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THE JOURNAL OF GEAR MANUFACTURING

SEPTEMBER/OCTOBER 1994

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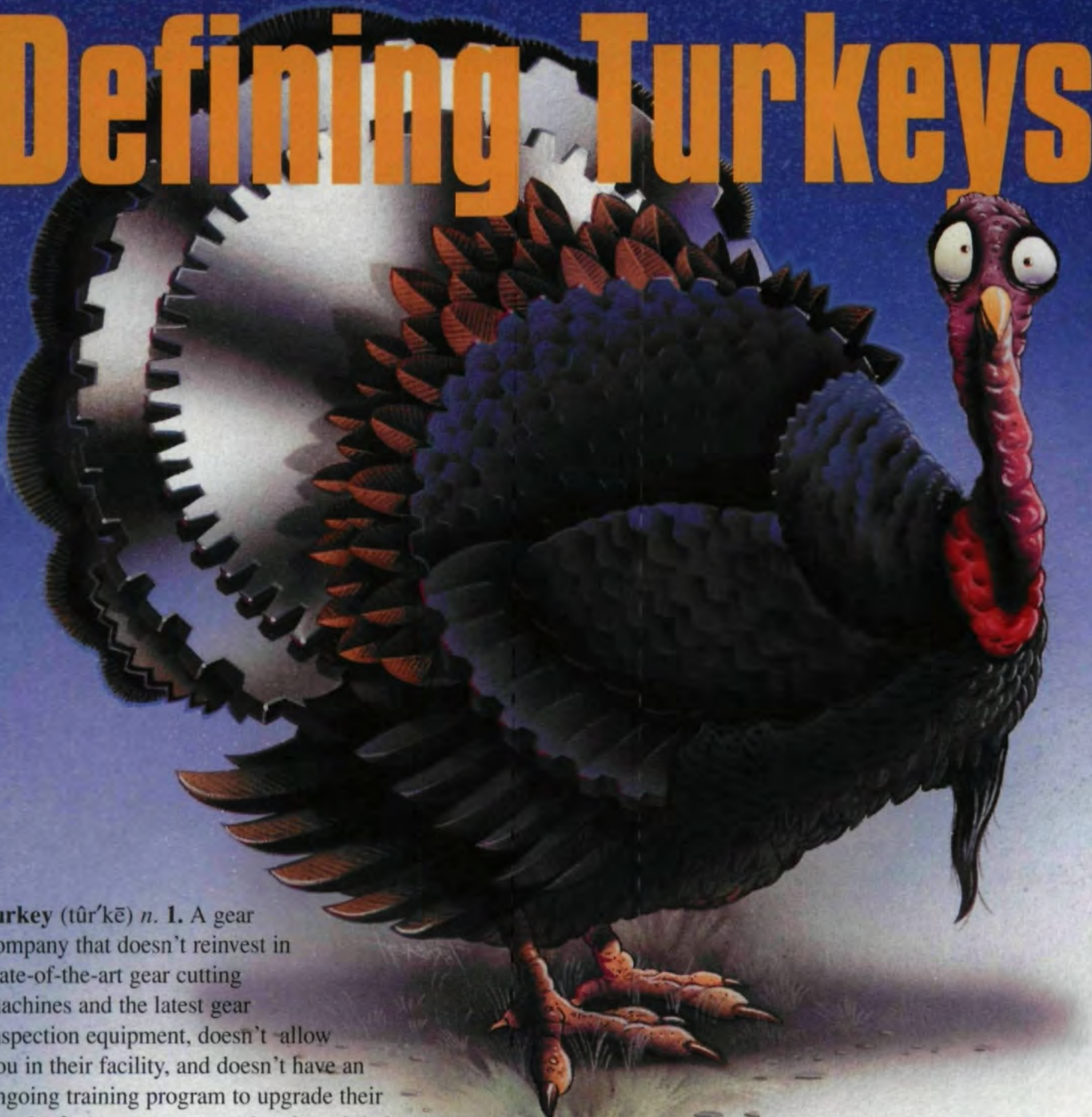

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GEAR TECHNOLOGY: THE NEXT TEN YEARS

A little more than ten years ago this month, the first *Gear Technology* came off the presses. It was a fledgling effort in every respect. The gear industry had never had a magazine of its very own before. Those of us involved in its production were like first-time parents; we were proud and excited, but unsure of what we'd let ourselves in for. None of us knew if this baby could really fly.

But it could, and with the help of you, our loyal readers and advertisers, it did.

Now, as we enter our second decade, we have more than 12,000 readers in 65 countries and a strong core of regular advertisers. We've made a lot of changes—we like to think they're improvements—since 1984, but we have never lost sight of our primary goal, which has been to be the central information resource for our readers. We have tried to report on both basics and cutting-edge issues in all aspects of the gear field, to address your questions and concerns and to keep you abreast of what is happening in our industry. That will continue to be our goal.

However, we do face this anniversary with a certain ambivalence. Our July/August 1994 issue was the last one to appear with the name of Peg Short on the masthead. In some respects, Peg has *been Gear Technology* from the very beginning. In the early days, when our office consisted of one desk, it was Peg's. She did everything: wrote and edited copy, sold ads, negotiated with suppliers and authors, laid out the pages and answered the phones. As we grew, it was Peg's clear editorial eye that made sure our i's were dotted, our t's crossed and our editorial focus sharp. Now Peg has moved on to other editorial responsibilities at another publishing company. Thanks, Peg. We'll miss you.

The man who will guide *Gear Technology*'s editorial and advertising in the coming years is Jim Spalding. Jim has covered the machine tool industry for 25 years. In addition to this editorial background, he brings a wealth of marketing and advertising experience to *Gear Technology* and a new perspective to our organization. Stop by and meet Jim at our *Gear Technology* booth (No. N2-7193) at IMTS '94 in September. He'd like your input on how *Gear Technology* can help you do your job better.

New faces and a new decade demand changes. We want to be able to provide you with far more than just "the same old stuff." We will be growing and changing along with you and the industry. But underlying the changes of the coming years will be our continuing commitment to being your information resource for the gear industry. The package may change, but the content, vital information to help you do your job more effectively, economically and successfully, will remain the same.

PUBLISHER'S PAGE

It's been an exciting ten years for us. We've learned a lot, had a lot of fun and made a lot of good friends—thanks to you. Now we invite you to come along as we begin the next phase of our journey.

Michael Goldstein,
Editor-in-Chief

GEAR TECHNOLOGY ADVERTISERS

SHOW NEW PRODUCTS AT IMTS

THE LATEST IN GEAR MANUFACTURING DESIGN AND TECHNOLOGY WILL BE ON DISPLAY AT IMTS—THE MANUFACTURING MACHINERY INDUSTRY'S MOST IMPORTANT MARKETPLACE—FROM SEPT. 7 TO SEPT. 15 AT CHICAGO'S MCCORMICK PLACE. OUR ADVERTISERS WILL BE DISPLAYING ADVANCES IN HOBBING, GRINDING, SHAPING, BROACHING, LUBRICATING AND OTHER IMPORTANT GEAR MANUFACTURING PROCESSES.

American Pfauter Limited Partnership of Loves Park, IL (Booth E3-4700) will feature the PE150C hobbing machine, the PSA300 shaping machine, the PHS254 hob sharpener, the Pfauter Mikron A15 gear hobbing machine, the CTX twin spindle machine, the GM16AC automatic lathe, the GLD20 sliding head automatic, the GAC65, the Hessapp DVH25 vertical turning machine, the Kapp VAC61 and GAS51 grinding machines, along with Pfauter-Maag cutting tools.

Bourn & Koch Machine Tool Co., Rockford, IL (Booth E2-2003) will feature the CBN gear grinder, a remanufactured Cincinnati 12C slant bed lathe with a NUM CNC Plus control and a multitude of other manufacturing solutions.

Fellows Corporation, Springfield, VT (Booth N2-7182). Fellows Corporation, J&L Metrology and Bryant Grinder Corporation are featuring advanced technology at the VT USA booth. Stop by our booth for the latest gear shaping machinery, gear cutting tools and CNC optical comparators with internal edge sensing. See the new Bryant Ultraline UL2 high-speed precision bore grinder.

Fette Tool Systems Inc., Brookfield, WI (Booth N2-6503) will feature Fette cobalt roughing and finishing end mills for end and center cutting, profiling and pocket milling. Also, they will have carbide indexable tools, including Univex and Twincut developments, heavy duty roughing and finishing hobs, hobs with interchangeable segments and thread rolling axial, radial and tangential heads for faster output than thread cutting.

GMI, Independence, OH (Booth N1-5228) will display the GFB-250/CNC-5 five-axis hard gear finishing machine, with new features to satisfy the requirements of gear manufacturers for super-fine surface finishes on gears.

Inductoheat Inc., Madison Heights, MI (Booth N2-6455) will feature an interactive, multimedia display system showcasing a wide variety of our induction heating capabilities, including standard and custom equipment and power supplies for heat treating, hardening, tempering, brazing, annealing and more. Our flexible, self-contained statiscan for vertically scan-hardening a wide variety of parts will also be on display.

Kanzaki Kokyukoki Manufacturing Company, Ltd., Amagaski, Hyogo, Japan (Booth N1-5228) will display the GFB-250/CNC-5 five-axis hard gear finishing machine, with new features to satisfy the requirements of gear manufacturers for super-fine surface finishes on gears.

Liebherr, Saline, MI (Booth N2-6953) will feature the revolutionary LC82 CNC gear hobbing machine from Liebherr, designed for dry cutting at high speeds. Also on display will be a Klingelnberg gear inspection station, a Lorenz gear shaper, an LS154 CNC and Oerlikon equipment.

Mitsubishi Machine Tool USA Inc., Itasca, IL (Booth E3-4299) will be introducing their new CNC gear grinding machine. It uses either CBN or vitrified wheel technology along with optional dressing capability. Also, they will display their new GC20 CNC gear hobber with state-of-the-art automation and carbide cutting.

Normac, Inc., Arden, NC (Booth N2-7181) will feature the new CBN2T grinding wheel profiling center for gear grinding and other form grinding applications and the Formaster CNC grinding wheel profiler for machine retrofit applications.

Pfauter-Maag Cutting Tools, Loves Park, IL (Booth E3-4700) will display its line of hobs (including the Wafer Hob and Opti-Gash Hob), shaper cutters (including the Wafer Shaper Cutter), milling cutters and shaving cutters.

Reishauer Corporation, Elgin, IL (Booth N2-7168) will feature the newest concept available in production grinding. Reishauer's continuous shift process used on the RZ362A guarantees high process stability, reduced idle times, very low perishable tool costs, quick setups and super-fast grinding times. The RZ362A material handling system will be provided by ASEA Robots.

Sala & BLM Corporation, Elk Grove Village, IL (Booth N2-6782) will feature CNC tube bending and endforming equipment; tube loading, sawing and deburring systems; circular saws for solid steel and tubing; and high-speed saws for aluminum and plastic.

Schunk Intec, Inc., Raleigh, NC (Booth N2-6329) will feature Schunk precision hydraulic chucks & arbors with a TIR of less than .00012" or 3 µm, Schunk gripping systems to facilitate factory automation, Schunk tandem centric clamping blocks for precision work-holding in lathes and machining centers and Schmalz vacuum components.

Sunnen Products Company, St. Louis, MO (Booth N2-7347) is introducing the new high-performance EC-3500 honing machine, which hones most parts twice as fast as previous machines. Also at the booth will be the CGM-5000 Krossgrinding® System, the HL-1500 wide range tube hone, the MBC-1805 power stroked hone, the MBB-1660 honing machine, the CV-616 vertical honing machine, the DMS^3 multi-spindle, single stroke hone, plus a full range of tooling, abrasives, honing fluids, superfinishing and gaging equipment.

Ty Miles Inc., Westchester, IL (Booth N2-7383) will feature a new user-friendly pull-up broaching system for broaching internal gear configurations—machine features influenced by feedback from world class gear broach users include guaranteed fast and easy broach and fixture changeovers, no steps, platforms or pits required, available broach force monitoring and SPC capability.

WMW Machinery Company, Inc., West Nyack, NY (Booth E3-4298). NILES Berlin, exclusively represented by WMW Machinery Company, Inc., will exhibit the new Profile gear grinder. Its two independent CNC-controlled grinding wheel slides, menu-driven CNC control and integrated software guarantee highest productivity. For additional information, please visit WMW Booth E3-4298. ■

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New Innovations in Hobbing — Part I

Robert Phillips
Pfauter-Maag Cutting Tools, L.P.
Loves Park, IL

Prior to the introduction of titanium nitride to the cutting tool industry in the early 1980s, there was very little progress in the general application of hobbing in the gear cutting industry. The productivity gains realized with this new type of coating initiated a very active time of advancement in the gear manufacturing process.

The purpose of this article is to give the reader a general understanding of some of the latest technology in hob design as well as its application. This is not to say that the article is meant to be all-inclusive. There are sure to be recent advancements and ideas in development that are not covered.

Any company that wishes to take full advantage of the latest advancements should contact its hob manufacturer to obtain the necessary help and direction for each application.

One of the biggest driving factors in the development of new processing for gears has been the promotion in this industry of continual improvement. In virtually all levels of design and manufacturing today, the philosophy of continual improvement has led to some rather abstract solutions to specific problems. The first thing that must be agreed on is the elimination of any boundaries that may have been established in prior years.

