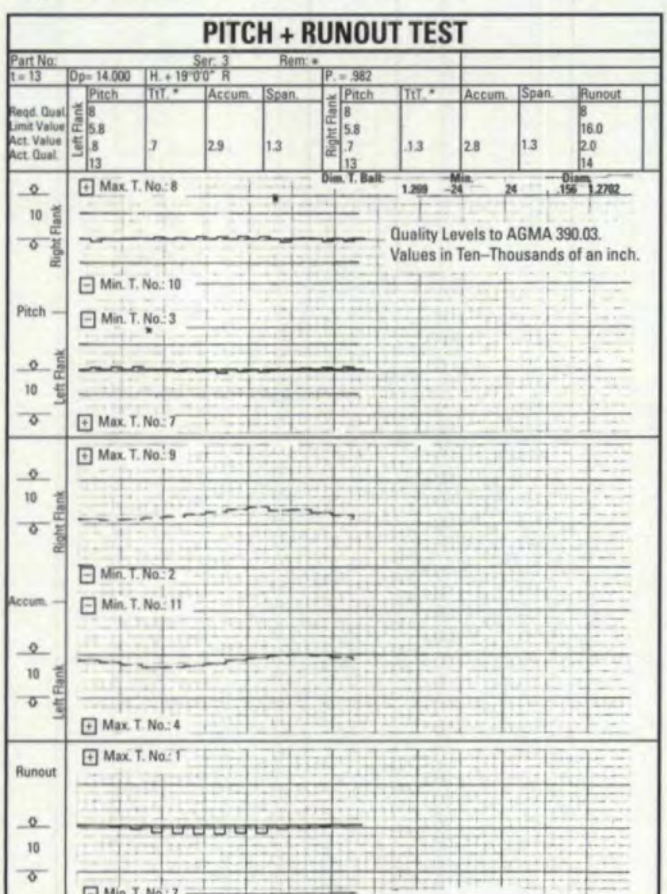


Fig.11 – Case 4

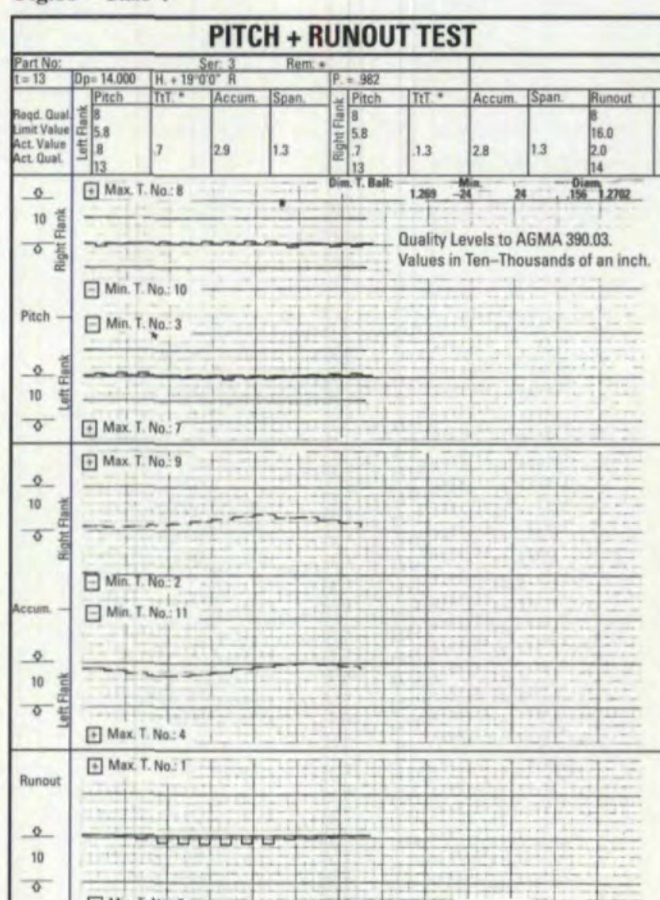


Fig.12 – Case 5

(.64 mm.) feed, 1 7/8 hob diameter, 28/hr.

Case History 4. Lift truck motor armature (Figs. 10–11). Thirteen teeth, 14 DP, 20° PA, 18.83° helical, .97 (24.6 mm.) face width, 86L20 material, induction hardened gear 58 – 63 Rc.

A. Rough hob 1" diameter hob, 800 rpm, .040 feed, 67/hr. Normal tooth thickness, .1406/.1387.

B. Finish hob 350 rpm, .025 (.64 mm.) feed, crowned, .0002/.001" (.005 – .025 mm.) 24/hr.

Case History 5. Lift truck armature shaft. (Figs. 12-13). Thirteen teeth, 14 DP, 20°PA, 14.75° helix, face 1.3" (33 mm.), crown-hobbed to AGMA 8.

A. Rough hob 600 rpm, .055" (1.4 mm.) feed, 1-7/8" (47.6 mm.) diameter hob TiCn, 45/hr.

B. Finish hob 325 rpm, .025" (.64 mm.) feed, 1-7/8" (47.6 mm.) diameter cutter TiCn 17/23/64. Normal tooth thickness, .1233/.1208 (3.1 mm.).

C. 14T, 16/32DP, 30°PA spline, 2.12" (53.8 mm.) face-finish splined, 22/48 Rc core hardness with carbide TiCN 1-7/8" (47.6 mm.) diameter hob, 34/hr.

For variation, Fig. 14 depicts a cast tooth form pump gear re-hobbed in alloy C

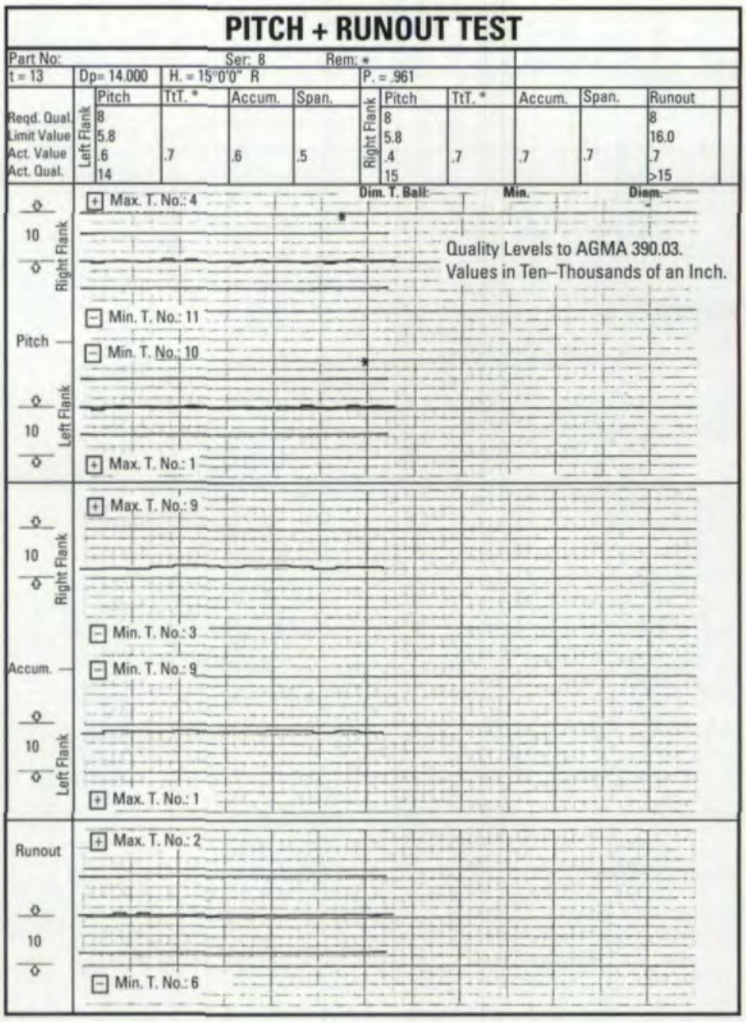


Fig.13 – Case 5

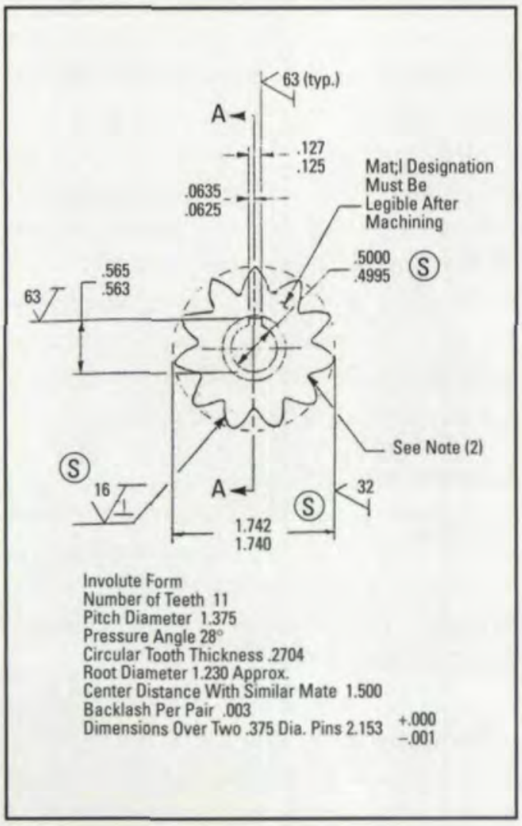


Fig.14

Table I

ISO	Rockwell Hardness	Deflection Resistence	W	Co	Ti	Ta	C
P20	90	90	60 83	5 10	5 15	0 15	6 9
M10	91.5	100	70 86	4 9	3 11	0 11	6 8
M15	89.5 93	120 220	75 95	5 9	0 10	0 12	5 7
K05	89 93	150 230	85 97	3 8	0 3	0 7	5 7
K10	90.5	120	84 90	4 7	0 1	0 2	5 6

Table II

Feed Module (DP)	Rough mm (in)	Finish mm (in)
> 12 [< 2]	3 – 4mm/rev (.120 – .160"/rev)	2 – 3mm/rev (.080 – .120"/rev)
> 12 [< 2]	2 – 3.5mm/rev (.008 – .140"/rev)	1.5 – 2.5mm/rev (.060 – .100"rev)

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Table III

Hardness HCR	Speed mm/min (in/min)
50–55	70 + 90 (220 – 290)
55–60	60 + 70 (190 – 220)
60–65	50 + 60 (160 – 190)

Table IV

Module DP	Speed mm/min (in/min)
1 + 5 (5 – 25)	60 + 90 (190 – 290)
6 + 12 (2 – 4)	50 + 70 (160 – 220)
> 12 (<2)	30 + 50 96 – 160)

Hastalloy. This presented a problem for the electromagnetic sensor, since this material is non-magnetic.

Tables I–IV (Ref. 2) define typical carbide grades, liberal feed rates for roughing and finishing and suggested speeds versus hardness and speed versus module. We tend to be more conservative with our speed and feeds to save hob wear and improve quality.

In conclusion, carbide rehobbing is an extremely economical approach to removing distortion caused by heat treatment. Besides approaching gear ground quality levels and finishes, carbide rehobbing can be used with small hobs capable of approaching blend-cut shoulders, which larger grinding wheels, including double helicals, cannot reach. Hardware cost is fractional, since the same machine can be used for roughing and finishing and mass production is readily obtainable with automatic loading. This process is a viable alternative to gear grinding. ■

References:

1. Glatzeder, Hans. "Skive Hobbing Hardened Gears," Liebherr Versahntechnik GMBH.
2. McElroy, William E. "Using Hobs for Skiving: A Pre-Finish and Finishing Solution." *Gear Technology*, May/June, 1993.

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DON'T MISS IT.

Minimizing Backlash in Spur Gears

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Nomenclature

B — Arc on both reference circles that corresponds to B_b on both base circles

B_b — Backlash on line of action

C — Distance between centers of reference circles

C_b — Basic center distance, $(N + n)/2P$ or $m(N + n)/2$

m — Module

N — Number of teeth on gear

n — Number of teeth on pinion

P — Diametral Pitch

p — Circular pitch on both reference circles, π/P or πm

$p/2$ — basic tooth thickness

R — Radius of gear reference circle, $N/2P$ or $mN/2$

r — Radius of pinion reference circle, $n/2P$ or $mn/2$

R_b — Radius of gear base circle

r_b — Radius of pinion base circle

S — Arc on both reference circles that corresponds to S_b on both base circles

S_b — Space on line of action due to an increase in center distance

ΔT — Deviation from $p/2$ on gear reference circle

Δt — Deviation from $p/2$ on pinion reference circle

ΔT_b — Change in tooth thickness on gear base circle

Δt_b — Change in tooth thickness on pinion base circle

Φ — Profile angle of generating rack

ϕ — Pressure angle

Abstract

Simplified equations for backlash and roll test center distance are derived. Unknown errors in measured tooth thickness are investigated. Master gear design is outlined, and an alternative to the master gear method is described. Defects in the test radius method are enumerated. Procedures for calculating backlash and for preventing significant errors in measurement are presented.