To help organize the subject matter in this article, a number of key topics have been identified. Each area will be covered in detail. The topics include

- Analytical Evaluation
- Materials
- New Tool Configurations
- Coatings
- Accuracy Improvements
- Dry Hobbing

We should begin by discussing the effects of changing individual features of a hob's design. Greater detail will be offered later in this article; it is only noted at this time to help explain the logic that helped set the direction of some changes and improvement.

Changing Individual Features of Hobs

The first area to explore is the effect of the number of gashes in the hob. Fig. 1 demonstrates that by changing the number of gashes from 10 to 20, the number of cutting edges producing the form doubles. This in turn reduces the chip load, thus increasing the tool life. It also improves the form accuracy by reducing the height of the generating flats.

Another approach is to increase the feed and keep the load constant. Increasing the feed reduces the machining cost.

The next variable to consider is the number of threads in the hob. In some respects, this can be compared to changing the number of gashes. The equivalent number of gashes can be thought of as the number of gashes divided by the number of threads. For example, Fig. 2 shows the effect of changing from a one-thread hob to a three-thread hob, keeping all other variables the same. Although four-gash hobs are uncommon, the effect is very similar. Both the chip load and quality are affected as before.

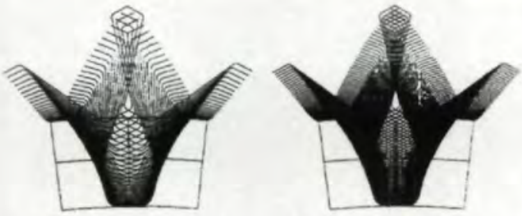


Fig. 1 1-Thread, 10 Gashes 1-Thread, 20 Gashes

One final area to examine in relation to the generating process is the effect of the number of teeth in the part. While this is not necessarily an item that is controlled by the hob design, it is important to see the resulting effect and determine what one might do to offset that effect. In Fig. 3, it is easy to see the impact on chip load, form generation and generating flats when comparing a tooth generated on a 10-tooth part with one generated on a 50-tooth part. Notice specifically the difference in the fillet produced and the amount of "sweeping" or generating that takes place on the 10-tooth part.

The gear manufacturer must realize that the objective in successfully applying the following concepts is to reduce the total manufacturing cost of the gear being evaluated. Shown in Fig. 4, the total manufacturing cost is the summation of the tool cost and the machining cost. To simply concentrate on only one of these factors may result in settling for a cost figure that does not represent the optimum machining rate. As you will see in some of the examples to follow, there are cases where it might be wise to sacrifice tool life to achieve greater gains in reducing the machining cost.

Analytical Evaluation

While there have been many ways to evaluate the performance of an existing application or to predict the results of a new one, one of the systems being used more and more is the comparison of lineal inches cut per hob tooth. With this system, when one knows the hob parameters and part specifications and utilizes life factors that have been developed historically, it is possible to estimate tool life with a certain degree of accuracy.

The first step is to calculate the number of usable teeth in the hob. To do this, it is necessary to determine the usable length of the hob with the following formula:

Usable Length = $HL - HB - NCP - RZ - GZ/2$

where:

HL = Hob Length

HB = Hub Length (total of both sides)

NCP = Normal Circular Pitch

PA = Pressure Angle

RZ = Roughing Zone

$= \sqrt{(\text{part o.d.} - \text{part wd}) \cdot \text{part wd}}$

GZ = Generating Zone

$= (2 \cdot \text{part addendum})/\tan PA$

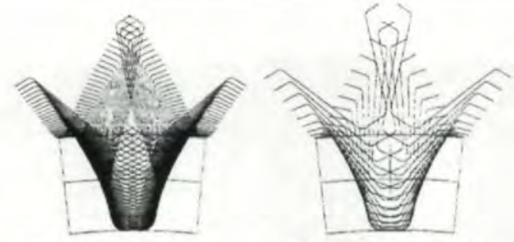


Fig. 2 1-Thread, 10 Gashes 3-Threads, 10 Gashes

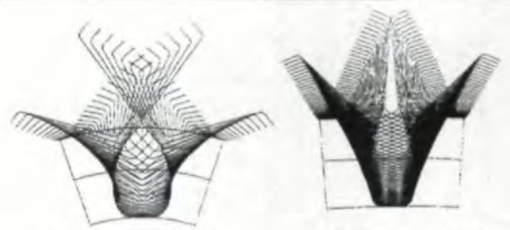


Fig. 3 1-Thread, 10 Gashes, 10 Teeth 3-Threads, 10 Gashes, 50 Teeth

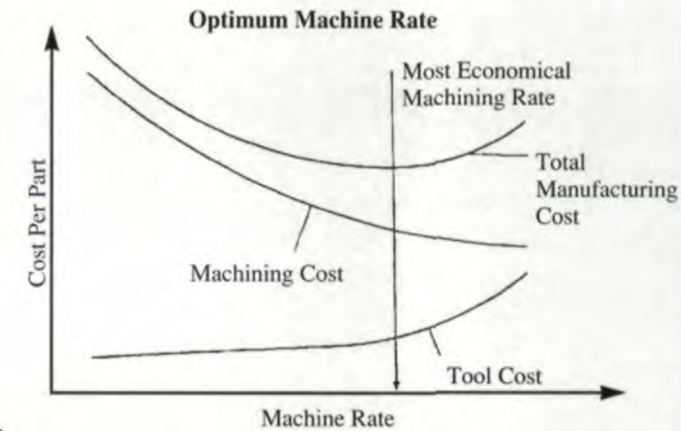


Fig. 4

Usable number of teeth = (Usable length/*NCP*) • number of gashes

Once the usable number of teeth has been calculated, the information shown below can be used to determine the total number of lineal inches (or number of parts) that can be cut per sharpening.

Life Factors at recommended speeds in soft steel

- Uncoated tool—80 lineal inches/tooth
- TiN-coated tool—125 lineal inches/tooth
- Disposable hob—250 lineal inches/tooth

The following example demonstrates how this system can be used:

Hob:	3.00 OD	8.00 length	12 gashes
	10 NDP	.314157 NCP	.125 hubs
Part:	40 teeth	4.2 OD	Spur
	.225 WD	.100 ADD	.75 Face

Usable length = 8.0 – .25 – .314 – .946 – .275 = 6.215"

Usable number of teeth = (6.215/.314) • 12 = 238 teeth

Robert Phillips *is Vice-President of Engineering with Pfauter-Maag Cutting Tools. He is the author of many articles and papers on gear cutting subjects.*

Table 1–Disposition of Allowances, Clearances & Tolerances

HOB DATA	CONVENTIONAL	DISPOSABLE
Diameter	3.5	2.0
Length	7.5	7.5
Number of Threads	4	3
Class	A	A
Material	CPM M53	CPM REX76
Coating	TiN	TiN
CYCLE DATA		
Feed Rate	0.090	0.06
Feed Scallop Depth	0.0002	0.0002
Cutting SFM	300	400
Cutting RPM	327	765
Floor-to-Floor Time (min.)	0.38	0.25

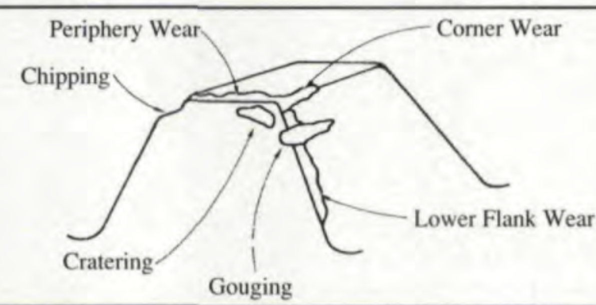


Fig. 5

Micro-chipping | Chipping Caused by Shear Stress | Chipping of Complete Tool Tip | Chipping of Complete Tool Tip | First Peel-back | Catastrophic Failure

Fig. 6

For a TiN-coated tool (125 lineal inches/tooth):

Life/Sharpening = 29,750 lineal inches
= 990 parts

Knowing the usable length of this hob is 6.215", it is possible to calculate the shift per piece.

6.215"/990 parts = .0063"/part

The goal of any of the proposed changes is to reduce the total cost of manufacturing the gear. One area that has been investigated recently is the effect of the outside diameter of the hob being used. In order to keep the cost of the tool low and reduce the approach and overrun dimensions, the designer tries to keep the diameter of the hob as small as possible. The small diameter also allows the hobbing machine to run at higher hob rpms while keeping the surface footage constant. Because of the timed relationship between the hob and the part, it should be obvious that the higher the rpm of the hob, the faster the gear will be produced. In some cases, as shown in Table 1, the use of premium steels, justified by lower material requirements, can also lead to gains in speed with the higher sfm.

Historically, a trade-off always has been required when determining the best possible diameter of the hob. While there are certain advantages to reducing the diameter, the main disadvantage is the reduction in number of sharpenings if the number of gashes is held constant. More recently, however, there has been a new approach to this design compromise. Normally an optimum diameter exists that may be considerably different from those suggested in the past. This will be discussed in greater detail later.

The field of wear analysis may have been somewhat misunderstood in the past. All too often, a wear problem is not noticed until catastrophic failure has taken place. In these cases, it is quite possible that the primary mode of failure is no longer recognizable. To establish some common ground, please refer to Fig. 5, which shows the different types of wear.

Earlier evaluations of the best solution to a specific wear problem may have been, in some cases, exactly opposite of what was correct. For example, the usual correction for an excessive flank-wear problem is to upgrade the substrate steel to a premium grade with increased wear resistance. With today's methods of evaluation, we attempt to review the wear performance as it progresses, leading up to the final failure. When this is done and the tool is investigated under a microscope, we may find that, in fact, the primary mode of failure is premature edge chipping. In that case, the solution may be a steel that has tougher characteristics. This method of evaluation may seem to be quite time-consuming, but if the manufacturer is able to determine the best material for a given application, the time is well spent. In Fig. 6, the progression of this type of wear is shown. Here you can see the wear take the following sequence:

- microchipping
- face chipping caused by shear stress
- chipping of complete tooth tip
- first evidence of peel-back
- catastrophic failure.

New Tool Configurations

The solutions of today's problems often take a direction or configuration that we would not have considered before because of self-imposed boundaries or limits. These limits must not be allowed to interfere with the thought process required to reach acceptable solutions to these new application challenges.

There is a definite compromise in the design of hobs when considering the diameter and the number of gashes versus the number of sharpenings available. In the past, the emphasis has been on achieving the maximum number of sharpenings because this reduced the tool cost per piece. More recently, it has become apparent that when considering the total cost to maintain a hob, a better solution might be to have a tool that is either smaller in diameter or the same diameter with more gashes. While this reduces the number of sharpenings, it also reduces the cost to maintain each hob.

Fig. 7 shows a comparison of the same hob with the only difference being the number of gashes. In this case the design changes from 14 gashes to 24. Because of this, the amount of life also changes from .365" to .132". Now considering the explanation of the advantages in increasing the number of gashes given earlier, it should be clear that there is certainly an optimum number of gashes for a given application. This optimum can change depending on the manufacturer's goal. For example, if the goal is simply to improve the life of the tool, the feed would be kept constant, thus reducing the chip load with the higher number of gashes and in turn reducing the amount of wear for a given number of lineal inches cut. Within these limits, it is possible to calculate the total cost for each number of gashes and arrive at the lowest or "optimum" cost. Table 2 is a spreadsheet that has been developed to make this evaluation quite simple. While the equations for this spreadsheet are not given here, they are available upon request.

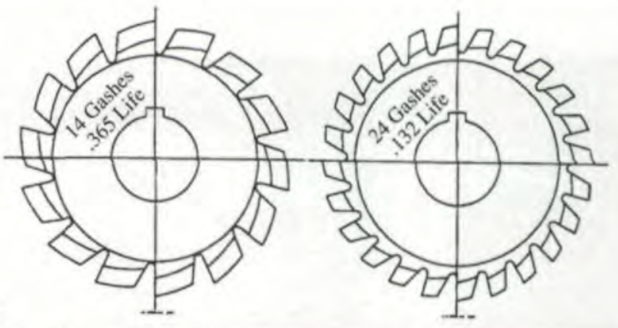


Fig. 7

Table 2—Hob Productive Performance Analysis—Constant Feed

EXAMPLE: Determine tool cost and machining cost per part for possible designs, assuming the hob is run at the same feed and speed as the conventional hob. The increase in the number of gashes is used to reduce chip load, thereby allowing the hob to cut more lineal inches for the same amount of wear.

NUMBER OF GASHES	14	16	18	20	22	24	26	28	32	35
Increase in # gashes	0.0%	14.3%	28.6%	42.9%	57.1%	71.4%	85.7%	100.0%	128.6%	150.0%
Usable tool length	0.326	0.259	0.208	0.169	0.138	0.112	0.090	0.074	0.045	0.031
Decrease in length	0.0%	20.6%	36.2%	48.2%	57.7%	65.6%	72.4%	77.3%	86.2%	90.5%
# of hob sharpenings	32	25	20	16	13	11	9	7	4	3
# of hob uses	33	26	21	17	14	12	10	8	5	4
Lin. inch cut per tooth	82.7	94.5	106.3	118.2	130.0	141.8	153.6	165.4	189.1	206.8
# of usable teeth	325	371	417	464	510	557	603	649	742	812
Lineal inch/hob (x1000)	886	912	932	932	928	947	926	859	702	671
Lineal inches per part	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8
# of parts cut per hob	42570	43807	44781	44755	44597	45492	44492	41280	33698	32250
# of parts per hob use	1290	1685	2132	2633	3186	3791	4449	5160	6740	8062
Hob price	$600	$600	$600	$600	$600	$600	$600	$600	$600	$600
Sharpen & recoat costs	$1600	$1250	$1000	$800	$650	$550	$450	$350	$200	$150
Cost to buy, maintain hob	$2200	$1850	$1600	$1400	$1250	$1150	$1050	$950	$800	$750
Tool cost/part	$0.052	$0.042	$0.036	$0.031	$0.028	$0.025	$0.024	$0.023	$0.024	$0.023
Tool cost/lin. inch	$0.002	$0.002	$0.002	$0.002	$0.001	$0.001	$0.001	$0.001	$0.001	$0.001
Hob RPM	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1
Axial feed rate (IPR)	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100
Chip load factor	.0286	.0250	.0222	.0200	.0182	.0167	.0154	.0143	.0125	.0114
Scallop depth at O.D.	.00079	.00079	.00079	.00079	.00079	.00079	.00079	.00079	.00079	.00079
Scallop depth on flank	.00027	.00027	.00027	.00027	.00027	.00027	.00027	.00027	.00027	.00027
Cutting cycle (minutes)	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24	0.24
Load & unload (minutes)	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10
Floor to floor (minutes)	0.34	0.34	0.34	0.34	0.34	0.34	0.34	0.34	0.34	0.34
Total up-time (hr)	56.4	56.4	56.4	56.4	56.4	56.4	56.4	56.4	56.4	56.4
# parts per hob use	1290	1685	2132	2633	3186	3791	4449	5160	6740	8062
# of hob changes	8	6	5	4	3	3	2	2	1	1
Hob change time (minutes)	25	25	25	25	25	25	25	25	25	25
Total down-time (hr)	3.3	2.5	2.1	1.7	1.3	1.3	0.8	0.8	0.4	0.4
Total production hours	59.8	58.9	58.5	58.1	57.7	57.7	57.3	57.3	56.9	56.9
Total production costs	$1494	$1474	$1463	$1453	$1442	$1442	$1432	$1432	$1421	$1421
Machining cost/part	$0.149	$0.147	$0.146	$0.145	$0.144	$0.144	$0.143	$0.143	$0.142	$0.142
Tool cost/part	$0.05	$0.04	$0.04	$0.03	$0.03	$0.03	$0.02	$0.02	$0.02	$0.02
Machining cost/part	$0.15	$0.15	$0.15	$0.15	$0.14	$0.14	$0.14	$0.14	$0.14	$0.14
Total cost/part	$0.20	$0.19	$0.18	$0.18	$0.17	$0.17	$0.17	$0.17	$0.17	$0.17

Now consider the same application with the goal changed to increased productivity. This will be accomplished by increasing the feed relative to the increase in number of gashes. By drawing on the information regarding the reduced chip thickness with the higher number of gashes, an argument can be developed that it should be possible to increase the feed of a hob with the higher number of gashes to a point where the chip thickness remains constant. In this case, the amount of wear for a given number of lineal inches cut will stay constant even though the feed is increased. Now if we go through the same exercise as in the previous example, we can find the optimum number of gashes for this new set of criteria, which may not be the same as in the previous example (see Table 3). One possibility must be considered at this point. Not all applications have the potential to increase the feed for a number of reasons. One is machine limitations; another is a maximum feed scallop height due to finish requirements or stock allowances for finishing operations. In these cases, the limit can be included

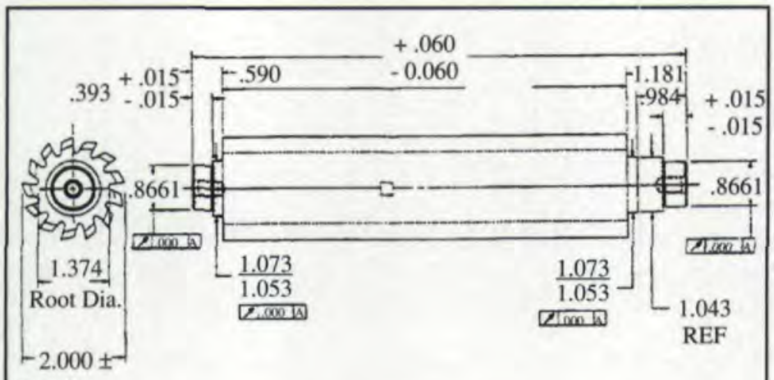


Fig. 8

Table 3—Hob Productive Performance Analysis—Increasing Feed

EXAMPLE: Determine tool cost and machining cost per part for possible designs, assuming the hob is run at a feed rate increased by the percent increase in number of gashes. The lineal inches cut per hob tooth will remain constant. The chip load factor will also remain constant. The feed rate remains constant after the scallop height reaches 0.0005

NUMBER OF GASHES	14	16	18	20	22	24	26	28	32	35
Increase in # gashes	0.0%	14.3%	28.6%	42.9%	57.1%	71.4%	85.7%	100.0%	128.6%	150.0%
Usable tool length	0.326	0.259	0.208	0.169	0.138	0.112	0.090	0.074	0.045	0.031
Decrease in length	0.0%	20.6%	36.2%	48.2%	57.7%	65.6%	72.4%	77.3%	86.2%	90.5%
# of hob sharpenings	32	25	20	16	13	11	9	7	4	3
# of hob uses	33	26	21	17	14	12	10	8	5	4
Lin. inch cut per tooth	82.7	82.7	82.7	82.7	82.7	82.7	82.7	82.7	82.7	82.7
# of usable teeth	325	371	417	464	510	557	603	649	742	812
Lineal inch/hob (x1000)	886	798	725	652	591	552	499	430	307	269
Lineal inches per part	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8	20.8
# of parts cut per hob	42570	38331	34830	31329	28380	26537	23957	20640	14743	12900
# of parts per hob use	1290	1474	1659	1843	2027	2211	2396	2580	2949	3225
Hob price	*$600*	*$600*	*$600*	*$600*	*$600*	*$600*	*$600*	*$600*	*$600*	*$600*
Sharpen & recoat costs	$1600	$1250	$1000	$800	$650	$550	$450	$350	$200	$150
Cost to buy, maintain hob	$2200	$1850	$1600	$1400	$1250	$1150	$1050	$950	$800	$750
Tool cost/part	$0.052	$0.048	$0.046	$0.045	$0.044	$0.043	$0.044	$0.046	$0.054	$0.058
Tool cost/lin. inch	$0.002	$0.002	$0.002	$0.002	$0.002	$0.002	$0.002	$0.002	$0.003	$0.003
Hob RPM	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1	360.1
Axial feed rate (IPR)	0.100	0.114	0.129	0.143	0.143	0.143	0.143	0.143	0.143	0.143
Chip load factor	.0286	.0286	.0286	.0286	.0260	.0238	.0220	.0204	.0179	.0163
Scallop depth at O.D.	.00079	.00103	.00131	.00161	.00161	.00161	.00161	.00161	.00161	.00161
Scallop depth on flank	.00027	.00035	.00045	.00055	.00055	.00055	.00055	.00055	.00055	.00055
Cutting cycle (minutes)	0.24	0.21	0.19	0.17	0.17	0.17	0.17	0.17	0.17	0.17
Load & unload (minutes)	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10
Floor-to-floor (minutes)	0.34	0.31	0.29	0.27	0.27	0.27	0.27	0.27	0.27	0.27
Total up-time (hr)	56.4	51.5	47.6	44.5	44.5	44.5	44.5	44.5	44.5	44.5
# parts per hob use	1290	1474	1659	1843	2027	2211	2396	2580	2949	3225
# of hob changes	8	7	6	5	5	5	4	4	3	3
Hob change time (minutes)	25	25	25	25	25	25	25	25	25	25
Total down-time (hr)	3.3	2.9	2.5	2.1	2.1	2.1	1.7	1.7	1.3	1.3
Total production hours	59.8	54.6	50.1	46.6	46.6	46.6	46.2	46.2	45.8	45.8
Total production costs	$1494	$1360	$1253	$1165	$1165	$1165	$1154	$1154	$1144	$1144
Machining cost/part	$0.149	$0.136	$0.125	$0.116	$0.116	$0.116	$0.115	$0.115	$0.114	$0.114
Tool cost/part	$0.05	$0.05	$0.05	$0.04	$0.04	$0.04	$0.04	$0.05	$0.05	$0.06
Machining cost/part	$0.15	$0.14	$0.13	$0.12	$0.12	$0.12	$0.12	$0.12	$0.11	$0.11
Total cost per part	$0.20	$0.18	$0.17	$0.16	$0.16	$0.16	$0.16	$0.16	$0.17	$0.17

in the evaluation, which then gives the best-fit solution for the given criteria.

Another item that must be taken into consideration is the ability of the end user to sharpen the new tool. If there are limitations on the number of gashes set by index plate availability on the sharpener, the final result may have to be compromised slightly.

Many times the factors that will reduce the machining cost will actually increase the tool cost. In many cases the ratio of machine cost to tool cost may be as high as 20 to 1. In these cases the reduction of machining cost by increasing the feeds and speeds, for example, will more than offset the possible increase in tool cost because of accelerated wear.

To continue this concept to the next level, consider the possibility of either reducing the diameter of the hob or increasing the number of gashes to a point that there is no sharpenable life in the tool. This is, in fact, a disposable hob that is not intended to be sharpened. While each method of obtaining a non-sharpenable tool (reducing diameter, increasing gashes) has its benefits, the method that has been most successful recently is the diameter reduction method. A sketch of a typical disposable hob is shown in Fig. 8. The diameter in this case has been reduced to the point where there is no longer any room for a bore, so a quick-change shank design was developed.

Disposable hobs offer the following quality benefits:

- Uniform part size
- Less accuracy variation
- No arbor runout effects on accuracy
- Reduced cutting forces due to low feed, *high rpm cutting*
- Improved productivity without the need for a higher number of threads.

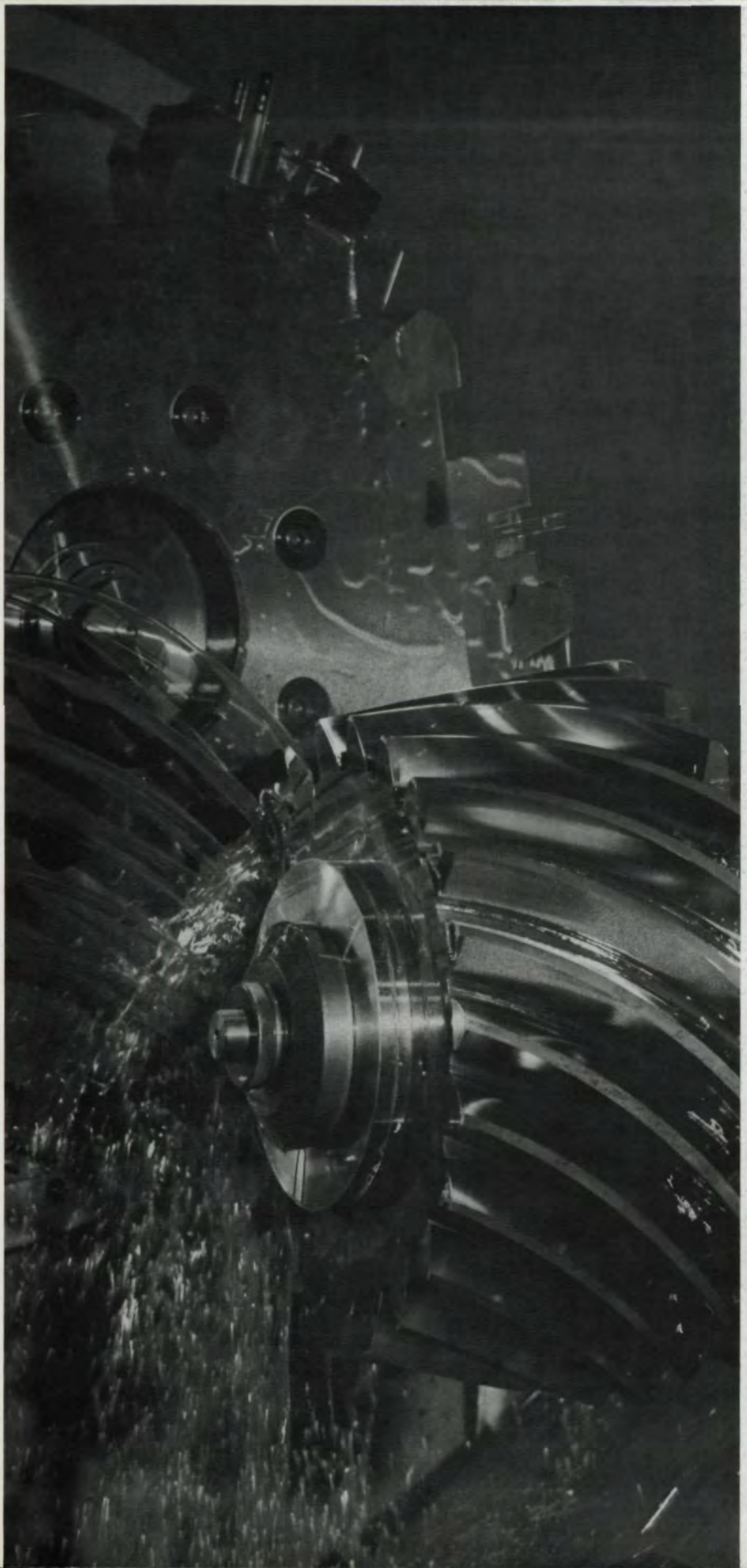
They also offer the following process benefits:

- Productivity gain due to small-diameter tool
- Quick-change tool
- No tool resharpening
- Tool is coated on all surfaces at all times
- No arbor maintenance
- Tool stays on machine longer
- Reduced tool inventory
- Reduced tool damage at changeover.