Introduction

An important part of designing for minimum backlash is the prevention of significant errors in measurement. Circular tooth thickness, for instance, is not measured directly, but is calculated from an equation that presumes perfect teeth. As a result, a master gear — even if perfect — may not indicate the center distance at which imperfect gears will mesh tightly with each other. Moreover, the magnitude of the error in measured tooth thickness varies with the number of teeth on the master gear.

Measured tooth thickness also varies with depth of contact and active face width, so the master gear should be representative of the gear that mates with the work gear. But most measurements are made with general purpose, off-the-shelf master gears.

Significant errors exist in many, if not most, roll test measurements. Evidence for this is the discrepancies in measurement that arise when the same gear is inspected by different people (buyer and seller, inspection and production departments, different inspectors using the same test equipment, etc.). The typical response has been to standardize the test equipment and inspection procedure — a maneuver which can reduce the discrepancies in measurement, but not necessarily the errors in measurement. The purpose of this article is to show how to deal with errors in measurement when designing for minimum backlash.

Basic Geometry

A very simple way of treating backlash is to start with no backlash (Fig. 1) and then examine the effect of an increase in center distance (Fig. 2).

In Fig. 1 the terms *basic center distance* (C_b), *profile angle* (Φ) and *reference circle* — which have been around for at least 30 years (Refs. 1–2) — are used in lieu of *standard center distance, standard pressure angle* and *standard pitch circle*. Consequently, in Fig. 2 there is no need for the qualifying adjective *operating*, which means that *operating center distance, operating pressure angle* and *operating pitch circle* can be shortened to *center distance* (C), *pressure angle* (ϕ) and *pitch circle*.

From Fig. 1 it is seen that

$$x + y = R_b \tan\Phi + r_b \tan\Phi,$$

and from Fig. 2 that

$$R_b(\phi - \Phi) + x + \frac{S_b}{2} + y + r_b(\phi - \Phi)$$

$$= R_b \tan\phi + r_b \tan\phi,$$

which, upon substitution for $x + y$, becomes

$$\frac{S_b}{2} + R_b(\tan\Phi - \Phi) + r_b(\tan\Phi - \Phi) =$$

$$R_b(\tan\phi - \phi) + r_b(\tan\phi - \phi),$$

where $\tan\Phi - \Phi = \text{inv}\,\Phi$ and $\tan\phi - \phi = \text{inv}\,\phi$, so that

$$S_b = 2(R_b + r_b)(\text{inv}\,\phi - \text{inv}\,\Phi). \quad (1)$$

This equation was first derived in a somewhat different way by Candee (Ref. 3), who remarked that it is "the shortest and most direct equation," and "does not require the determination of tooth thickness on new pitch circles" — an apparent reference to the well-known derivation by Buckingham (Ref. 4).

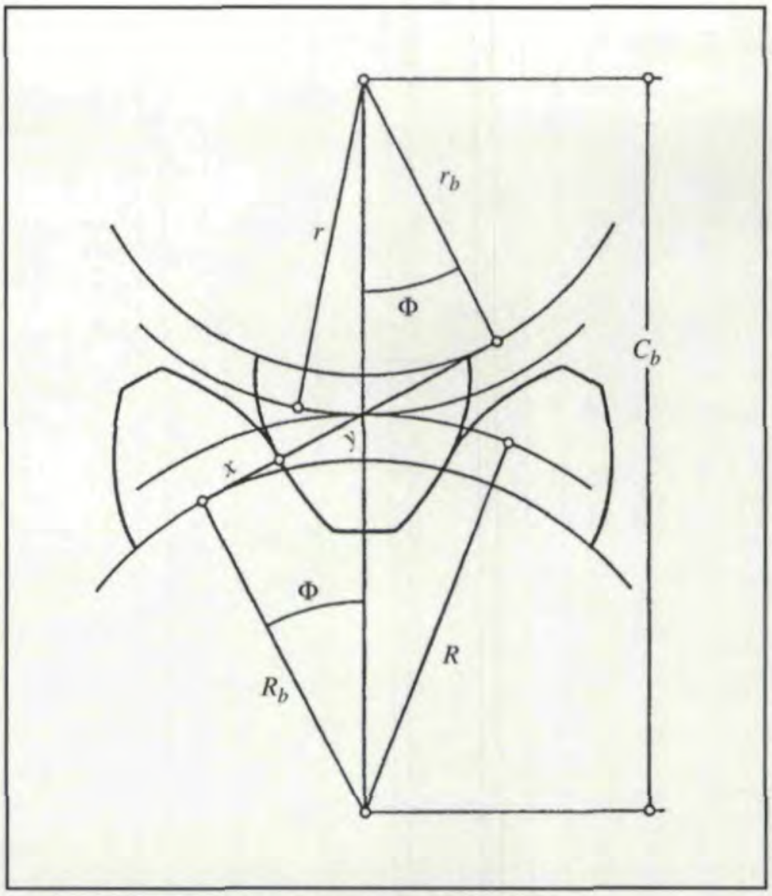

Fig. 1 — Reference circles and tooth profiles in contact.

From Fig. 2 and from the definition of the involute (the curve generated by a point on a string as it is being unwound from a circle), it is seen that $S_b/2$ can be wrapped onto either base circle. Consequently, with the gear fixed, the pinion is free to rotate $S_b/2r_b$ radians, both *CW* and *CCW*, or S_b/r_b total. Likewise, with the pinion fixed, the gear is free to rotate $S_b/2R_b$ radians, both CW and CCW, or S_b/R_b total.

It should be observed that space S_b could be eliminated by increasing the tooth thickness on either the gear (by setting $\Delta T_b = S_b$), or the pinion ($\Delta t_b = S_b$) or some combination of both ($\Delta T_b + \Delta t_b = S_b$). In general, with an allowance for backlash,

$$S_b = \Delta T_b + \Delta t_b + B_b, \quad (2)$$

where S_b is obtained from Eq. 1. And it should be observed that if the teeth were thinned, then both ΔT_b and Δt_b would be negative, which would make B_b greater than S_b.

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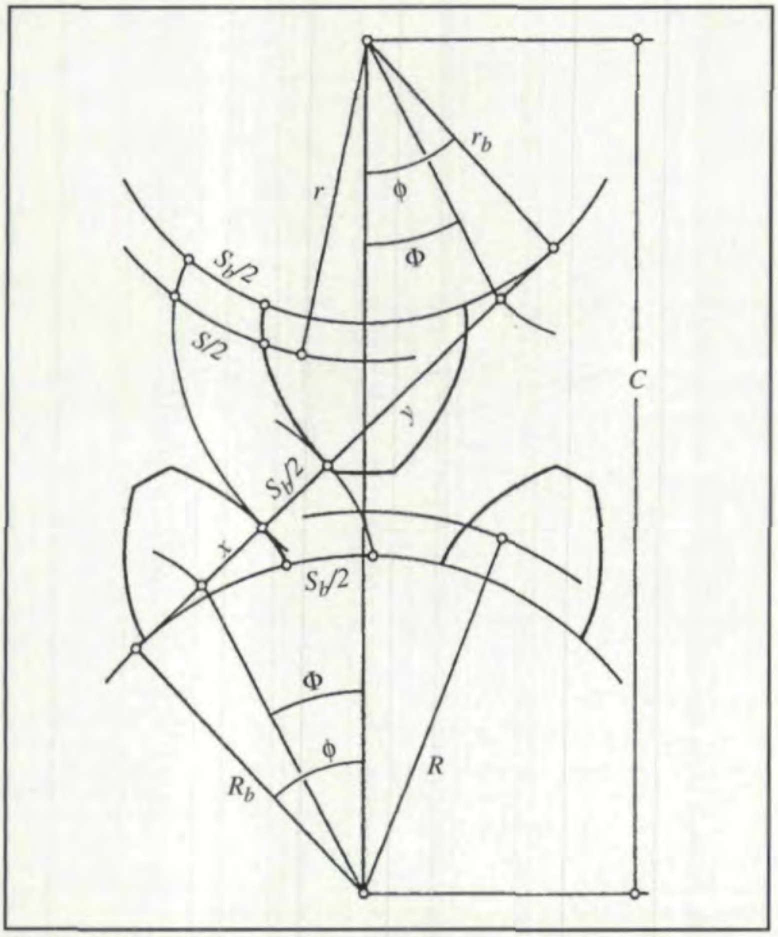

Fig. 2 — Effect of an increase in center distance.

Since tooth thickness on the base circle varies with tooth number, it is customary to work with tooth thickness on the reference circle, where for all tooth numbers, the tooth thickness is, in most cases, nearly equal to the basic tooth thickness ($p/2$). Further, it is convenient to work with the deviations from $p/2$, namely, with ΔT and Δt, mainly because they can be exchanged for backlash (as will be seen in Eq. 4). These deviations from $p/2$ are, of course, related to the offset of the generating rack; i.e., $\Delta t = 2\Delta C \tan\Phi$ where ΔC is the rack offset.

The relation of an arc on the reference circle to an arc on the base circle is shown in Fig. 2, where it is seen that arcs $S_b/2$ and $S/2$ subtend equal angles. Specifically, for the pinion,

$$\frac{S_b/2}{r_b} = \frac{S/2}{r}, \text{ or } S_b = \frac{S_b}{r}$$

where, from Fig. 1, the $r_b = r\cos\Phi$, so that $S_b = S\cos\Phi$. Similarly, for Δt_b and B_b in Eq. 2, $\Delta t_b = \Delta t\cos\Phi$ and $B_b = B\cos\Phi$. Likewise, for the gear,

$$\frac{S_b/2}{R_b} = \frac{S/2}{R}, \text{ or } S_b = S\,\frac{S_b}{R}$$

where, from Fig. 1, the $R_b = R\cos\Phi$, so that $S_b = S\cos\Phi$, $\Delta T_b = \Delta T\cos\Phi$ *and* $B_b = B\cos\Phi$. Thus, from Eqs. 1 and 2 it is seen that

$$S_b = S\cos\Phi = (\Delta T + \Delta t + B)\cos\Phi = 2(R_b + r_b)(\text{inv}\,\phi - \text{inv}\,\Phi),$$

or

$$B = 2\,\frac{R_b + r_b}{\cos\Phi}\,(\text{inv}\,\phi - \text{inv}\,\Phi) - (\Delta T + \Delta t),$$

where, from Figs. 1 and 2, the

$$R_b + r_b = C_b\cos\Phi = C\cos\phi,$$

so that

$$\cos\phi = \frac{C_b}{C}\cos\Phi, \qquad (3)$$

$$B = 2\,C_b(\text{inv}\,\phi - \text{inv}\,\Phi) - (\Delta T + \Delta t), \qquad (4)$$

and

$$B_b = B\cos\Phi. \qquad (5)$$

Unknown Errors

It is important to remember that ΔT and Δt (in Eq. 4) are subject to unknown errors in measured tooth thickness, because conventional measuring methods (span, pins, master gear) presume perfect teeth. For example, given a perfect $36T$-$12P$-20° gear the measured tooth thickness can be found from

$$M = \frac{10.8367}{12} + \Delta t\cos 20°,$$

where M is the span measurement across 4 teeth and 10.8367 is the span dimension for $P = 1$ and $\Delta t = 0$. Thus, for a span measurement of 10.8367/12, from

$$M = \frac{10.8367}{12} = \frac{10.8367}{12} + \Delta t\cos 20°,$$

the $\Delta t = 0$, and for two such gears meshed at a center distance of $C = C_b = (36 + 36)/(2 \times 12) = 3$ inches, the backlash along the line of action is zero; i.e., from Eqs. 3, 4, and 5, the $B_b = 0$. However, for the same span measurement on a gear that has been photographically reduced to $P = 12.05$, from

$$M = \frac{10.8367}{12} = \frac{10.8367}{12.05} + \Delta t \cos 20°,$$

the $\Delta t = 0.00399$, $C_b = 36/12.05$, and at $C = 3$, the $B_b = 0.0012$, not $B_b = 0$ as for $P = 12$. Similarly, for a gear that has been photographically enlarged to $P = 11.95$, the $\Delta t = -0.00402$, $C_b = 36/11.95$, and at $C = 3$ the $B_b = -0.0009$ (and interference).