The success of this type of tool is dependent, of course, on its proper application and the full utilization of its benefits. In order to do this, the hobbing machine must be capable of hob speeds in the range of 2,000 rpms for high-speed steel hobs. This approach is one of the latest examples of application optimization through joint efforts of machine builders and tool manufacturers. ■

Editor's note: *Part II of this article, which will cover accuracy improvement, materials, coatings and dry hobbing, will appear in the next issue.*

Acknowledgement: *This article was first presented at the SME Gear Processing and Manufacturing Clinic, April 1994, Indianapolis, IN.*



A Comparison of ISO 4156/ANSI B92.2M-1980 With Older Imperial Standards

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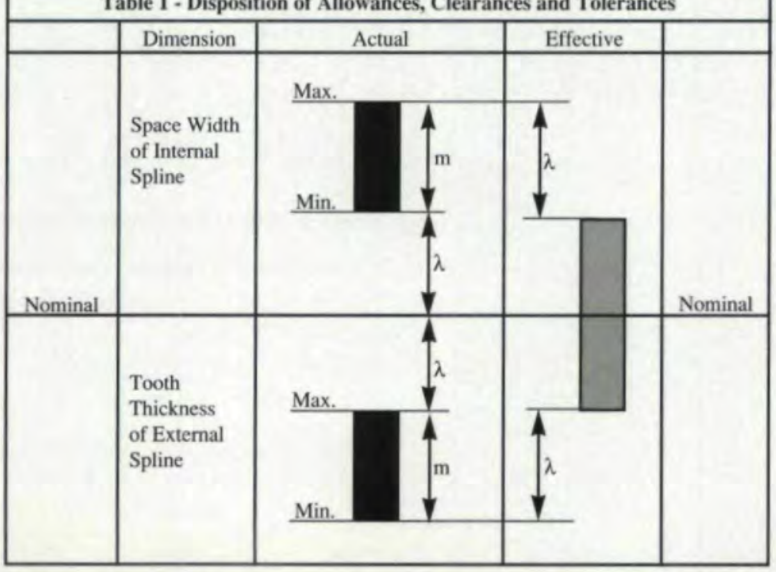
The purpose of this article is to discuss ISO 4156/ANSI B92.2M-1980 and to compare it with other, older standards still in use. In our experience designing and manufacturing spline gauges and other spline measuring or holding devices for splined component manufacturers throughout the world, we are constantly surprised that so many standards have been produced covering what is quite a small subject. Many of the standards are international standards; others are company standards, which are usually based on international standards. Almost all have similarities; that is, they all deal with splines that have involute flanks of 30°, 37.5° or 45° pressure angle and are for the most part flank-fitting or occasionally major-diameter-fitting.

Table 1 - Disposition of Allowances, Clearances and Tolerances



Although ISO 4156 was published in 1981 and is widely used in Europe, we find that it is not used as frequently as it should be in the United States and United Kingdom for a number of reasons. One is that many engineers do not seem to be aware of it; another is that because it is a metric module standard, the circular pitch is different from that of imperial-standard-based diametral pitch splines. This fact can necessitate buying new cutters; however, this should not present any problem where new projects are involved, as a module cutter is no more expensive than a DP (diametral pitch) cutter. Also, DP cutters have to be periodically replaced, so why not change to metric splines? Furthermore, spline gauges designed to a module standard are no more expensive than DP gauges. These arguments, of course, cannot apply to components that have been made for many years, and therefore have to be interchangeable.

In our opinion, the main reason that ISO 4156 is not used more in these countries is lack of awareness. We hope the following is informative to those who have not heard of it and encourages those who are aware of it to use it. We feel this is necessary because quality control these days is of the utmost importance. This standard (unlike the previous imperial standards) has many classes of fits that will guarantee sliding, clearance or interference. Also, in general, components manufactured to this standard are of a higher quality, as tighter tolerances have to be maintained to achieve these better fits. Section 8.1 of the standard is of paramount importance to component manufacturers. This paragraph has wide-reaching implications, as it means that closer tolerances have to be maintained to obtain the desired fits, meaning that quality control during manufacture and gauging will have to be increased, but that is inevitable anyway in today's competitive global markets.

ISO 4156/ANSI B92.2M-1980 — Its Origins

During the 1970s it was realized in the United Kingdom that with metrication in mind, a new standard for involute splines was required. Up to this time and indeed still today, the most widely used standards were BS 3550-1963 and ANSI B92.1-1970. Both of these standards are, of course, imperial standards based on DP. The DP is the number of teeth per inch of pitch-circle diameter and as such provides a standard series of tooth sizes on various pitch-circle diameters. The proposed standard, of course, would have to be, like European standards, metric-module-based. Similar to DP, the module is the ratio of the pitch circle in millimeters to the number of teeth and provides a standard series of tooth sizes. It was not satisfactory simply to convert DP to module because the pitch-circle diameters and tooth proportions would still be imperial-based and would therefore give no degree of interchangeability with metric components.

AFNOR (the French national standards authority) and DIN were consulted to see if there were any standards in existence on which the new standard could be based. The existing French standard E22-141 was considered to be out of date, as it made no reference to variation allowance (discussed later). The German standard DIN 5480 seemed to be the only other possibility, since it did take into account variation allowance. The result of this consultation was the realization that more than a new metric standard was required. What was needed was an international standard that could be used throughout the world.

To achieve this, working groups were established at the various standards organizations; the working groups consisted of representatives from the United Kingdom, France, Germany, Scandinavia, the United States and Japan, among others. At a meeting of all members in Paris, two draft proposals were put forward for consideration: One proposal was based on DIN 5480, and the other was a completely new draft proposal by AFNOR and British Standards. The basis of the tolerances for this draft was taken from ISO 286; i.e., tolerances for plain parts with an additional tolerance for variation allowance.

After discussion, the new draft was adopted and contributions from all member organizations were invited. The result of the collaboration is ISO 4156. This standard also has been published in the United Kingdom as BS 6186, in France as E22-144/E22-145, and in the United States as ANSI B92.1-1980. It is used extensively in Europe and is gradually becoming more and more popular throughout the United States, where transition from imperial to metric is taking place.

Explanation of Variation Allowance & Effective Size

To enable ISO 4156 to be compared with the older standards, it is necessary to understand the concept of effective fit. The effective tooth thickness of the space width of a component is the most important element because it is this size that determines whether the fit is good or bad. It is not only the *actual* tooth thickness or space width that determines the resultant fit between an external and internal spline. This is because, unlike plain parts, splines have multiple engaging surfaces. Each of these surfaces is subject to an error of either profile, spacing/index or parallelism, which must be allowed for if the parts are to fit. This variation allowance is determined by allocating to each of the elements a permissible tolerance. The sum of these tolerances is then referred to as an error allowance e or variation allowance λ.

It is generally accepted in most standards that because it is unlikely that these errors will occur simultaneously and in their maximum amounts on the same spline, a variation allowance consisting of a percentage of the sum of the positive profile error, the index error and the parallelism error for the length of engagement is allowed.

To achieve a suitable fit, the nominal tooth thickness of the external spline and the nominal space width of the internal spline usually are equal to one half of the circular pitch at the

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pitch-circle diameter. The internal spline is allocated a space width greater than nominal. A machining tolerance m is determined to arrive at a maximum and minimum actual space width, and the variation allowance λ is then subtracted from each value to arrive at a maximum and minimum effective space width.

Similarly, the external spline is allocated a tooth thickness that is less than nominal. A machining tolerance is used to arrive at a maximum and minimum actual tooth thickness, and the variation allowance is added to each value to arrive at a maximum and minimum effective tooth thickness. Table 1 shows a typical fit condition between an external spline and an internal one.

ANSI B92.1-1970

This standard is probably the most widely used standard today. As can be seen from Table 2, it shows four classes of fit. The maximum effective tooth thickness of the external spline and the minimum effective space width of the internal spline are equal to nominal. The machining tolerance and effective clearance are different for each class of fit, resulting in Class 4 being the closest fit and Class 7 being the loosest.

According to this standard, the allowable errors for a Class 5 spline (the most widely used) are

For 30 teeth, 24/48 DP, 30° PA
Total Index Variation = 0.038
Profile Variation = +0.005/-0.010
Lead Variation = 0.010 (for a spline length of 25)

As described previously, it is not accepted that all elements will occur in the same position. In this standard, 60% of the accumulated values of total index, twice the positive profile and lead variation are used to calculate the variation allowance; i.e., 0.6 (0.038 + 0.010 + 0.010) = 0.035 (Due to rounding of various elemental errors the value is stated as 0.038 in the ANSI standard.)

For Classes 4, 6 and 7, the machining tolerances and allowable errors for a Class 5 spline are multiplied by the following factors:

Class 4 = x 0.71
Class 6 = x 1.4
Class 7 = x 2.00

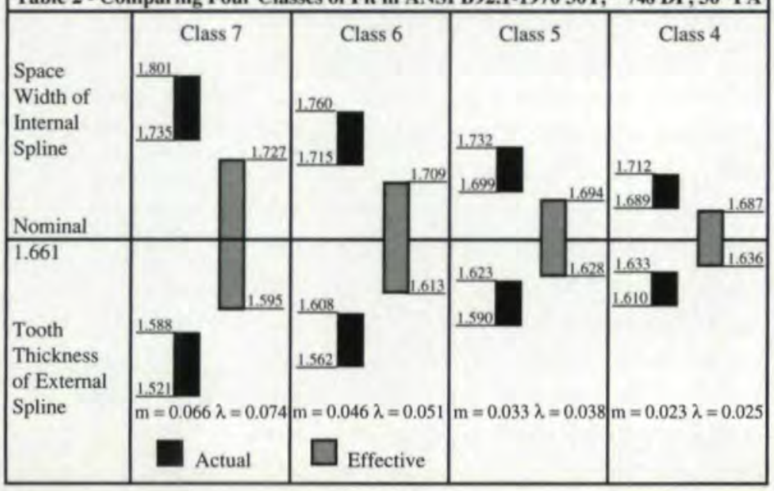
Because the minimum effective space width of the internal and the maximum effective tooth thickness of the external spline are equal, a fit is always guaranteed. To ensure this, the GO plug gauges and the GO ring gauge have a tooth thickness/space width equal to nominal (with a tolerance and wear allowed), which means that one single GO plug gauge and one GO ring gauge can be used for checking all four classes of fit. The NOGO gauges have to be unique to each class of fit.

The only disadvantage of this system is that, while a fit is guaranteed (provided the spline is accepted by the gauges), the type of fit is not. For example, if you were designing components requiring an extremely loose fit, Class 7 tolerances would be used. However, in the case of a poorly made component with splines on maximum metal condition and with excessive profile, index and lead error, it is possible for the gauges to accept the spline, providing, of course, nominal size is not exceeded. The fit obtained from this condition would be a tight fit rather than the desired loose fit.

BS 3550-1960

In this standard, the tooth proportions are similar to those in ANSI B92.1-1970. The allowable elemental errors (profile, spacing and lead errors) are identical to those of a Class 5 spline in ANSI. The only pressure angle referred to is 30°. There are just two classes of fit. The internal splines have only one tolerance band (identical to a Class 5 ANSI spline). For Class 2, both external and internal splines have the same tooth thickness and space width dimensions as an ANSI Class 5 spline. For Class 1 splines, the machining tolerance and variation allowance are identical to Class 2. However, a clearance or deviation allowance Cv is introduced, thus ensuring a

Table 2 - Comparing Four Classes of Fit in ANSI B92.1-1970 30T, 24/48 DP, 30° PA



degree of looseness between the male and female splines. The clearance is shown in Table 3. As shown, the tooth thickness dimensions of the external splines are displaced or adjusted by the clearance *Cv*. It also can be concluded that while BS 3550 has fewer fits, a Class 1 spline does ensure clearance.

DIN 5480

DIN 5480 has been in existence for many years and is predominantly used in Germany, but also is used in many other countries. It is a metric module standard and was therefore considered a basis for ISO 4156. Its popularity is due to its 1) being a metric standard, and 2) offering a vast diversity of fits that ANSI B92.1-1970 and BS 3550 do not. In the past, if you wanted to produce a metric spline, and it was necessary to achieve a certain fit, DIN 5480 was the only metric standard available to enable you to achieve this (the only other metric standards that were available were older standards that did not take into account variation allowance and therefore were unsuitable).

In this standard, the outside diameter is considered as a basis rather than the pitch-circle diameter; i.e., DIN considers that the outside diameter of a spline should conform to a preferential series of whole numbers. To ensure this and to enable common cutting tools to be used in manufacture, it is necessary to correct the tooth profile; i.e., it is not possible for the nominal tooth thickness (0.5 x circular pitch) to occur always on the pitch-circle diameter, but sometimes on a corrected pitch circle, either larger or smaller. This correction does not cause any problem to the spline designer, as all corrected sizes are specified in the standard.