For the aforesaid $36T$-$12P$-20° gear, the dimension over 1.92 pins is 39.0886 for $P = 1$ and $\Delta t = 0$. Thus, for a pin measurement of 39.0886/12 on a $P = 12.05$ gear, the $\Delta t = 0.00502$ (obtained by solving the pin equations for Δt), and for two such gears meshed at $C = 3$, the $B_b = -0.0008$ (an interference, versus a backlash of 0.0012 for span). Similarly, for the same pin measurement on a $P = 11.95$ gear, the $\Delta t = -0.00494$ and $B_b = 0.0008$ (versus an interference of 0.0009 for span). In short, for both $P = 12.05$ and 11.95, the backlash for pins was opposite in sign to the backlash for span.

It is of interest to note that the error in base pitch, relative to $P = 12$, was 0.0010 inches for both $P = 12.05$ and 11.95, and that an error of this magnitude is not uncommon in formed gearing (molded plastic, die cast, powder metal, stamped, cold-drawn). Further, when the combination of tolerances (for outside radius, tip round, bearing clearance, center distance) is large relative to whole depth, as in the case of fine-pitch formed gearing, it usually is necessary to design for minimum backlash, so as to avoid a contact ratio of less than unity (Ref. 5).

And it should be noted that no generalization can be drawn from these idealized examples, since the error in measured tooth thickness varies with the number of teeth spanned, the pin diameter and the tooth number.

When the two-flank roll test is not practical, the ΔT and Δt (in Eq. 4) should be adjusted to account for the allowable deviations in runout, tooth alignment, profile and spacing. An approximate adjustment can be deduced from similar designs, namely, from the discrepancy between calculated backlash and measured backlash — which also includes the aforementioned unknown error in measured tooth thickness. But if similar designs are not available, then it is necessary to estimate the individual adjustments (Ref. 6), and to assume a value for the unknown error in measured tooth thickness.

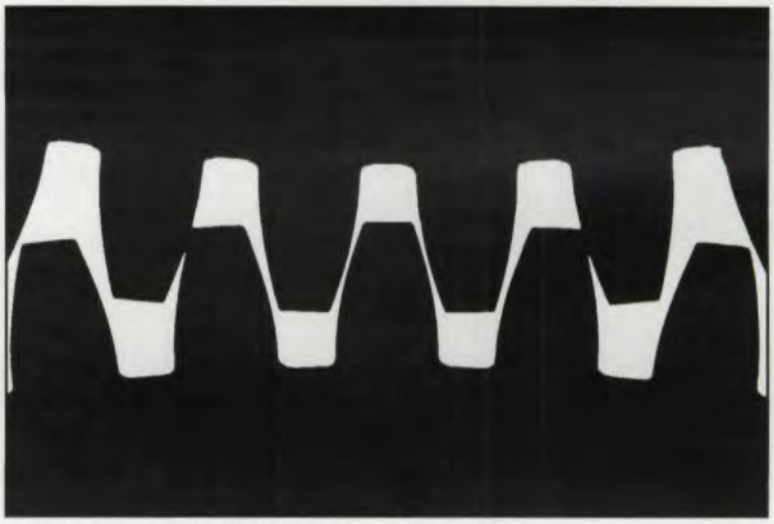

Fig. 3 — A mesh in which some teeth fail to make contact.

Characteristics of Master Gears

When a master gear is used to measure tooth thickness, the teeth will be thinner than the value indicated by the master gear, not thinner or thicker as when span or pins are used. To illustrate, consider two gears — one imperfect and one perfect — in mesh with a master gear at the same center distance. It is clear that the teeth on the imperfect gear will be thinner than the teeth on the perfect gear (Ref. 7). As in all functional gaging (gears, splines, screw threads, bores), size is sacrificed for defects in form and position. Consequently, since the teeth are thinner than that indicated by the master gear, the backlash will be somewhat greater than that indicated by Eq. 4.

Moreover, the magnitude of the error in measured tooth thickness will vary with the number of teeth on the master gear. This can be easily confirmed by meshing gears of various tooth numbers with a single gear of slightly different pitch and measuring both the tooth-to-tooth composite variation (also known as tooth-to-tooth composite tolerance, tooth-to-tooth composite error and originally as *kickout* [Refs. 8–9]) and center distance.

For example, when 192*T*-127*P*-20° and 180*T*-120*P*-20° master gears are rolled together, the kickout is only about 0.0002 inches, despite the fact that the difference in base pitch is 0.0014 inches. But when 120*P*-20° relatively high-grade gears (error in base pitch ≤10% of 0.0014 inches) of various tooth numbers are rolled with the 192*T*-127*P* master gear, the kickout increases as the tooth number decreases, to about 0.0019 inches for a 12*T*-120*P* pinion. This effect, which is well-known (Ref. 10), is the result of partial tooth contact (see Fig. 3), the degree of which varies with contact ratio. Conversely, the magnitude of the center distance error — namely, the amount by which the maximum center distance exceeds the sum of the reference radii — decreases as the tooth number decreases, from about 0.006 inches for the 180*T*-120*P* master gear to about 0.003 inches for a 12*T*-120*P* pinion.

A single master gear, therefore, should not be used to check any and all tooth numbers. The master gear method has, in Mark Twain's words, "a certain degree of merit . . ., but like chastity — it can be carried too far!"

Master Gear Design

The ideal master gear would have the same tooth number, same depth of contact and same pitch circle as the gear that mates with the gear to be inspected; i.e., with the work gear. And when the face width of the mating gear is less than the face width of the work gear, the ideal master gear would have the same face width as the active face width of the mating gear.

Because depth of contact depends upon the outside radius of the mating gear, the tip round or chamfer on the mating gear, and the center distance between mating gear and work gear, all three of which vary, the master gear should be designed to match the maximum depth of contact. Specifically, the outside radius of the master gear should be the same as the maximum outside form radius of the mating gear (maximum outside radius less the radial effect of minimum tip round or chamfer). And the tooth thickness of the master gear should be such that the maximum roll test center distance between master gear and work gear is the same as the minimum center distance between the mating gear and the work gear. That is, for $B = 0$ in Eq. 4, the

$$\Delta T = 2\ C_b(\text{inv}\ \phi - \text{inv}\ \Phi) - \Delta t, \qquad (6)$$

where ΔT is for the master gear, inv ϕ is obtained from Eq. 3 (wherein C is the minimum center distance between mating gear and work gear), and Δt is for maximum tooth thickness of the work gear.

The tooth thickness of most work gears will be less than maximum, so in most roll tests the depth of contact between master gear and work gear will be slightly greater than the maximum depth of contact between mating gear and work gear.

In most cases an acceptable approximation to the ideal tooth number can be obtained by simulating the roll test on a computer. In particular, the ideal master gear is meshed with the work gear, and then a kickout, equal to the kickout tolerance, is induced by altering the diametral pitch of the work gear while maintaining the center distance by reducing the tooth thickness of the work gear. Next, the tooth number is altered, but only to the point where there is no significant change in kickout or tooth thickness.

It is of interest to note that the same computer simulation can be used to establish realistic tolerances for master gears. The procedure is the same, except that, instead of tooth number, the diametral pitch of the master gear is altered. The resulting change in diametral pitch is then converted into a change in base pitch, which in turn is converted into tolerances for circular pitch and profile.

And it is worth noting that the same computer simulation can be used to determine the tight-mesh center distance for a gear pair with different thermal expansions. At elevated temperatures, for instance, the diametral pitch of a metal gear will be different from that of a mating plastic gear, which means that the equations for tight-mesh center distance are not valid.

Alternative to Master Gears

From the standpoint of the designer, the master gear method leaves much to be desired. Aside from the need for a variety of tooth numbers and face widths (usually not available), there is the aforementioned unknown error in measured tooth thickness. Also, there is the uncertainty in kickout between mating gears — as is evident from the following illustration. Consider a 20*P* master gear, two 19.95*P* gears, and two 20.05*P* gears, all perfect and with the same tooth number. The kickout between the 20*P* and the 19.95*P* will be about the same as between the 20*P* and the 20.05*P*. But there will be no kickout between the 19.95*P* gears, nor between the 20.05*P* gears. And the kickout between the 19.95*P* and 20.05*P* will approach the sum of their kickouts against the 20*P*. In short, the designer has no way of knowing how the kickouts will combine.

The unknown error in measured tooth thickness and the uncertainty in kickout can be avoided when the tooth thickness of one or both members of a gear pair is readily adjustable. For instance, when one member is a hobbed pinion and the other a formed gear (molded plastic, die cast, powder metal, stamped), the hobbing machine operator can simulate the assembly process by drawing parts at random from the gear lot and using them to check the pinions. The pinion specification would read: ROLL TEST CENTER DISTANCE WITH ANY MATING GEAR X.XXXX/X.XXXX, and KICKOUT WITH ANY MATING GEAR .XXXX. Also, because all teeth are somewhat unsymmetrical and misaligned, the pinion drawing would identify the mating flanks and active face width.

Roll Test Center Distance

Equations for roll test center distance are derived from Eqs. 3 and 4. That is, for $B = 0$ in Eq. 4, the

$$\text{inv}\ \phi = \text{inv}\ \Phi + \Delta T + \Delta t\ ,\ \frac{\Delta T + \Delta t\ ,}{2\ C_b} \qquad (7)$$

and from Eq. 3,

$$C = C_b\ \frac{\cos\ \Phi}{\cos\ \phi}. \qquad (8)$$

Because the total composite variation (also known as total composite tolerance, total composite error, and to a lesser extent as *rollout*) must fall within the limits of roll test center distance, and since Eqs. 7 and 8 are valid only for perfect teeth, it follows that the corresponding limits of ΔT and Δt are the limits of perfect teeth. In other words, the rollout must fall within the size tolerance.

It is pertinent to note that the German equivalents of rollout and kickout are also single words, namely, *Wälzfehler* (roll error) and *Wälzsprung* (roll jump), respectively.

From Eqs. 7 and 8 it is seen that inv ϕ must be converted into cos ϕ. This conversion can be easily carried out on programmable pocket calculators that sell for as little as $30 (Ref.11). Even so, the test radius method, which was devised to bypass Eqs. 7 and 8, is still in widespread use.

Test Radius Method

Originally, test radii were obtained by simply adding $\Delta t/(2\tan\Phi)$ and $\Delta T/(2\tan\Phi)$ to the reference radii of the work gear and master gear, respectively. But, since $(\Delta t + \Delta T)/(2\tan\Phi)$ is just the first term of an infinite series (Ref. 12) that method was subject to a significant error.

Nowadays, the test radius of the work gear is defined as the radial distance from the center of the work gear to the reference circle of the master gear, as obtained by setting $\Delta T = 0$ (in Eq. 7) and subtracting the reference radius of the master gear from the center distance (Eq. 8).

It is of course necessary to set $\Delta T = 0$ in Eq. 7, since the test radius of the master gear is sill obtained in the original way, i.e., by adding $\Delta T/(2\tan\Phi)$ to the reference radius. As a result, the designer is forced to specify the master gear tooth number (or range of tooth numbers), since the test radius of the work gear varies with number of teeth on the master gear. (The range usually narrows to a single number for large values of Δt.) In addition, the designer is forced to specify a tolerance on master gear tooth thickness, since the error in work gear tooth thickness varies with master gear tooth thickness.

For example, given an 8*T*-20*P*-20° pinion whose tooth thickness is 0.02830 inches greater than the basic tooth thickness (to eliminate undercut), and a 40*T* master gear.