To calculate the variation allowance, 60% of the tooth thickness machining tolerance is used. This means that profile, spacing and lead errors are not specified. This can be a disadvantage if a splined component manufacturer wishes to check these errors on conventional or computerized universal gear checking machines. In the writer's opinion, DIN 5480 is an excellent standard, but the vast choice of fits must cause confusion to the inexperienced. In addition, the machining tolerances for some of the closer fits are extremely tight and the variation allowances consequently minuscule, which must make manufacturing on a production basis difficult. Having said this, these days with modern computerized machinery and measuring machines, which have eliminated a great deal of human error, it is now possible to manufacture components that are accurate and consistent, so perhaps there is nothing wrong in setting limits that were at one time considered impossible. Table 4 shows a small selection of fits from this standard.

Table 3 - Comparing Two Classes of Fit in BS.3550.1963 30T, 24/48 DP, 30° PA

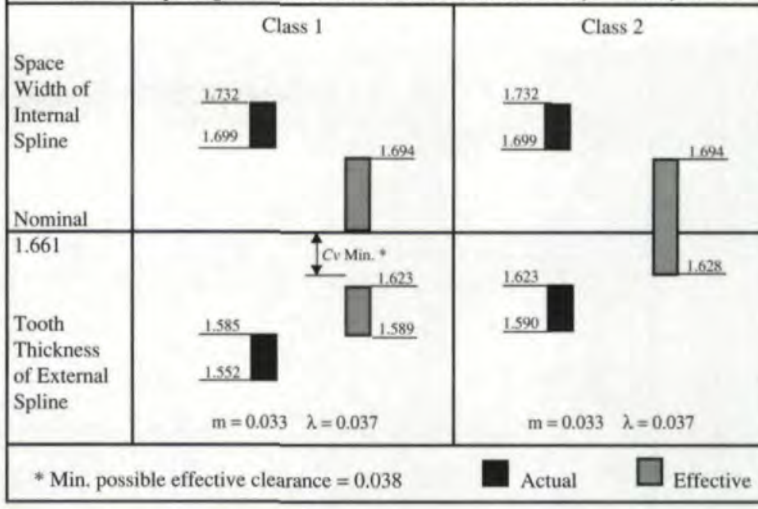
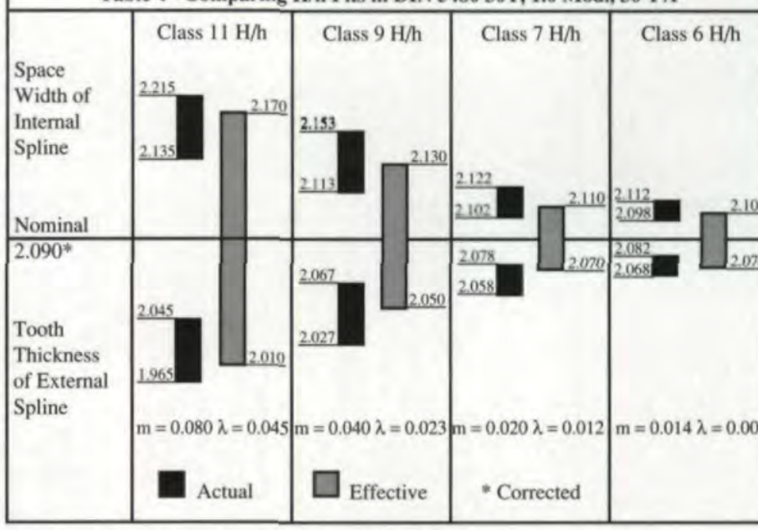


Table 4 - Comparing H/h Fits in DIN 5480 30T, 1.0 Mod., 30°PA



ISO 4156/ANSI B92.2M-1980

This new standard also allows engineers to select various types of fits; i.e., sliding, close sliding, clearance and interference. Because the splines are not outside-diameter-based, the tooth profiles are not corrected, and the resultant outside diameters are not whole numbers and, consequently, do not conform to a preferred series. This, however, does not present any problem, because a suitable tooth number can be selected for a particular application.

The nominal tooth thickness/space width is calculated in the same way as ANSI B92.1-1970

Table 5 - Comparing Four Classes of Fit H/h in ISO 4156/ANSI B92.2M - 1980 30T, 1.0 Mod., 30° PA

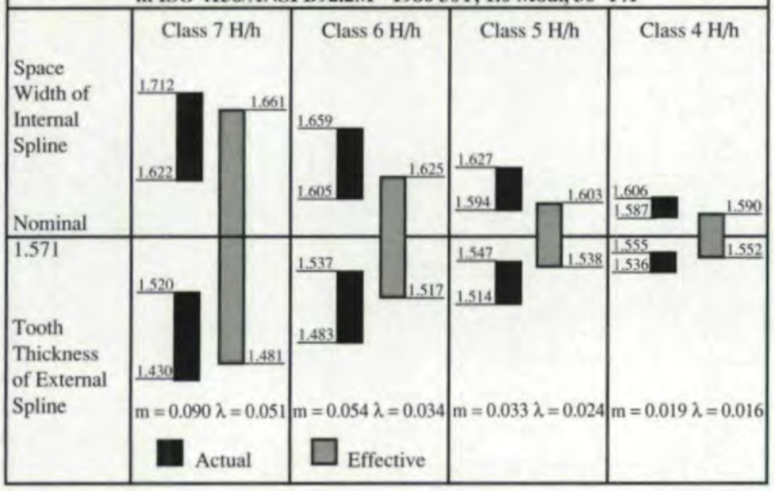
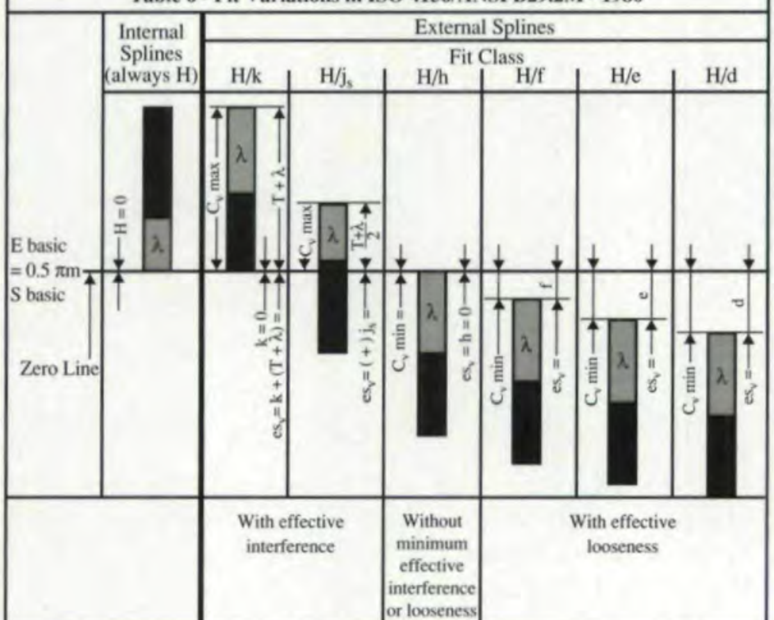


Table 6 - Fit Variations in ISO 4156/ANSI B29.2M - 1980



Graphical representation of deviation allowances for the spline fit classes

and BS 3550; i.e., 0.5 x circular pitch. There are four basic classes of fit, referred to as 4H/h, 5H/h, 6H/h and 7H/h. The suffix H refers to the internal spline and h to the external (as described in ISO 286 limits and fits for plain parts).

Table 5 shows the four H/h fits and can be compared with the four ANSI fits in Table 2. As can be seen, the fits obtained for Classes 4, 5 and 6 in ISO 4156 are significantly closer than those from the older ANSI standard (particularly Classes 4 and 5). The machining tolerances for both standards are approximately equal; consequently, the closer fits obtained in ISO 4156 are due to a reduction in the variation allowance λ.

As stated earlier, Section 8.1 of the standard has wide-ranging implications for the spline manufacturer. It states that the variation allowance λ should be calculated using the following expression:

$$\lambda = 0.6\sqrt{\text{Index Variation}^2 + \text{Profile}^2 + \text{Lead Variation}^2}$$

The expression is different from that used in ANSI B92.1-1970 and BS 3550-1960, which states that variation allowance is calculated as follows:

$$\lambda = 0.6\,[\text{Index Variation} + 2\,(\text{Positive Profile Variation}) + \text{Lead Variation}]$$

The result of the expression specified in the newer standard is a value that is less than that of the old standard, as can be seen from the following example. A 30-tooth, 1.0 module, 30° PA Class 5 spline is used.

Total Index Variation = Fp
Fp in micrometers = $3.55\sqrt{L} + 9$
Fp in micrometers = $3.55\sqrt{47.124} + 9$
Fp in micrometers = 33.370
Fp in millimeters = 0.033

where the length of arc L = 0.5 x Pitch Circle Circumference.

Total Profile Variation = Ff
Ff in micrometers = $2.5\,\phi f + 16$
Ff in micrometers = $2.5(1.375) + 16$
Ff in micrometers = 19.4375
Ff in millimeters = 0.019

where Tolerance Factor $\phi f = m + 0.0125\, m.Z.$

Total Lead Variation = F
F in Micrometers = $1\sqrt{g} + 5$
F in Micrometers = $1\sqrt{25} + 5$
F in Micrometers = 10
F in Millimeters = 0.010

where g is the spline length in millimeters.

Substituting the above values into the new expression in ISO 4156:

$$\lambda = 0.6\sqrt{(0.033)^2 + (0.019)^2 + (0.010)^2} = 0.024.$$

Referring to Table 2, we see that the allowable errors of index, profile and lead for an ANSI B92.1-1970 spline are reasonably comparable to the above (it must, of course, be appreciated that the above value for Ff is total and should therefore be interpreted as ± 0.0095).

The variation allowance, however, for an ISO 4156 spline is less than that for an ANSI B92.1-1970 spline; consequently, when machining splines to the later standard, greater care and more rigorous quality control have to be applied.

The above demonstrates that by using ISO 4156, it is possible to maintain closer fits than those obtained from the older imperial standards.

Further Fit Variations

By using ISO 4156, it is possible to achieve fits other than those shown in Table 5; i.e., the H/h fits. This is made possible by introducing further fit classes to the external spline while maintaining the H fits on the internal spline (it is beneficial to adjust external splines, as these are usually machined, while many internal splines are produced by broaching).

These additional fits are designated f, e, d, j_s and k, and are obtained by applying a deviation allowance (Cv) to the shaft-tooth thickness. The deviation allowances to guarantee clearance for a 30-tooth, 1.0 module spline are as below:

$$f = -0.020$$
$$e = -0.040$$
$$d = -0.065$$

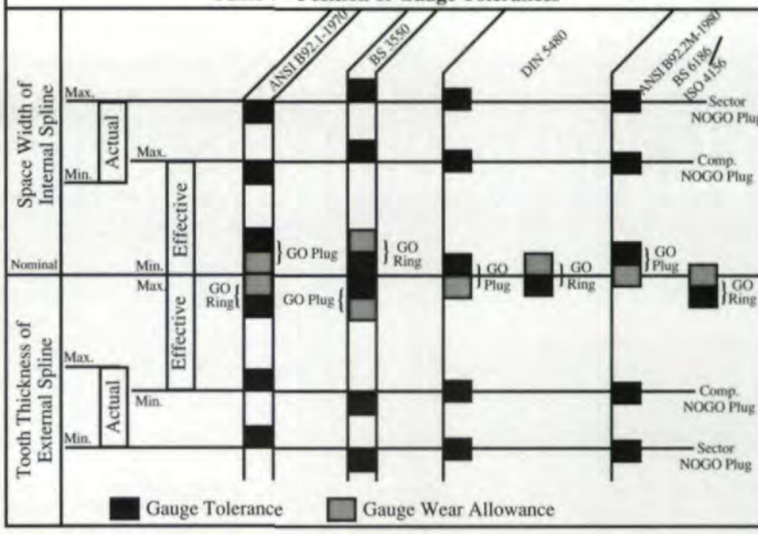
The clearance becomes progressively larger from f to d and ensures looseness. If interference is required, it is possible to employ fits k or j_s. Fit k guarantees interference, and with j_s it is most probable that interference will exist. Spline tooth thickness values for the two fits can be determined by referring to Table 6, which also shows the clearance fits.

It should be remembered that when using fits f, e or d, the shaft major diameter and minor diameter also should be adjusted by a value equal to the deviation allowance divided by the tangent of the pressure angle.

ISO 4156 specifies only side-fitting splines. The older imperial standards also contained major-diameter-fitting splines, which could be an advantage in maintaining concentricity; in this instance, the spline flanks were used as drivers only.

The fact that ISO 4156 does not contain major-fitting splines should be offset by the greater accuracies achievable by using the close tooth thickness tolerances recommended in this standard. This, however, may be debatable by spline manufacturers, who find the major diameter easier to control than the tooth thickness.

Table 7 - Position of Gauge Tolerances



Comparison of Gauging Procedure

All the standards that have been used for comparison have a section that clearly defines the method of tolerancing for gauging. Note the following points upon which all standards are in agreement:

An internal spline should at the very least be checked with

1. a full-form GO plug gauge designed to check "min. effective" space width,

2. a sector-type NOGO plug gauge designed to check "max. actual" space width,

3. an additional composite NOGO plug gauge to check "max. effective" space width (this is very occasionally required if it is necessary to restrict the effective size).

An external spline should at the very least be checked with

1. a full-form GO ring gauge designed to check "max. effective" tooth thickness,

2. a sector-type NOGO ring gauge designed to check "min. actual" tooth thickness,

3. an additional composite NOGO ring gauge designed to check "min. effective" tooth thickness where it is necessary to restrict the effective size.