Accordingly, in Eq. 7 the $\Delta t = 0.02830$, $\Delta T = 0$, $n = 8$, $N = 40$, $P = 20$, $C_b = (n + N)/2P = 1.2$, and $\Phi = 20°$. Then, from Eq. 8 the $C = 1.2353$, so the pinion test radius is $1.2353 - N/2P = 0.2353$. (For a 30T master gear, the pinion test radius would be 0.2346, not 0.2353.) Next, given that the master gear tooth thickness is 0.00080 inches greater than the basic tooth thickness ($\Delta T = 0.00080$), the master gear test radius is $N/2P + \Delta T/(2 \tan \Phi) = 1.0011$. Consequently, the roll tester is set to $0.2353 + 1.0011 = 1.2364$ inches.

But for this center distance the pinion tooth enlargement will not be $\Delta t = 0.02830$. Instead, for $C = 1.2364$ in Eq. 3 and $\Delta T = 0.00080$ in Eq. 6, the $\Delta t = 0.02845$, which is in error by 0.00015 inches. Because this error is a biased error (not a random error), "good gaging practice" dictates that it not be greater than about 5% of the tooth thickness tolerance. In other words, in this particular case the test radius method would not be valid if the tooth thickness tolerance were less than about 0.0030 inches.

Because the constraints on master gear tooth thickness and tooth number can be rather severe, the test radius method tends to raise the cost of manufacture. Another specification that tends to raise the cost of manufacture is rollout — a holdover from the days when rollout did not fall within the limits of test radius (Ref. 13). When not specified, rollout is free to increase as size variation decreases, and whether the backlash is caused by rollout or size variation is usually of no consequence.

A rollout specification is needed only in special cases, such as in high-speed gearing and in various type of anti-backlash gearing. A rollout specification is not an effective check for index error, since index error can be not only large for a small rollout (Refs. 14–15), but also small for a large rollout (as when the rollout opposes the accumulated spacing error).

Backlash Calculations

Assigning numbers to the center distance (C in Eq. 3) and to the sum of tooth thicknesses ($\Delta T + \Delta t$ in Eq. 4) is not an easy matter. The task would be fairly simple if the axes of rotation were parallel, but housing bores are not coaxial, bearing clearances are not equal, journals are not coaxial, runouts of bearing races are neither equal nor in phase, and mountings for gears on motors, encoders, etc. are tilted, as are the fixed studs on which gears are mounted. Furthermore, tooth reactions cause unequal radial shifts within the bearing clearances.

The center distance and the increase in the sum of tooth thicknesses are obtained by projecting both axes of rotation onto the axial and pitch planes, respectively. (The axial plane contains the nominal positions of both housing axes. The pitch plane is perpendicular to the axial plane and parallel to the housing axes.) In the axial plane the distance between the skewed axes at the point where the gears make contact is the center distance. In the pitch plane the product of the angle between the axes (in radians) and the smaller of the two face widths is the effective increase in the sum of tooth thicknesses.

If manufacturing distributions for all pertinent dimensions are available, then the backlash distribution can be obtained by simulating the assembly process on a computer (Refs. 16–17). Parts are drawn at random, assembled on the computer, and then the backlash (B in Eq. 4) is computed for each and every assembly. This method does not involve mathematical statistics. Moreover, it can handle any type of frequency distribution (bimodal, rectangular, skewed, decentered, sorted, small quantities).

If manufacturing distributions are not available (as when it is not cost-effective to ascertain and control both the shape and centrality of each distribution), then only the maximum backlash and minimum backlash can be obtained. Specifically, maximum backlash is that for maximum center distance and minimum tooth thicknesses. Minimum backlash is that for minimum center distance and maximum tooth thicknesses, plus the effective increase in the sum of tooth thicknesses due to misalignment.

The minimum backlash can be slightly negative (a potential, but improbable interference), since the range of the backlash distribution will be less than the difference between maximum backlash and minimum backlash.

The magnitude of the potential interference is usually dependent upon the manufacturing process. In particular, parts made with hard tooling (molded plastic, die cast, powder metal, stamped, cold-drawn) tend to have decentered distributions, so the potential interference cannot be as great as for parts whose dimensions are readily adjustable during the course of manufacture.

In most designs the potential interference can range from 15 to 25 percent of the difference between the maximum backlash and minimum backlash, depending upon the shape and centrality of the various distributions.

Errors in Roll Testing

The two-flank roll test has always been plagued by significant errors in measurement. For example, about 30 years ago Michalec and Karsch conducted a correlation study (Ref. 18), wherein a number of companies inspected a large assortment of precision gears for test radius, rollout and kickout. The discrepancies between the participants were indeed significant. For instance, the average of all the greatest discrepancies for test radius (each gear had a greatest discrepancy, namely, the difference between the highest and lowest reported measurements) was an incredible 0.0016 inches!

If a similar correlation study were to be conducted today, the discrepancies probably would be similar, since there have been no marked improvements in inspection practice and test equipment. Techniques for preventing significant errors in measurement have been known for some time (Refs. 19–20), but they are seldom implemented.

To insure against significant errors in measurement, the designer can, in self-defense, invoke a process specification (as is done for heat treating, application of dry-film lubricants, etc.) that spells out the roll test requisites, namely, mounting method, axis alignment, permissible hysteresis and nonlinearity of the movement, master gear dimensions and tolerances, and how to determine the checking load, checking speed, load correction and temperature correction. Without an effective process specification, the designer can do little more than cross his/her fingers and hope for the best. ■

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GEAR FUNDAMENTALS

Computerized Hob Inspection & Applications of Inspection Results — Part I

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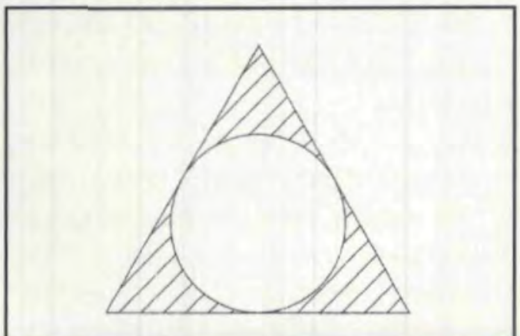


Fig. 1

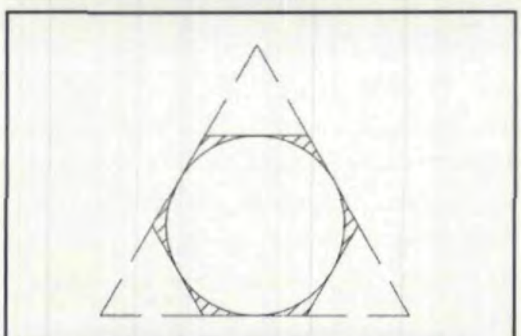


Fig. 2

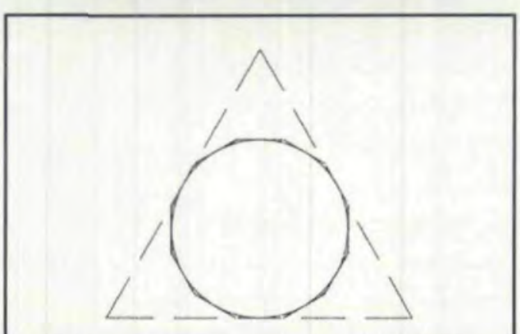


Fig. 3

Introduction

Can a gear profile generated by the hobbing method be an ideal involute? In strictly theoretical terms — no, but in practicality — yes. A gear profile generated by the hobbing method is an approximation of the involute curve. Let's review a classic example of an approximation.

Do regular polygons and circles have anything in common? Yes. One can approximate a circle simply by increasing the number of sides of a polygon.

Let's assume that one has scissors and can cut only a straight line. Let's cut the simplest polygon, a triangle, from a piece of paper. It hardly resembles a circle (Fig. 1). The shaded area shows the variation between a triangle and a circumscribed circle. But if the number of the sides of a polygon is doubled (Fig. 2), the variation between a polygon and a circle is reduced dramatically. Yet, if the number of sides is doubled once again, the variation can hardly be seen (Fig. 3).

By increasing the number of sides of a polygon further, it is possible to get so close to a circle that the variation becomes negligible — the difference cannot be seen or even measured.

The process of generating a gear on a hobbing machine is based on a similar idea of approximation. Hobbing is a process that generates a number of connected lines which approximate an involute curve.

Involute Generation on a Hobbing Machine

A hobbing machine cannot cut curves, but it can cut lots of straight lines in a certain pattern. Therefore, the idea of approximation is utilized in order to generate an involute. Every cutting edge of a hob cuts a straight line. The number of straight lines (enveloping cuts) should be large enough so that the difference between the involute and the combination of straight lines becomes negligible.

Figs. 5 and 6 show gear profile generation as seen by an observer who rotates with the gear.

Fig. 5 shows an approximation of an involute generated by only three cutting edges of a hob. The shaded area illustrates the variation between the involute and the approximating cuts.

If the number of cutting edges is increased, as in Fig. 6, the variation becomes less apparent. The involute variation generated by an ideal hob can be calculated as follows:

Profile Variation = $[\pi^2 \cdot Z_o \cdot M_n \cdot \sin(NPA)]/(4 \cdot Z2 \cdot i^2)$

Z_o — Number of hob starts
M_n — Normal module
NPA — Normal pressure angle
$Z2$ — Number of gear teeth
i — Number of hob gashes

As one can see from the formula, an exponential reduction in variation can be obtained by increasing the number of gashes. A gear generated by means of an ideal hob, an ideal machine and an ideal fixture will have a profile curve that is an approximation of an involute, in the same way an equilateral polygon approximates a circle. The whole topology of a gear tooth consists of numerous cuts in lead and involute direction (Fig. 7).

A center of every single generating cut lies on the line of action (Fig. 8). The dashed lines depict hob cutting edges from the point of view of an observer rotated the gear.

After a hob with a sufficient number of cutting edges is selected, the hob should be able to generate an ideal involute or at least an involute with a predictable variation. Why does it sometimes fail? Well, because we live in an imperfect world, especially when it comes to a hobbing machine, a work-holding fixture, a blank or a hob, all of which effect

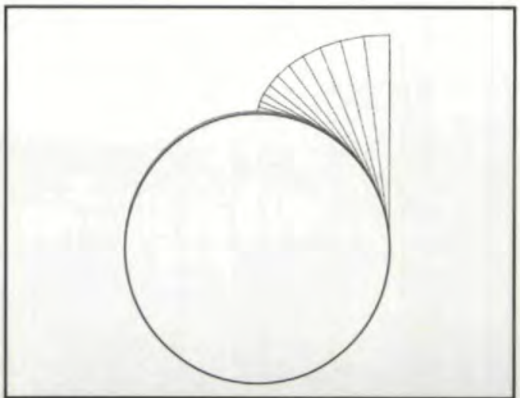


Fig. 4

Fig. 5

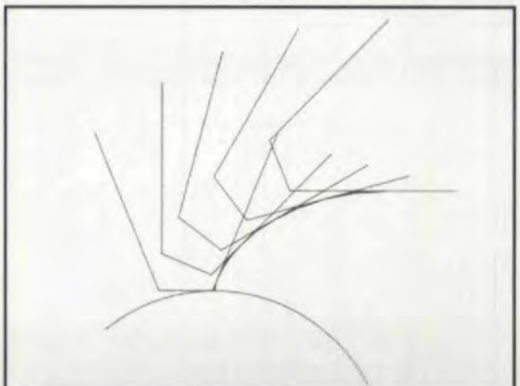
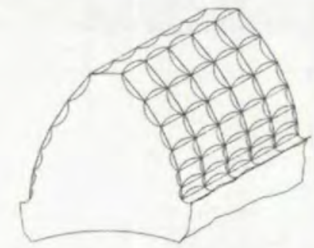


Fig. 6



$$\text{Profile variation} = \frac{\Pi^2 * Z_0 * M_n * \text{Sin(NPA)}}{4 * Z2 * i^2}$$

Z_0 – Number of hobs starts
M_n – Normal module
NPA – Normal pressure angle
Z2 – Number of gear teeth
i – Number on hob gashes

Fig. 7

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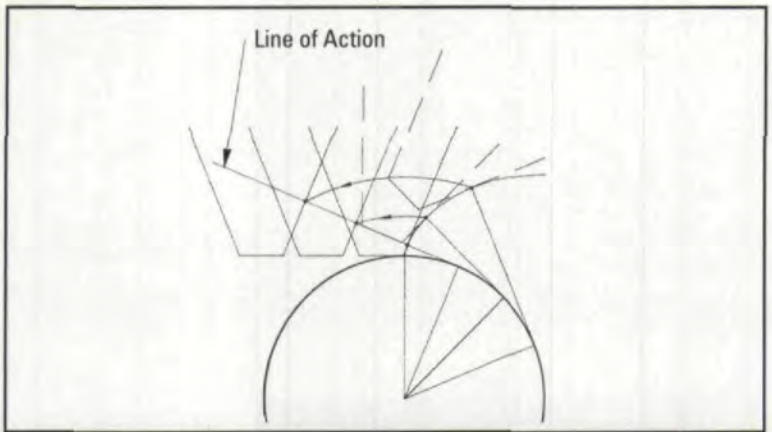


Fig. 8

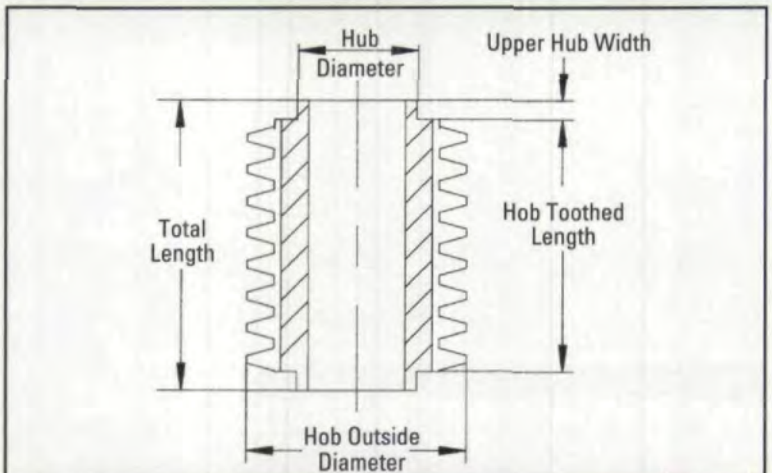


Fig. 9

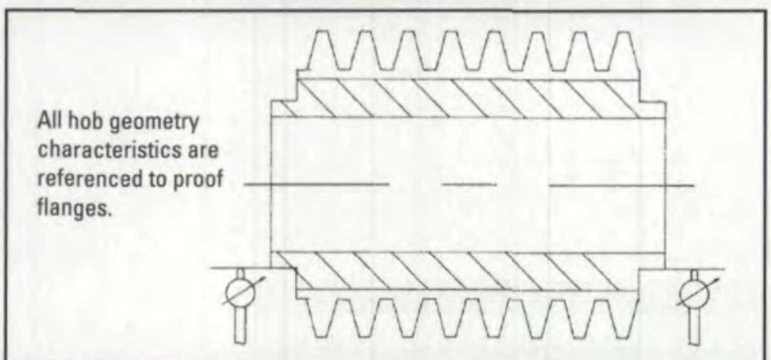


Fig. 10

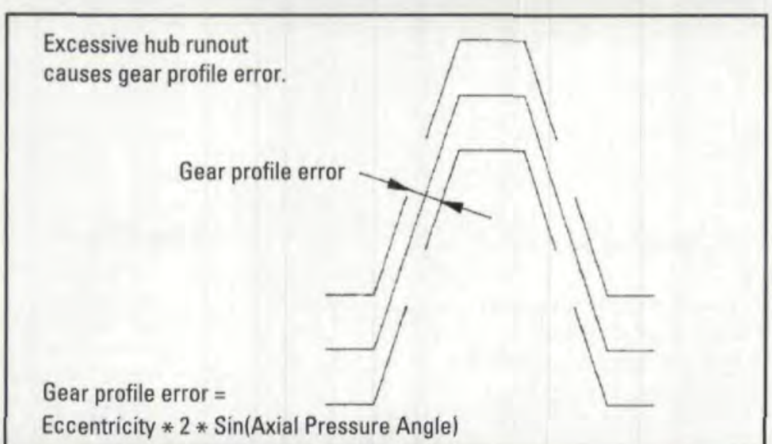


Fig. 11

the hobbing process.

Our discussion, however, will be limited to reviewing the effects of hob errors on the accuracy of involute generation.

Hob Geometrical Characteristics

The ultimate goal of hob inspection is to make sure that during the hobbing process, the cutting edges of a hob have a minimum deviation from their theoretical positions.

There are several hob geometrical characteristics. Some of them, like the line of action, show the direct variation of cutting edges from their theoretical positions at the points where they generate gear involutes. Most characteristics, however, can only show the displacement of cutting edges indirectly.

Commonly accepted characteristics that a hob manufacturer or a hob user might check include the following: Radial runout of proof flanges, face runout, rake, flute index, flute lead, lead and thread-to-thread variation, outside diameter, pressure angle, line of action, radial and axial relief and tooth thickness.

Radial Runout of Proof Flanges

Most hobs have ground proof diameters or hubs on both sides (Fig. 9). These diameters are used by operators to indicate a hob when mounting it on the machine (Fig. 10). All hob geometrical characteristics are referenced to proof diameters. Usually proof diameters are checked first.

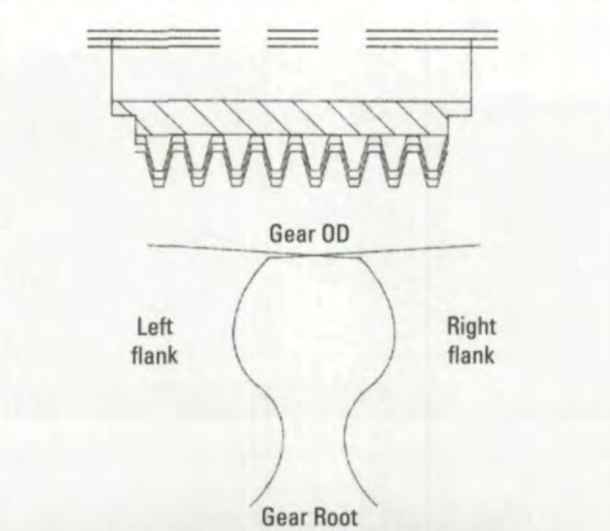
Excessive hub runout causes a gear profile error that can be approximately calculated as shown in Fig. 11.

$$\textit{Profile error} = 2 \bullet \textit{eccentricity} \bullet \sin(\textit{axial pressure angle})$$

Fig. 12 shows the effect of hob radial runout on gear involute. Fig. 13 shows the least squares method for determination of a circle's center and out-of-roundness. This method allows one to determine concentricity and out-of-round amount very precisely.

A CNC inspection machine will automatically check the hub's runout at a specified position (Fig. 14).

Evaluation should include the determination of total runout, out-of-round and concentricity errors (Fig. 15). The results of inspection and evaluation can also be presented in circular form as shown in Fig. 16. This chart shows a round surface that is magnified 2000 times. The distance to the center of the best fit circle has the



Effect of hob radial runout on gear involute

Fig. 12

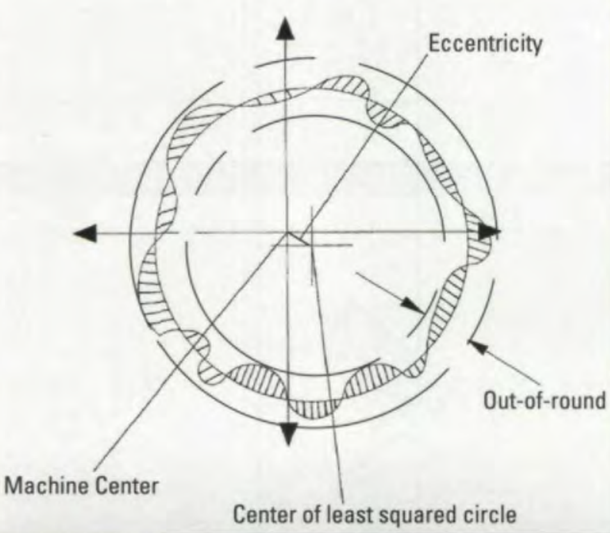


Fig. 13

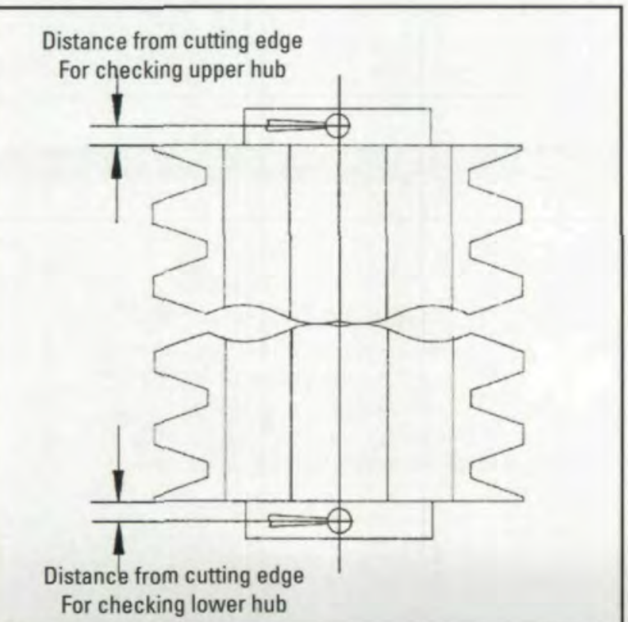
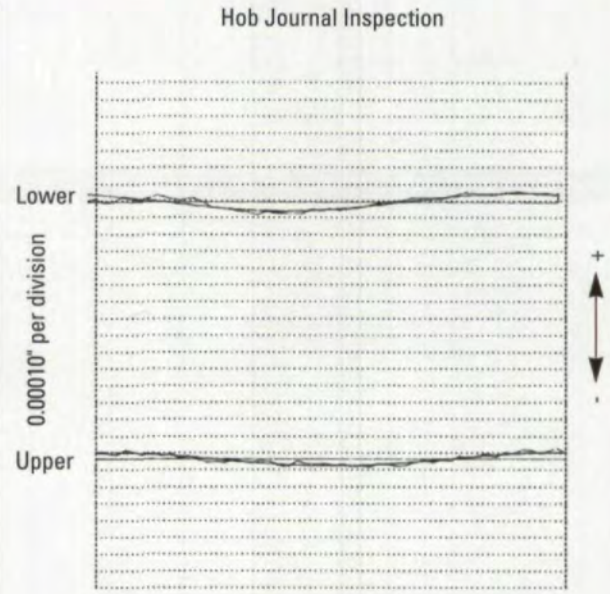


Fig. 14

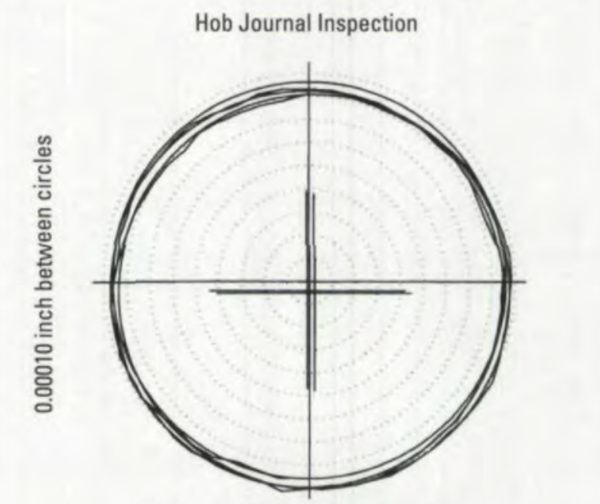
Hob ID	4191 Left Contact	N.D.P.	7.236467	Left NPA	20.000000
Serial	001	Axial Lead	0.8660000	Lead Hand	Right
Operator	Ed	Whole Depth	0.23622	Gash Hand	Left
Probe	0.07847L	Number Gashes	10	Required Quality	AGMA: B
Inspected 04/15/93	09:20:48	Number Threads	2	c:\Roto Hob\HB006.HOB\MS009.MES	
Metric/Inch	Inch	Right NPA	20.000000	Magnif 2000.00	Scale 0.40



Journal	Toler.	Upper	Lower
Total Runout	0.00070	0.00011:A	0.00014:A
Out of Round		0.00004	0.00009
Eccentricity		0.00004	0.00005
Average Diameter		1.71883	1.70886