The position of the gauge tolerance and wear allowances is the place where the various standards do not agree. As can be seen from Table 7, ANSI B92.1-1970 places the gauge tolerance within the part tolerance, BS 3550 puts the gauge tolerance outside the part tolerance, and in DIN 5480 and ISO 4156, the gauge tolerances are bilateral. ■

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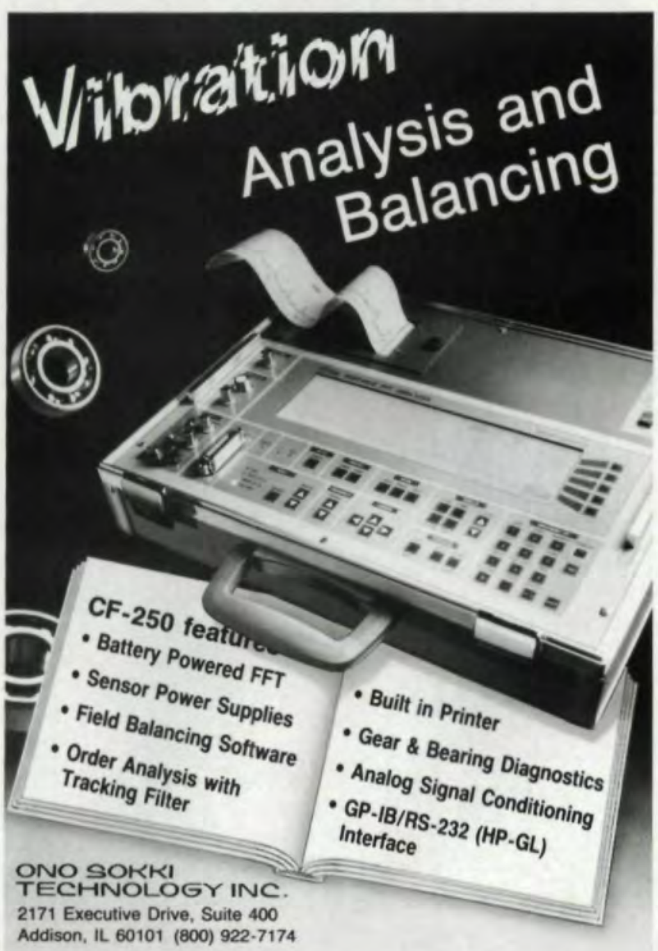
CIRCLE A-16 on READER REPLY CARD



CIRCLE A-15 on READER REPLY CARD

CIRCLE A-17 on READER REPLY CARD

PEOPLE ON THE MOVE

Gear Research Institute

Evanston, IL. The **Gear Research Institute** has moved its headquarters from Lisle, IL, to the Basic Industries Research Laboratory (BIRL) at Northwestern University in Evanston, IL. The move is the result of an agreement whereby BIRL will manage the Institute.

GRI's character and goals will remain unchanged. It will operate under the same advisory board, officers and bylaws and will continue to be a source for cooperative industry-driven research.

Considerable synergism between GRI and BIRL is expected. BIRL is dedicated to the commercialization of technology and is currently working on issues related to wear and corrosion resistant coatings, advanced materials, sensors and controls, cathodic protection and electromagnetic interference. The BIRL facility is a 130,000 sq. ft., well-equipped research laboratory.

The institute's new address is Gear Research Institute, BIRL/NU, 1801 Maple Avenue, Evanston, IL 60201. Phone: (708) 491-5900. Fax: (708) 491-5986.

Columbia Gear

Avon, MN. Columbia Gear Corporation, a custom gearing design and manufacturing company, has achieved ISO-9002 certification for its manufacturing and assembly operations. It is one of the first North American custom gearing suppliers to achieve ISO certification.

Columbia Gear, a subsidiary of Vesper Corporation, is a supplier of custom spur and helical gearing and related services.

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GEAR FUNDAMENTALS

Quality Gear Inspection — Part I

Robert E. Smith
R. E. Smith & Co., Inc.
Rochester, NY

Introduction

Quality gear inspection means doing the "right" inspections "right." A lot of time and money can be spent doing the wrong types of inspection related to function and doing them incorrectly. As we will discover later, such things as runout can creep into the manufacturing and inspection process and completely ruin any piece of data that is taken. This is one of the most important problems to control for quality inspection.

There are two main reasons for performing any gear inspection:

1. Meeting print specifications. The specifications, such as AGMA 390.03 or 2000-A88 criteria, may be included on the print. They also may be spelled out independently for such things as profile, lead, pitch, etc. The goal of such an inspection is making the parts meet the expectations of the user. Meeting an AGMA quality level has a lot more to do with meeting the rating requirements of the gears than it does with aspects of function such as noise and transmission error.

2. Diagnostic purposes. These inspections are done for the purposes of solving machining problems and controlling noise and transmission error. Many times these will require some nontraditional methods of evaluation.

In this article, more time will be spent on the nontraditional diagnostic techniques than on the traditional AGMA level characteristics. AGMA 2000-A88 has much more detail and better definitions for the inspection of gears than AGMA 390.03. Anyone involved in gear specification and inspection should have a copy of AGMA 2000-A88 and become very familiar with the measuring methods and interpretations described within.

Good Data on Purpose

It is very easy to get bad data and not know it. In order to have confidence in the data that one takes, it is imperative to develop good techniques. This section will deal with some of the problems that may be encountered with various measuring methods.

Runout Control. Runout anywhere in the manufacturing or inspection process is one of the most damaging factors affecting gear quality. It will have an impact on anything else that one wants to do or measure. Factors affecting runout include:

Blank Quality. Quality blanks are the foundation upon which a good gear is built. It does no good to expect gear characteristics such as lead, profile and spacing to come out within a few ten-thousandths of an inch when one allows the bore tolerance to vary one or several

Photo courtesy of Le Count Inc., White River Junction, VT.

Fig. 1 — Expanding mandrels.

thousandths of an inch. Certainly, it is easier for an operator to control the size of a simple circle than it is to control that of many gear teeth with a complex form such as an involute. Even a perfect machine can't make a good gear out of a bad blank. Even if it did, the gear wouldn't work in the application, nor would the inspection come out right unless the mounting were duplicated.

Tooling Quality. Good tooling in machining and inspection is a necessity for the manufacture and inspection of quality gears. If a gear is put on a solid arbor, there must be some clearance between the bore and arbor. If there is clearance, there will be some eccentricity, which will result in runout; therefore, it is best to use some kind of expanding arbor. This can fill up the clearance and center the blank on the axis of rotation. Some types of expanding arbors are shown in Figs. 1 and 2. They can consist of expanding collets, hydraulic expansion types, interference ball sleeves or precision expanding jaw arbors that can control runout to .0001 or .0002".

Proofspots. Proofspots are reference bands (axial and radial) that are machined true with the actual bore, journals and shoulders of gear blanks (see Fig. 3). These can be checked while mounted on the machine that finishes the teeth, while mounted on the inspection machine or while mounted in the final application. For even more accurate work, the proofspots are checked and marked for the amount and location of the high point of runout. This high point and amount are duplicated at every step in the process to control the very highest quality gears.

Centers in shafts and their relation to teeth and journals. Unfortunately, most inspection machines for cylindrical gears use centers for the mounting of gears. This is most convenient, but not very accurate. "Murphy's Law" says that even though the journals and teeth are machined from centers, they will never run true again. Gears are rarely run on centers in their applications. They are mounted from the journals, and the teeth should be inspected from the journals. At the very least, the journal should be checked on the inspection machine before the teeth. Some new CNC machines will check teeth relative to the journals, even though they may mount the part from centers.

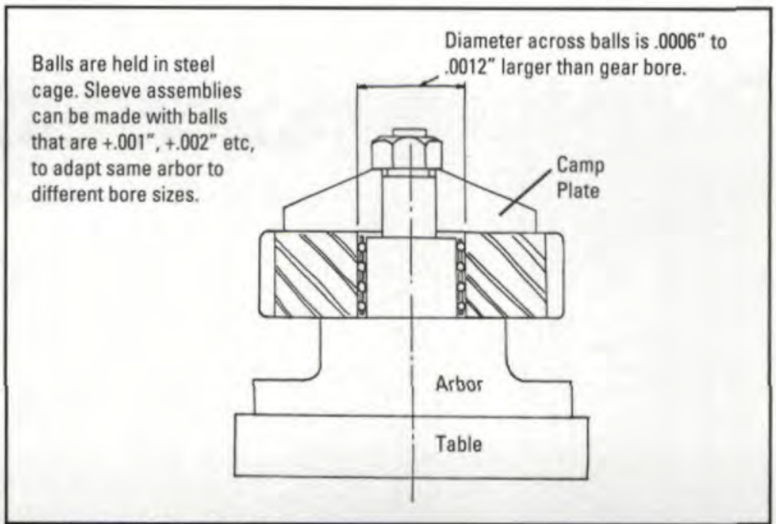

Fig. 2 — Ball sleeve arbor.

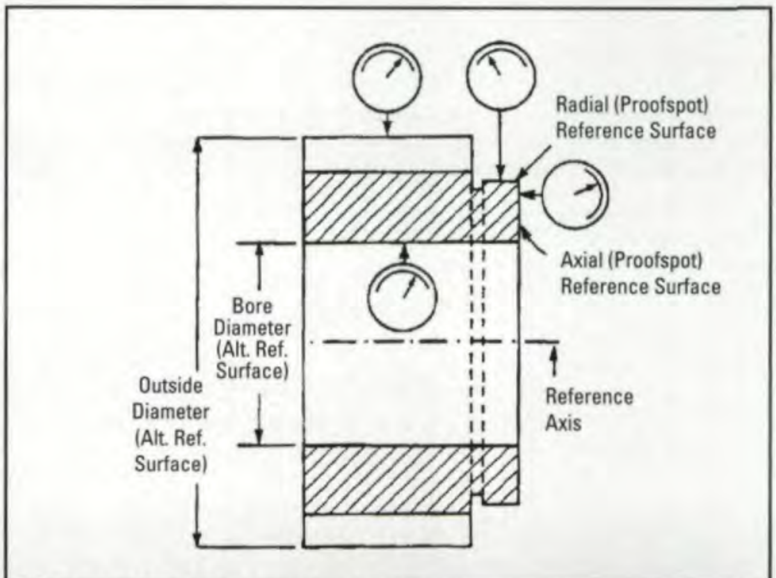

Fig. 3 — Proofspots (reference surfaces).

Remember, if runout exists in the inspection operation, it will influence anything else measured about the gear, such as profile, lead and pitch variation.

Spacing. The term *spacing* is used as a general term to refer to the accuracy with which teeth are positioned around a gear. Spacing has no numerical value and refers only to a group of numerically valued tooth position measurements such as pitch or index. Many of the terms used to describe spacing are very confusing and often misused. For that reason it is important to be careful about what someone is describing. Symbols are very useful in order to keep terms straight. Spacing issues to be addressed in inspection include:

Tooth-to-tooth spacing. This is probably one of the most misunderstood terms. It was used in older versions of AGMA 390 and was an unfortunate choice of words. Many people

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Single-Probe—Precision Method				Two-Probe Method				
A Accumulated Pitch Variation	B Diff. Between Readings in Column A		C Dif. Between Adj. Pitches	D		E Diff. Between Adj. Pitches	F Readings Minus Average	G Accumulated Pitch Variation
Index Variation V_x	Pitch Variation V_p		Spacing Variation V_s	Teeth	Readings	Spacing Variation V_s	Pitch Variation V_p	Index Variation V_x
A 0								0 Ref.
B +2	B minus A	+2	2	A to B	0 →	2	+2	+2
C +2	C minus B	0	2	B to C	-2 →	2	0	+2
D +4 } 6	D minus C	+2	8	C to D	0 →	8	+2	+4 } 6
E -2	E minus D	-6	8	D to E	-8 →	8	-6	-2
F 0	F minus E	+2	4	E to F	0 →	4	+2	0
G -2	G minus F	-2	4	F to G	-4 →	4	-2	-2
A 0	A minus G	+2	0	G to A	0 →	0	+2	0
Total Index Variation $V_{ap}=6$	Max. Pitch Variation $V_p=-6$		Max. Spacing Variation $V_s=8$	$N=7$	-14 Sum -2 Avg.	Max. Spacing Variation $V_s=8$	Max. Pitch Variation $V_p=-6$	Total Accumulated Pitch Variation $V_{ap}=6$

Fig. 4 — Relationships of pitch, spacing and index spacing or accumulated pitch.

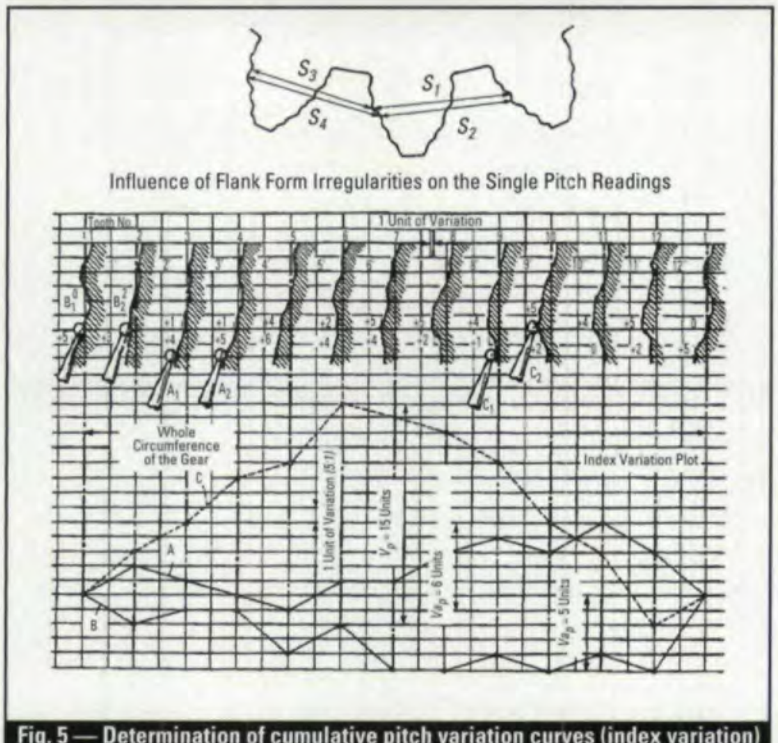

Fig. 5 — Determination of cumulative pitch variation curves (index variation) on the basis of single or span pitch measurements.