Fig. 15

Hob ID	4191 Left Contact	N.D.P.	7.236467	Left NPA	20.000000
Serial	001	Axial Lead	0.8660000	Lead Hand	Right
Operator	Ed	Whole Depth	0.23622	Gash Hand	Left
Probe	0.07874L	Number Gashes	10	Required Quality	AGMA: B
Inspected 04/15/93	09:20:48	Number Threads	2	c:\Roto Hob\HB006.HOB\MS009.MES	
Metric/Inch	Inch	Right NPA	20.000000	Magnif 2000.00	Scale 2.00



Journal	Toler.	Upper	Lower
Total Runout	0.00070	0.00011:A	0.00014:A
Out of Round		0.00004	0.00009
Eccentricity		0.00004	0.00005
Average Diameter		1.71883	1.70886

Fig. 16

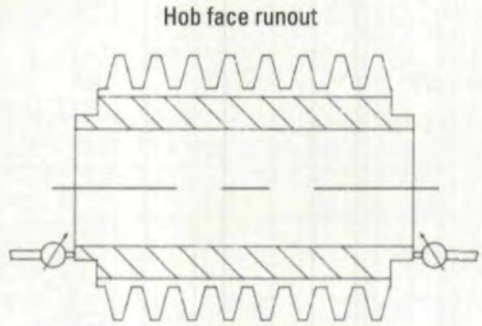
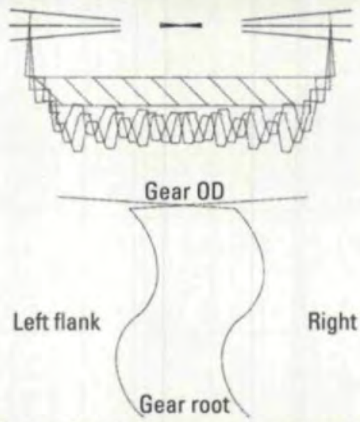


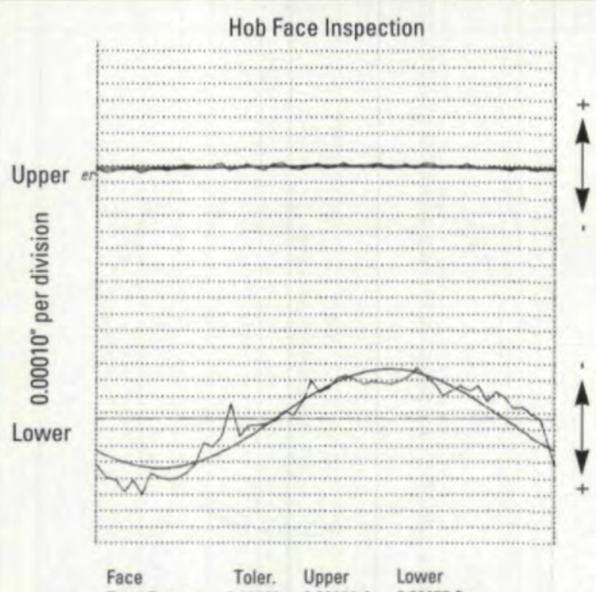
Fig. 17



Effect of hob face runout on gear involute

Fig. 18

Hob ID 4191 Left Contact		N.D.P.	7.236467	Left NPA	20.000000
Serial	001	Axial Lead	0.8660000	Lead Hand	Right
Operator	Ed	Whole Depth	0.23622	Gash Hand	Left
Probe	0.07874L	Number Gashes	10	Required Quality	AGMA: B
Inspected 04/15/93	09:20:48	Number Threads	2	c:\Roto Hob\HB006.HOB\MS009.MES	
Metric/Inch	Inch	Right NPA	20.000000	Magnif 2000.00	Scale 0.40



Face	Toler.	Upper	Lower
Total Runout	0.00030	0.00006:A	0.00075:?
Out of Flat		0.00005	0.00043
Eccentricity		0.00001	0.00030
Vertical Position		5.44237	2.42440

Fig. 19

Fig. 20

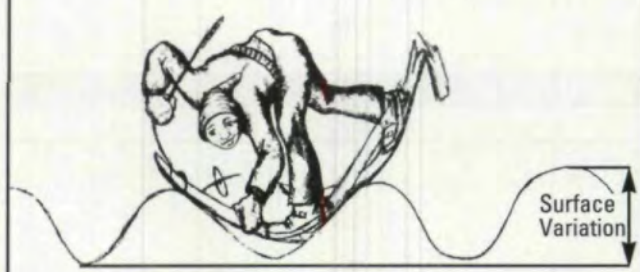


Fig. 21

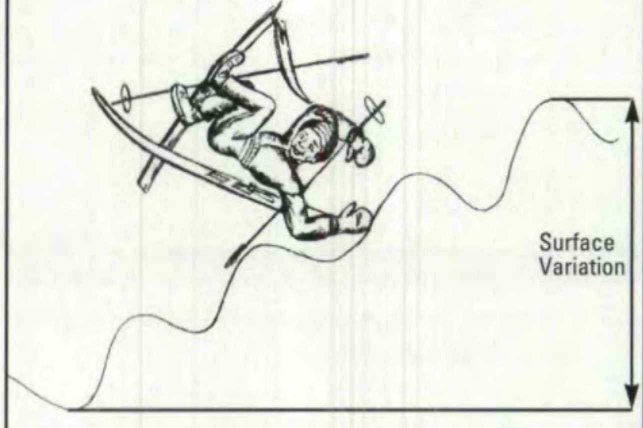


Fig. 22

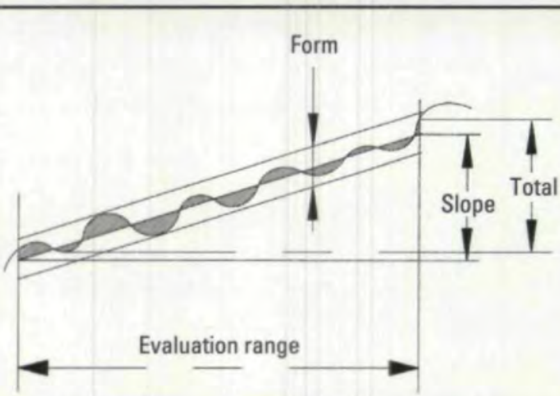


Fig. 23

same magnification; thus, eccentricity can be scaled. All the numerical evaluations are also displayed on the chart. The results of hub inspection on both sides are superimposed so the runout inaccuracies can be compared visually.

The evaluation program may have a built-in AGMA, DIN, and ISO hob tolerance system. If the operator specifies the required quality class, the program should automatically compute the required tolerance. For characteristics which have quality classification, the actual quality may also be automatically determined and displayed.

Face Runout

Hob faces are frequently utilized for clamping during mounting a hob on either a hobbing machine or a hob sharpening machine. The hob faces have to be trued (Fig. 17). Excessive face runout can result in involute variation (Fig. 18). Inspection and evaluation of face runout include out-of-flat and eccentricity (Fig. 19).

Rake

Some simplification of surface variation may be useful for process analysis and problem solving.

General Surface Variation Components. The variation of any surface from its ideal condition can be simplified as a variation of a mountain terrain in relation to a flat surface. One could ski on a mountain with a steady and even drop (Fig. 20), or on a horizontally undulating terrain (Fig. 21). But frequently mountain terrain is a combination of both (Fig. 22).

The concept of breaking down the total surface variation into several components is widely used in many applications, including hobs and gears. Fig. 23 illustrates the least squares method for the determination of form and slope error components. Frequently, slope and form errors are useful even if not specified by DIN/AGMA/ISO standards. The breakdown of the total value into slope and form components helps to determine the sources of errors and better identify any needed process adjustments.

Hob Rake Inspection. Hob rake is a line resulting from the intersection of a tooth face with a plane that is normal to the hob axis. If this line crosses the hob center, it is called a zero rake.

Rake offset is the amount by which the design rake line is distant from the plane of a hob axis (Fig. 24). Hob rake offset is zero if

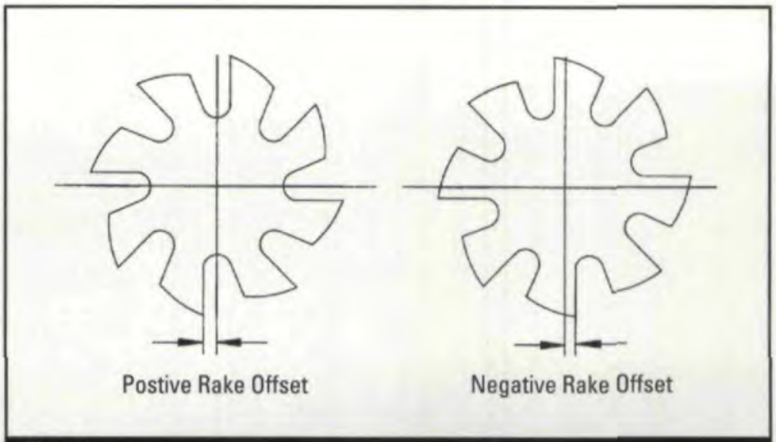


Fig. 24

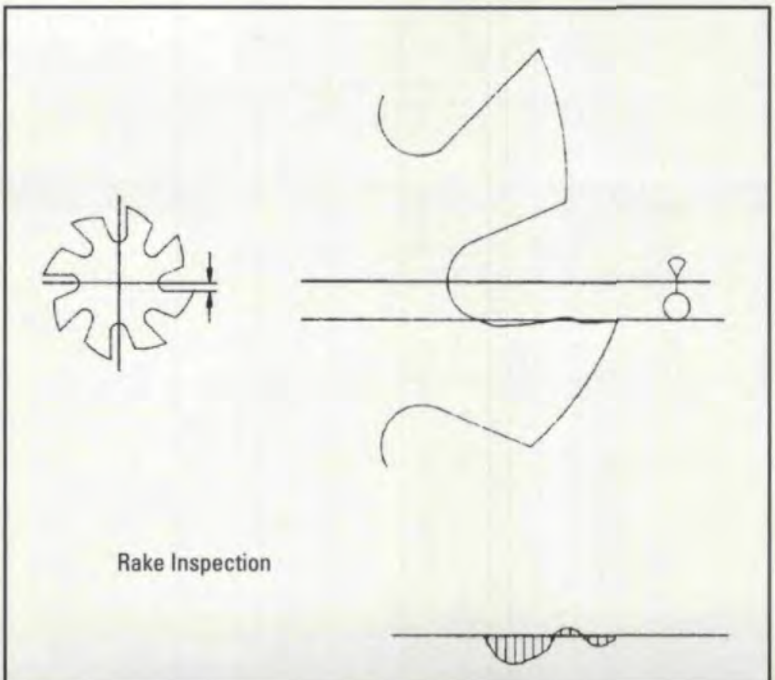


Fig. 25

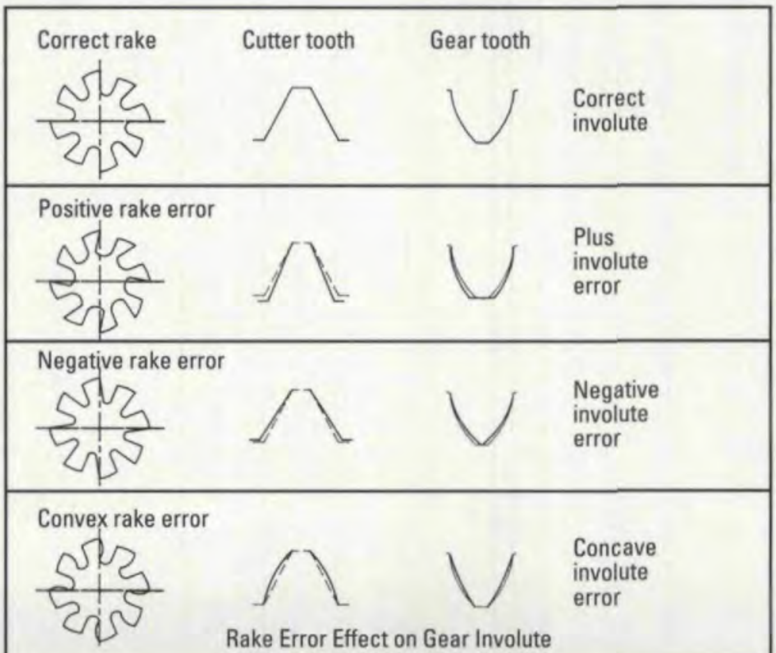


Fig. 26

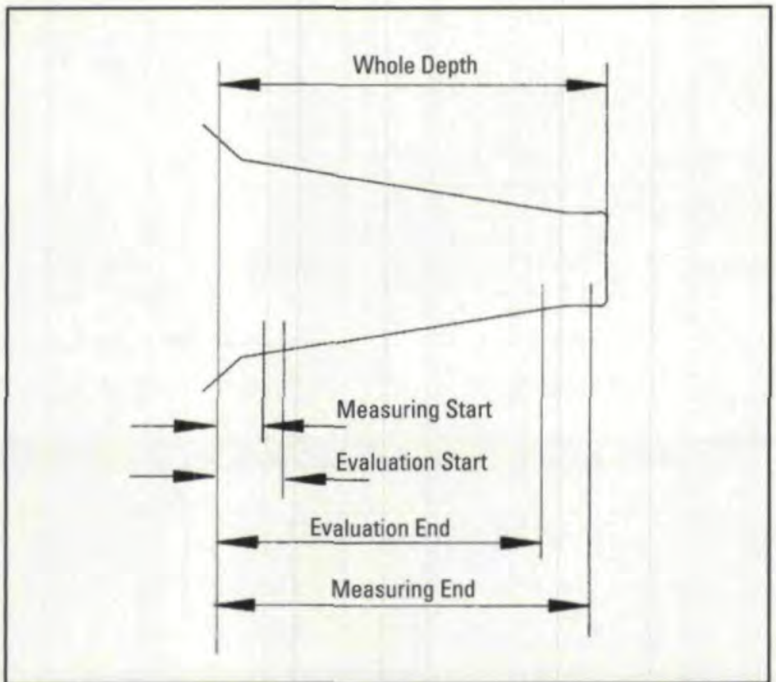
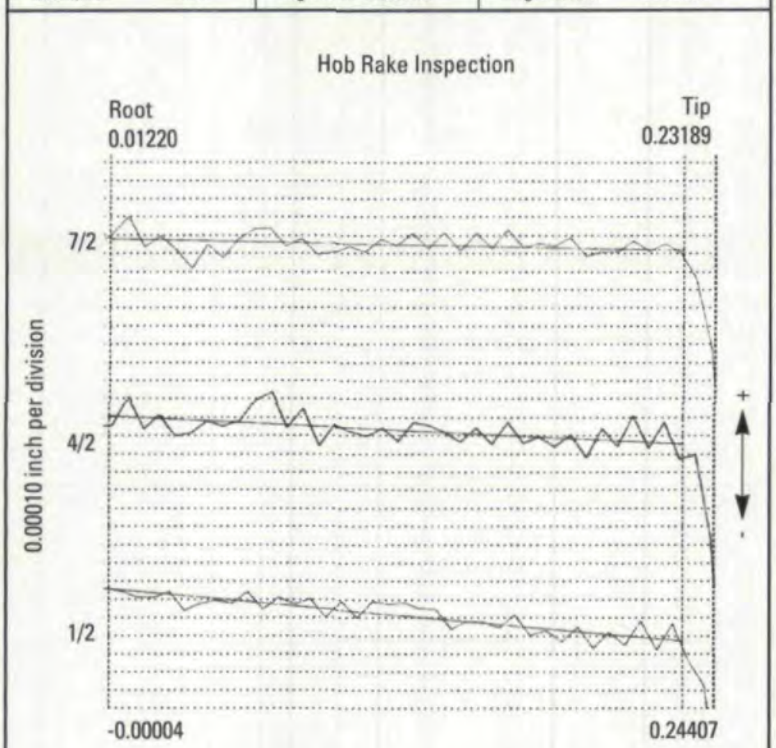


Fig. 27

Hob ID 4191 Left Contact		N.D.P.	7.236467	Left NPA	20.000000
Serial	001	Axial Lead	0.8660000	Lead Hand	Right
Operator	Ed	Whole Depth	0.23622	Gash Hand	Left
Probe	0.07874L	Number Gashes	10	Required Quality	AGMA: B
Inspected 04/15/93	09:20:48	Number Threads	2	c:\Roto Hob\HB006.HOB\MS009.MES	
Metric/Inch	Inch	Right NPA	20.000000	Magnif 2000.00	Scale 25.0



Gash/Tooth	Toler.	1/2	4/2	7/2
Total	0.00070	0.00038:A	0.00037:A	0.00034:A
Slope		−0.00028	−0.00016	−0.00007
Form		0.00019	0.00028	0.00028

Fig. 28

the rake line crosses the hob center. The rake offset is negative if the rake line makes an obtuse angle with the tooth outside diameter, and positive if the rake line causes an acute angle with the hob outside diameter. Most hobs have 0° rake, although there are some hobs with positive or negative rake. Rake is measured on the tooth face as shown in Fig. 25. The task of inspection is to find out rake deviation from the design geometry.

Rake inaccuracy affects the gear profile. Fig. 26 shows samples of typical rake inaccuracy and relevant gear profile errors. The inspection results should identify inspection distance in reference to the tip of the tooth (Fig. 27). A sample of rake inspection on several different teeth is shown in Fig. 28. In addition to the measuring and evaluation ranges, the chart should also identify tooth and raw number, magnification and scale, required standard, quality, and tolerance, if any. Inspection of helical flute hobs requires probe contact point adjustments, since flute angle changes as the probe moves from the root to the tip of the tooth. Evaluation results include the following characteristics for every measured rake: Total error, slope error, form error, and actual quality per AGMA, DIN or ISO standard for every tooth. If the system has off-line capabilities, the various evaluations could be performed without having to recheck a hob. ■

Editor's Note: *The second half of this article, will appear in our next issue.*

Acknowledgements**:** *The author wishes to thank Ed Driscoll for his advice, support, and creativity that inspired and helped write many parts of this paper; John Lange, for co-authoring a paper on a similar subject presented at an AGMA symposium in 1989; Lauren Brombert for her meticulous and creative editing help; Rachel Haisman, for creating the illustrations that helped simplify the description of surface variation characteristics and Richard Considine for computerizing them, and Esther Munsey for her constructive editorial help. Gear inspection charts are courtesy of Roto-Technology, Inc., Dayton, OH. This article was first presented at the AGMA Gear Manufacturing Symposium, held in Detroit, MI, October, 1993.*

References:

1. AGMA Standards
2. ANSI B94.7, 1980
3. DIN 3968-1069, 3000=1962, 3960-1980
4. VDI/VDE 2606
5. Kotlyar, Yefim and John Lange, "CNC Inspection and Evaluation of Gear Cutting Tools," AGMA, 1989.

PEOPLE ON THE MOVE

ATA Gears, Inc.

Cleveland, OH. Mr. **Rauno Eliander** has been named president and Mr. **Thad N. Schott** has been named vice-president of North American operations of ATA Gears, Inc., Cleveland, OH. ATA Gears, Inc., is the U.S. subsidiary of ATA Osakeyhtö of Finland, a manufacturer of spiral bevel gears.

T. S. Alloys

Houston, TX. Mr. **Michael Likos** has been promoted to co-general manager of T. S. Alloys of Houston, TX, where he will report to **Troy Marshall**, vice president of Avesta Sheffield, Inc.

T. S. Alloys, a stainless steel and aluminum service center, is wholly owned by Avesta Sheffield, whose world headquarters are in Schaumburg, IL. T. S. Alloys' regional marketing area includes Texas, Arkansas, Louisiana, New Mexico, Arizona and Mexico.

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Sins of Omission: A credit for the article "Investigation of Surface Layer and Wear Behavior of Nitrided Gear Drives," which appeared in our last issue was omitted. Our thanks to Kolene® Corporation, Detroit, MI, who brought the article to our attention and provided technical assistance in its preparation.

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If you have a photo of gears being hobbed, ground, shaped, heat treated, etc. that you think might look great on the next issue of Gear Technology please send it for review to:

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Gear Metrology Standards and ISO 9000

VIEWPOINT

"Viewpoint" provides an opportunity for readers to respond to articles that appear in our pages. Letters should be address to Viewpoint, Gear Technology, P. O. Box 1426, Elk Grove Village, IL, 60009 or faxed to 1-708-437-6618. Letters submitted to "Viewpoint" become the property of *Gear Technology*. Names will be withheld upon request, however, no anonymous letters will be published. We reserve the right to edit letters in the interest of space. Opinions expressed by contributors are not necessarily those of the editor or the publishing staff.

I noted with interest the beginning of *Gear Technology's* three-part series on ISO 9000 certification. I also recently attended Brown & Sharpe's/Leitz gear metrology seminar. Both events caused me to smile and reflect.

For the last 40 or more years, I have been hearing about the deplorable condition of the American gear industry. I heard it again at the Brown & Sharpe/Leitz seminar.

Amy Zuckerman makes an extremely important distinction in her article on ISO 9000. She notes that the emphasis is upon "*quality assurance*" not "*product assurance*." She notes that it is possible to make a very poor product very efficiently and be ISO 9000 certified.

Will that help the gear industry? Will it help any industry? Will it strengthen or improve our position in world trade?

One of the things to be gleaned from the Brown & Sharpe/Leitz metrology seminar is that there are standard measurement gauges which most industrialized nations use. The more frightening fact gleaned from the seminar is that gear measurement equipment and its data have never been traceable to these standards.

Is it a wonder that the American gear industry has been in a "deplorable condition " or a state of confusion?

I have learned of a program called "A National Policy for Gear Metrology from NIST and DOE." For general information, the players are listed as: ASME, AGMA, Penn State University, the National Bureau of Standards and Technology (formerly NSB, now NIST), and the DOE Y-12 Plant.

Having a single traceable standard for gear metrology may go further toward establishing the viability of the American gear industry than anything ISO 9000 may hope to offer. A paramount condition at ISO 9000 is "traceability," but ISO 9000 certifies the traceability of the processes, not the product.

The gear industry may need to evaluate carefully the "promise" of ISO 9000. The gear industry MUST attain a world-verifiable and traceable metrology for its product if it ever hopes to eliminate its "deplorable condition."

It should be noted that this gear metrology investigation began with the "Leitz" last year. It appears that this may be where the money taken from INFAC is now being directed.

I still question the ultimate value of dependency upon national governments to establish the direction and performance of industry.

Clem Miller,
President, Miller & Associates
Crown Point, IN

Designing Hardened & Ground Spur Gears to Operate With Minimum Noise

Robert E. Smith
Robert Errichello
Don McVittie

When designing hardened and ground spur gears to operate with minimum noise, what are the parameters to be considered? Should tip and/or root relief be applied to both wheel and pinion or only to one member? When pinions are enlarged and the wheel reduced, should tip relief be applied? What are the effects on strength, wear and noise? For given ratios with enlarged pinions and reduced wheels, how can the gear set sizes be checked or adjusted to ensure that the best combination has been achieved?

This is a complex question, requiring consideration of both rating and noise. Three of our technical editors have reflected on the subject.

Robert E. Smith replies: The primary consideration for the minimization of noise is to reduce tooth-to-tooth transmission error under the loaded operating conditions. This means making the teeth as conjugate as possible under load; a true involute running with a true involute. For lightly loaded gears, this means making the involutes as perfect as possible. For gears operating under significant loads, the teeth can have profile modifications in the form of tip or root relief to allow for tooth deflections such that they are true involutes when loaded.

Even with profile modifications, there are a couple of other subtle problems that can cause noise that is objectionable to the human ear. Many times unground gears can be less accurate, yet sound more pleasing to the ear than precisely ground gears. This is because random spacing errors and runout or accumulated pitch errors cause a masking noise that tends to cover up the pure tones and harmonics that are generated by the mesh frequencies. The accumulated pitch errors can cause sidebands of mesh frequency that appear like a "white noise" floor level in the spectrum. When grinding gears, it is not unusual for this background noise to be much lower than the gear mesh frequencies. This makes the pure tone effects more objectionable to the human ear, even though the overall dbA level may be the same.

The second subtlety comes from a phenomenon called "ghost harmonics." This is usually a high pitched noise that is caused by waviness in the profile or lead of the teeth. These waves are commonly called undulations. The critical ones lie parallel to

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Robert E. Smith
is the principal in R. E. Smith & Co., Inc., gear consultants in Rochester, NY.

Robert Errichello
is the principal in GEAR-TECH, a gear consulting firm in Albany, CA.