(probably most) visualize it as the error between one tooth and the next (involves two teeth). In reality, it is not. It is the difference between one pitch and the adjacent pitch (involves three teeth). A better choice of words would have been *adjacent pitch difference.* AGMA 2000-A88 no longer uses the term tooth-to-tooth spacing. It is now defined as *spacing variation,* V_s. The ISO symbol is f_u, but an earlier proposal, Δf_p, was more descriptive. This parameter has no useful purpose and is not toleranced in AGMA or ISO standards.

Pitch Variation. Pitch variation is the algebraic plus or minus (±) difference in the transverse plane between the true position pitch and an actual pitch measurement. This is the correct measure of the placement of a tooth relative to its adjacent tooth. This factor is most important for evaluating the positional accuracy of a gear and the resulting influence on tooth stresses. The AGMA symbol is V_p and the ISO symbol is f_p. This is the measure of spacing that is toleranced in the AGMA and ISO standards.

Index Variation. Index variation is the displacement of any tooth from its theoretical position relative to a datum tooth. This is the actual location of any tooth around a gear. It is not toleranced, but other values relative to spacing, such as pitch variation and accumulated pitch variation, can be easily interpreted from this data.

Problems With Spacing Measurement. Spacing is usually measured by one of two systems—the *two-probe system* and the *single-probe/index system.* The two-probe system is more likely to produce bad data than the single-probe/index system because of the influence of surface finish (irregularities) on the accuracy of data. The influence is greater on calculation of accumulated pitch variation than it is on pitch variation. Fig. 4 is an illustration from Section 9 of AGMA 2000-A88. It shows the correlation between a single-probe/index measurement and a two-probe measurement of the same gear. In this case, the gear is a theoretical one and has perfect finish. As one can see, in columns A, G, B and F, the results are the same for either measuring method. However, in reality, the results will never be the same, and the worse the finish, the worse the correlation.

More realistic results are evident in the data shown in Fig. 5. This is from Appendix E of AGMA 2000-A88. The upper part of the figure shows what happens because of surface irregularities to the span distance between two probes placed at slightly different locations on the teeth. The bottom part of the figure shows the numerical results of measuring a gear three times with the probes located in different positions. Because of surface irregularities, there will be a greater scatter in the results from the two-probe method. The difference is caused by the accumulation of surface measuring errors as well as pitch errors in the calculation process of the two-probe system. This calculation is not necessary in the single-probe system. The reading for each tooth is direct, and surface errors affect only that one reading.

Measuring the gear several times with the probe in a slightly different position and then averaging the results will give a more accurate answer for the true pitch and accumulated pitch values. This would be true for either system.

Another advantage of the single-probe data is that it gives a direct visual picture of the placement of each tooth around a gear. Pitch variation is the distance between any two adjacent readings, and accumulated pitch variation is the total distance between the highest and lowest reading. When using the two-probe system, it is necessary to calculate the average measured pitch, subtract this from each reading to obtain pitch variation values, and then accumulate the pitch variation readings to obtain the accumulated pitch.

Problems With Size Measurement. The specified size of gear teeth, their circular tooth thickness, is not measured directly. It is usually measured by one of four other means: vernier tooth calipers (chordal tooth thickness), over pins or wires, span measurement (base tooth thickness) or by double-flank composite methods, which measure functional tooth thickness (see Fig. 6).

The first three of these methods also have the limitation that the measuring device only touches a few points on one or a few teeth. Therefore, they may not find the largest tooth size around the gear. They also measure the tooth size independent of the rotational axis of the gear. Even if the tooth size is correct, runout will vary the functional tooth thickness. Some teeth will be functionally larger than others if they are at the high point of runout. It is necessary to make allowances for other variations in tooth shape and position (involute, lead, runout, etc.) in relation to gear quality level (see Fig. 7). If this is not done, the gear may run out of backlash with its mate.

The double-flank composite method will find the largest functional tooth thickness anywhere around the gear. It is probably the best

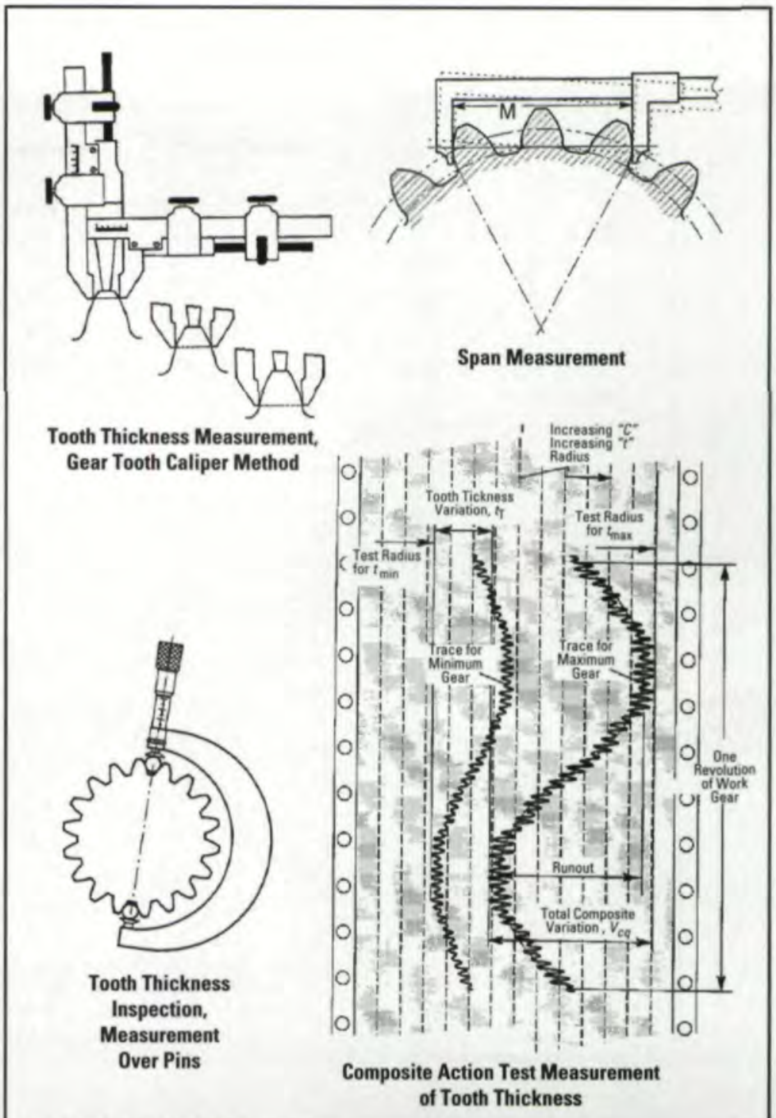

Fig. 6 — Method of size measurement.

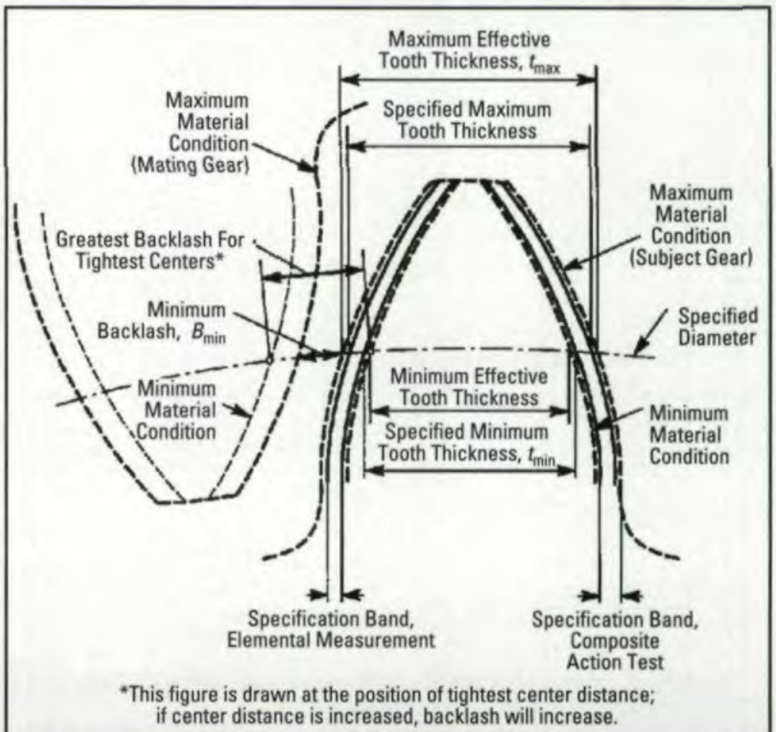

Fig. 7 — Tooth thickness, transverse plane.

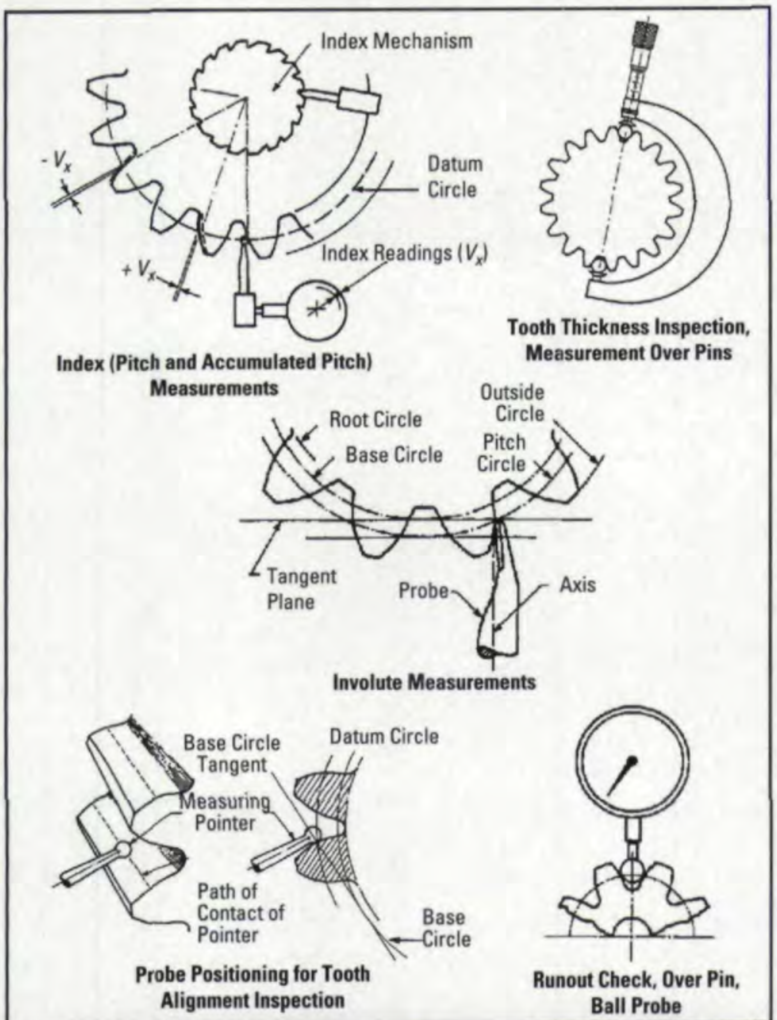

Fig. 8 — Elemental measurements.

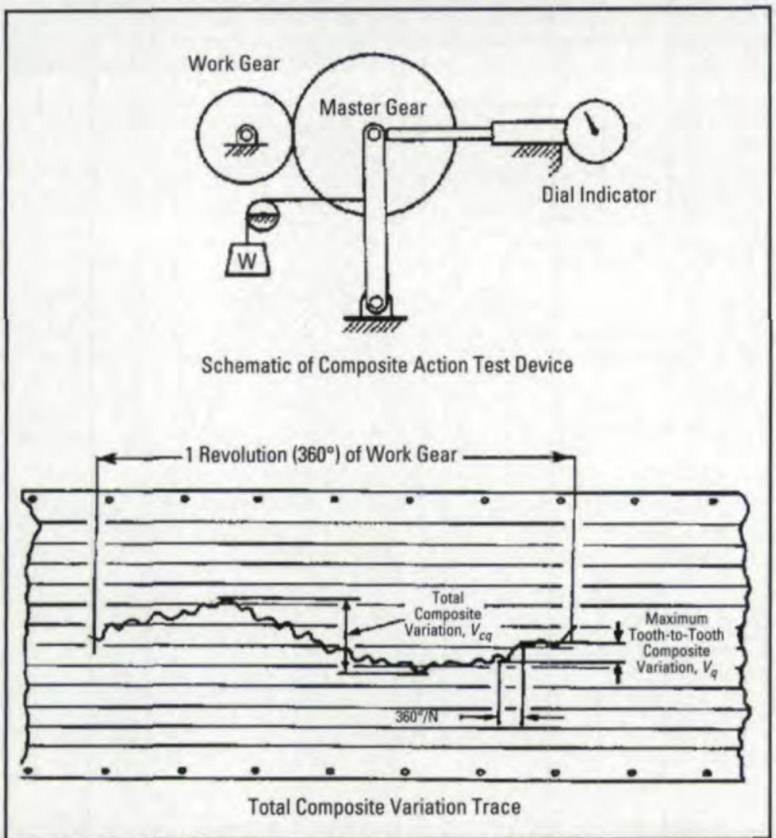

Fig. 9 — Double-flank composite action measurement.

method for finding the size of gear teeth. Refer to AGMA Standard 2002, *Tooth Thickness Specification and Measurement*, for a more detailed treatment of this subject.