Don McVittie
is president of Gear Engineers, Inc., Seattle WA.

the instantaneous line of contact. They are caused by the final drive gears of the work spindle on the gear generating machine. There will be an integer number of waves in one revolution around the gear being manufactured, and the number will be the same as in the final work spindle drive gear. The result can be a very small waviness in the involute profile and/or lead charts and is frequently ignored in the assessment of these charts. The waviness can often be smaller in amplitude than .0001", yet cause significant noise. However, these peaks can be detected by single flank transmission error testing, as well as by noise analysis through the use of an FFT analyzer. If you know the number of teeth in the final work spindle drive gear, you can predict where the peaks will be found in the spectrum.

A good practical guide to the application of profile modification may be found in the *Gear Handbook*, McGraw-Hill, 1962, Chapter 5, pp. 22 and 23, by P. Dean. He states: "In general, gear teeth which carry a load in excess of 2,000 lbs. per inch of face width for more than a million cycles should be modified. Those under 1,000 lbs. per inch of face do not generally require modification." Formulas for the amount of tip relief on the driving and driven gears are given in this reference. Another reference is the *Handbook of Practical Gear Design*, McGraw-Hill, 1984, by Dudley. This contains several sections on profile modification and generally indicates putting tip relief on both members.

From a noise reduction standpoint, the best advice is not to use more profile modification than is absolutely necessary. It can cause a rapid loss of profile contact ratio and conjugacy. Start out with little or none in order to meet the noise requirements and then life test to see if the parts will survive. Many people start with too much modification, in anticipation of tooth failure, and then wonder why the gears are noisy.

Bob Errichello replies: One of the most important decisions a gear designer makes is selecting the number of teeth in the pinion. There is a preferred number that provides a good balance between pitting resistance, bending strength and scuffing resistance. See AGMA 901-A92 for an algorithm for calculating the preferred number of teeth.

Profile shift (enlarging/reducing) is used to

- Prevent undercut
- Balance specific sliding
- Balance flash temperature
- Balance bending fatigue life
- Avoid narrow top lands.

The profile shift should be large enough to avoid undercut and small enough to avoid narrow top lands. In general, the profile shifts for balanced specific sliding, balanced flash temperature and balanced bending fatigue are different, but nearly the same if the pinion has the preferred number of teeth (see AGMA 901-A92).

Profile modification is used to minimize the detrimental effects of tooth deflections, assembly tolerances and tooth variations. Proper profile modification increases load capacity and reduces noise.

Fig. 1 shows the interference that

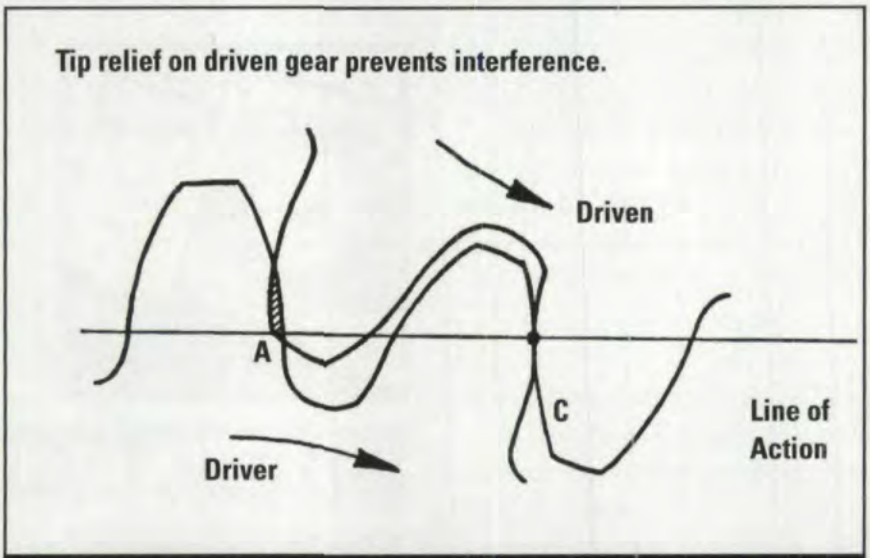

Fig. 1 – Interference occurring at last point of contact (Point A).

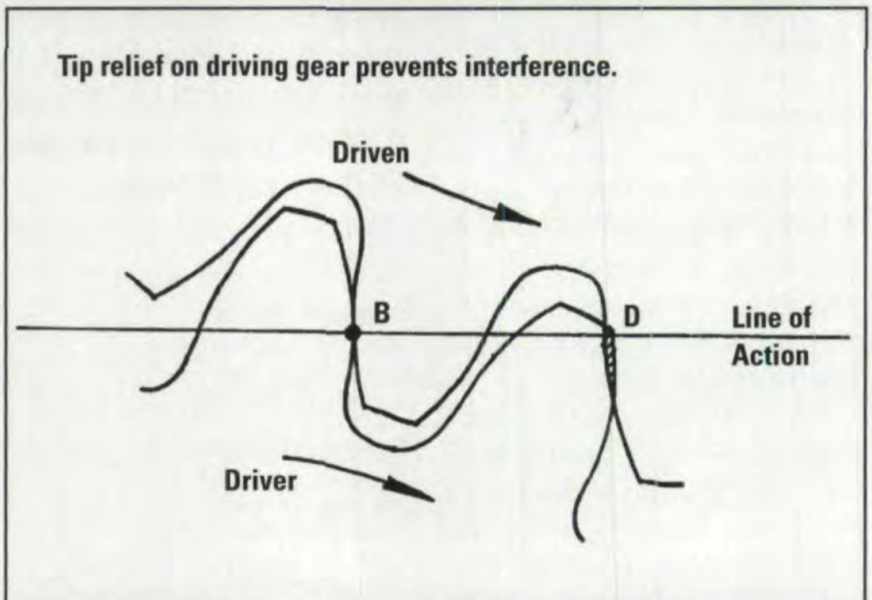

Fig. 2 – Interference occurring at last point of contact (Point D).

occurs at the first point of contact (Point A) with unmodified involutes. The interference is due to deflections of the pair of teeth in contact at Point C, which lengthen the base pitch of the driven gear and shorten the base pitch of the driver. The interference can be eliminated by profile modification that removes material from the tip of the driven gear (shaded zone in Fig. 1).

Fig. 2 shows interference at the last point of contact (Point D) with unmodified involutes. The interference is due to defections of the pair of teeth at Point B, which lengthen the base pitch of the driving gear and shorten the base pitch of the driven gear. The interference can be eliminated by profile modification that

removes material from the tip of the driving gear (shaded zone in Fig. 2).

Experience shows that the profile modification should be larger at the first point of contact than at the last point of contact. Generally the first point of contact is more critical because the oil film is not well developed and the frictional forces are higher as the gear teeth approach the pitch point rather than recede from it.

With speed reducers, the first point of contact is at the start of active profile (SAP) of the pinion teeth where they are especially susceptible to pitting.

With speed increasers, the first point of contact is at the tip of the pinion teeth and the SAP of the gear teeth where both members are especially susceptible to scuffing.

With regard to the kinematics of the gears, it does not matter whether the material is removed in the form of tip relief or root relief. However, the involute profile is more difficult to control near the base circle because the curvature is rapidly changing. Therefore, tip relief is generally preferred over root relief for ease of manufacturing.

Don McVittie replies: The parameters to be considered in determining the amount of profile shift and the distribution of profile shift between gear and pinion are best covered in AGMA 901-A92, *Rational Procedure for the Design of Minimum Volume Gears*, Annex A. The Maag *Gear Book* also has good basic information on why and how. Both are available from AGMA.

Parameters which reduce noise are usually:

- High accuracy, particularly small adjacent pitch variation, good helix (lead) matching and convex rather than concave involute profile deviation.
- High transverse contact ratio. The following increase it:
  - Low operating pressure angles
  - High numbers of teeth
  - Long (extra depth) teeth
  - High axial contact ratio — 2.0 or more. It is increased by:
  - Higher helix angles
  - Finer pitches — smaller modules
  - Wider face width may be bene ficial, but can cause problems with lubrication and load distribution.
  - Some engineers strive for integer values of axial contact ratio, believing that this reduces the variation in stiffness caused by the variable length of the helical lines of contact. This is probably most important with axial contact ratios less than 2.0.

The optimum design varies, depending on the application. For example, the best practice for industrial conveyor drives is much different from that for marine reduction gears or automotive gears. Usually some compromise is required between optimum design for bending strength and optimum design for minimum noise. An optimum design for pitting resistance can be a good starting point. The vibration response of the housing can also be important. A light weight housing with many low natural resonance frequencies will respond to excitations from the gear teeth which wouldn't be important in a heavier housing with higher natural resonant frequencies.

Tip relief can be applied as tip and root relief on the pinion or as tip relief on both gear and pinion. With the pinion driving, the required gear tip relief or pinion root relief is usually about 1/3 more than the required pinion tip relief. It is hard to manufacture accurate pinion root relief if the pinion's start of active profile is near the base circle. Sometimes it's impractical to achieve the required root relief in a few degrees of roll near the pinion base circle, so both parts are tip relieved.

In the U.S., stock hobs used to be furnish with a "ramp," an area of higher pressure angle near the root of the hob tooth, which cut tip relief in both gear and pinion. The amount of tip relief on the part varied with the number of teeth and the profile shift, so the effect was hard to control. Most manufacturers now achieve tip and root relief by grinding or by shaving with specially designed cutters.

For a discussion of how much tip relief to allow and where to start modification on the profile, see Dudley's *Handbook of Practical Gear Design*, pages 8.12 through 8.21.

Properly designed and manufactured tip and root relief on spur gears or tip, root and end relief on helical gears reduce dynamic load by assuring smooth meshing as the teeth engage and disengage at the ends of the lines of contact. Reduction of the dynamic load is beneficial for strength, pitting resistance and noise. It also encourages the formation of an oil film between the teeth which reduces the tendency toward micropitting and abrasive wear. The benefits apply to all heavily loaded gears with or without profile shift. ■

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MAY 16–20
AGMA Training School for Gear Manufacturing. Daley College, Chicago, IL. For more information call AGMA headquarters at (703) 684-0211 or fax (703) 684-0242.

MAY 24–25
AGMA Technical Education Seminar, "Gear Failure Analysis." Hilton Commonwealth Hotel, Florence, KY (Cincinnati). Will cover overload, bending fatigue, Hertzian fatigue, cracking and gear failure photography. For more information, call (703) 684-0211 or fax (703) 684-0242

JUNE 13–15
University of Cincinnati, Center for Industrial Heat Treating Processes. Royce Hotel–Metropolitan Airport, Detroit, MI. For more information, call (513) 556-2709 or 2710 or fax (513) 556-3390.

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UK Gear Metrology Lab, University of Newcastle, Britsh Gear Association. 1994 International Gearing Conference. University of Newcastle. Themes include research, design, components, materials, production and industrial practice. For more information, contact Jane Wallace, Dept. of Mechanical Eng., Stephenson Building, University of Newcastle, Newcastle upon Tyne, England, NE1 7RU.

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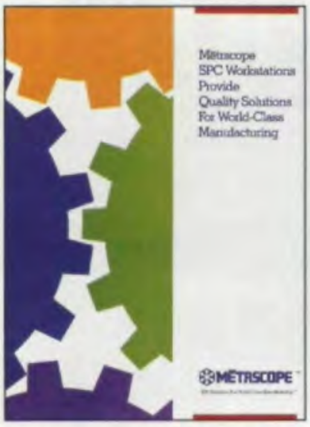

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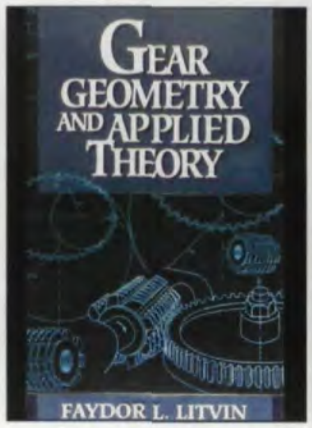

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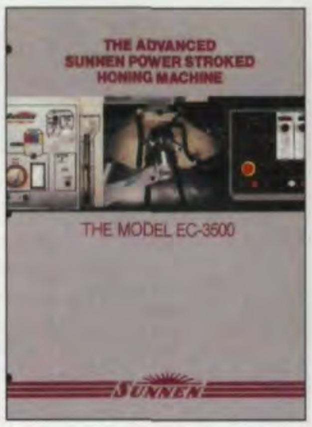

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