Elemental vs. Composite Measurement. There are two general methods in use for the measurement of gear quality. These are the elemental and composite methods. Further, there are two different composite methods—double-flank and single-flank.

Elemental Methods. Elemental measurement involves the measurement of discrete aspects of gear quality—such as involute, tooth alignment (lead), pitch variation, runout, accumulated pitch variation and tooth size—by using a small probe to explore the individual characteristic (see Fig. 8). It is very useful for medium-size gears, but difficult to use for very fine pitch or very large diameter gears. The AGMA standard limits elemental tolerances to pitches coarser than 20 DP. However, it is possible to make elemental measurements as fine as about 100 DP with some equipment. The upper limit for diameter is generally about 40". This limit is imposed by a lack of available inspection machines, although there are a few that can go larger.

These methods are generally useful for diagnostic purposes as well as for quality determination.

Double-Flank Composite Methods. Double-flank measurements are made by rolling the test gear in tight mesh with a master gear while detecting center distance variation (see Fig. 9). The quality determination for the same gear may come out different if it is measured by elemental methods. For this reason, both methods should not be used for the same part. The maker and user should agree on the method to be used.

Advantages. This method provides a fast check that uses inexpensive equipment. It is good for the determination of runout prior to any subsequent finishing of the gear teeth, and it is excellent for the determination of functional tooth thickness.

Disadvantages. It measures a composite of all characteristics and is not as useful for diagnostics. It can tell when teeth are conjugate (very low tooth-to-tooth composite), but it is not very good at telling how non-conjugate the teeth are. It measures both sides of the teeth at the same time and can produce some strange tooth-to-tooth results. It is also possible to have some very bad involute shapes but not have this method produce large tooth-to-tooth variations.

It also measures the characteristics of gear quality in a radial direction (center distance variation). Gears don't operate this way. They operate in a tangential direction.

When gears are made by certain finishing operations, such as shaving and abrasive hobbing, it is possible to have a double-flank measurement show very little runout (maybe a few ten thousandths of an inch) when in reality it has several thousandths of an inch accumulated pitch variation. Remember that gears function tangentially. Runout is a radial measure, while accumulated pitch is a tangential measure (see Fig. 10).

Single-Flank Composite Methods. Single-flank measurements are made by rolling two test gears or a test gear and a master gear together at their proper mounting distance and with backlash. Transducers measure the angular motion and record any characteristics of nonuniform motion (see Fig. 11). This is a true tangential measurement and is indicative of the functional characteristics of the gear. This nonuniform motion is called transmission error.

Advantages. Single-flank measurement is a truly functional, tangential measure. It is good for diagnostic purposes. It is especially useful for the measure of profile shape and conjugacy as well as for accumulated pitch variation.

Disadvantages. It is not good for measuring tooth alignment (lead) and size. It is a slower test than the double-flank method. It uses equipment that is more expensive and not as readily available.

Which Inspections to Perform?

Obviously, if a print calls for certain inspections, these must be done. However, one also should think of what are the most important characteristics desired of the gears. Quietness? Positional accuracy? Strength and durability? Size? Some combination of these? The greatest measurement effort and best techniques should be applied to the most important parameters. For example, if positional accuracy is most important, a lot of time should not be spent measuring runout. It can be very misleading. See Fig. 12 for a chart of recommended measuring methods. Standards now in process with

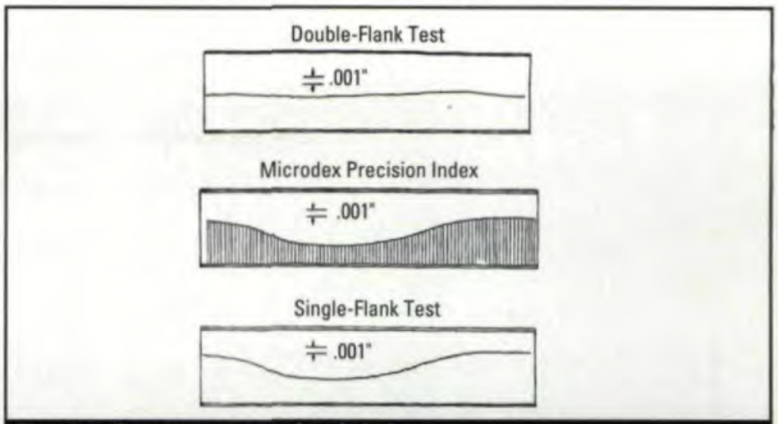

Fig. 10 — Runout vs. accumulated pitch.

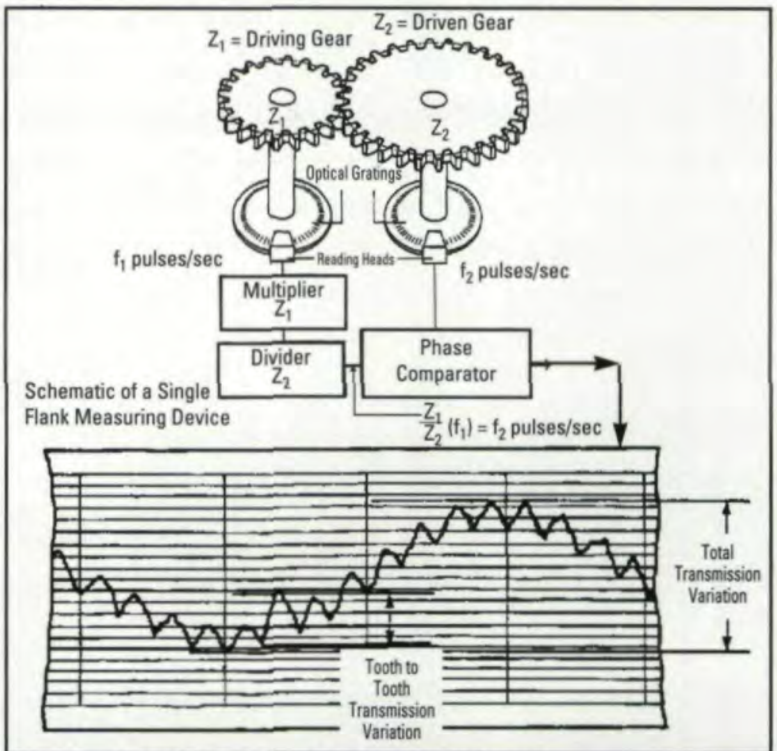

Fig. 11 — Single-flank composite action measurement.

Measurement Method Order of Preference 1, 2, 3	Important Quality Criteria				
	Noise	Accuracy	Strength or Surface Durability	Tooth Size	Finish
ELEMENTAL					
Profile	2	2	1		
Lead			1		
Pitch Variation	3	1	2		
Acc. Pitch Var.		1			
Runout		2			
SIZE					
Funct. Tooth Thick.				1	
Span				2	
Over Pins or Wire				2	
Vernier Tooth Mic.				3	
DOUBLE-FLANK COMP.				1	
Tooth-to-Tooth	3	3			
Total Comp.		3			
Funct. Tooth Thick.				1	
SINGLE-FLANK COMP.					
Tooth-to-Tooth	1	1			
Total Comp.		1			
MICROFINISH					
Roughness-Stylus					1
WAVINESS					
Contact Pattern					2
Single Flank					1

Fig. 12 — Quality criteria vs. measuring method.

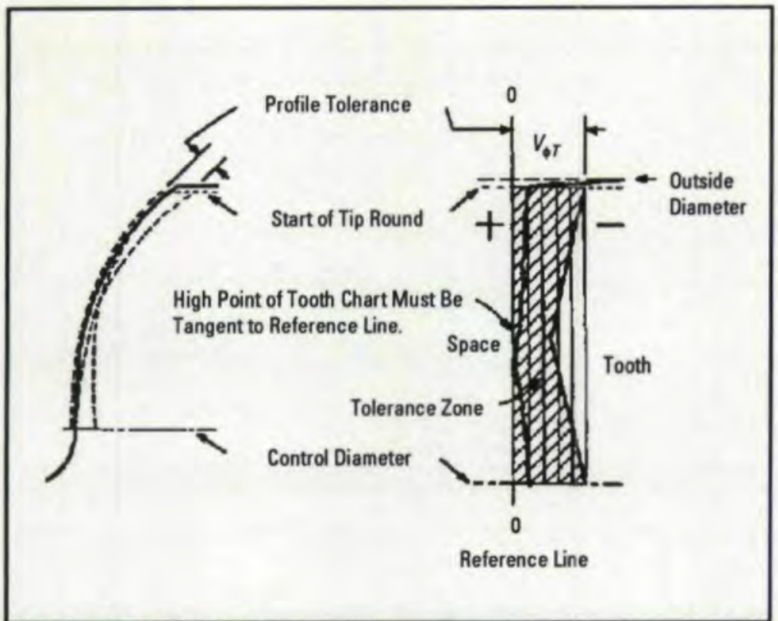

Fig. 13 — AGMA K chart.

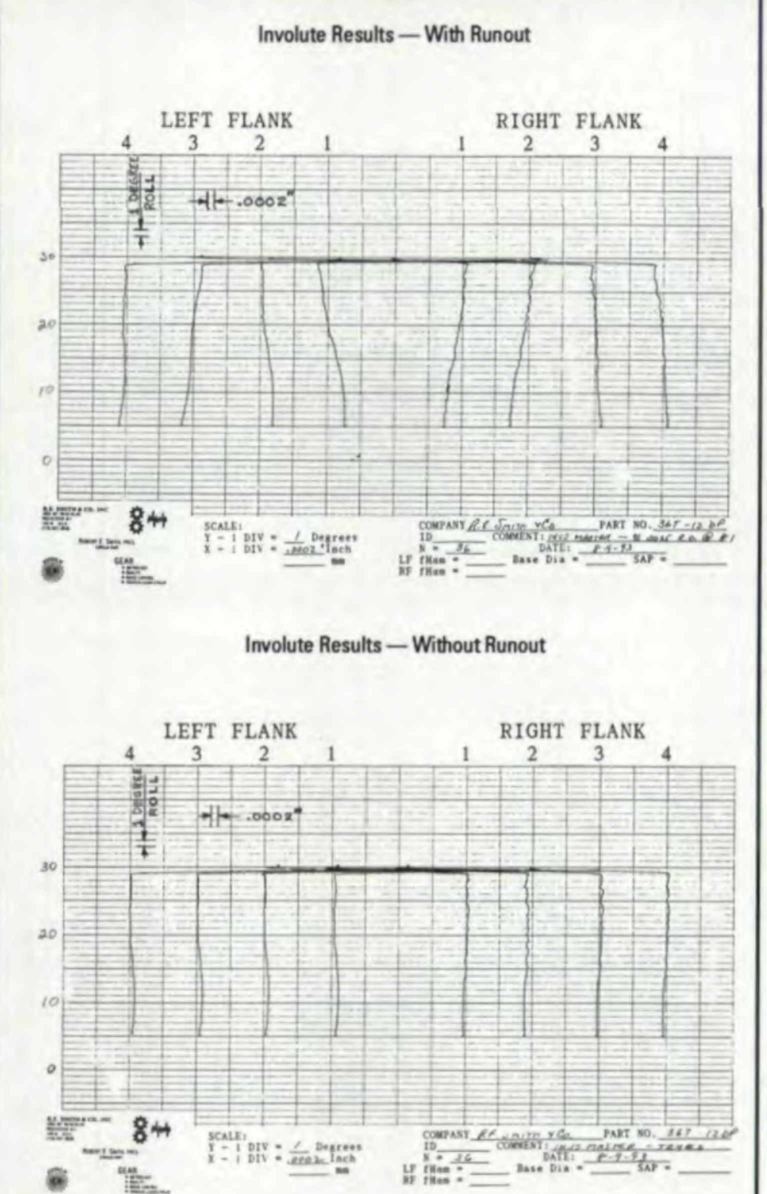

Fig. 14 — Involute results of master gear with and without runout.

the ISO accuracy committee include one that defines "families" of inspection tolerances that apply to various applications. One should put the most effort on the proper group of tolerances as it applies to the application. In other words, spend your measurement money wisely.

Question the Specifications. It is wise for the gear maker to spend some time with the gear user to arrive at the most important specifications and measuring methods. The importance of agreement on these points can be shown by example: It is not unusual for someone to specify an AGMA Q8 quality level but want to hold tooth thickness to a tolerance of .001". The problem is that the amount of runout allowed by level Q8 may not be compatible with the specified tooth thickness tolerance. It may be necessary to make a Q10 or higher gear in order to hold the tooth thickness variation within the specified tolerance. This costs more money.

Question the Standards. Many times the print will only call for an AGMA Q number. These quality levels have been established primarily for gear rating evaluation. Depending on the application, this specification will not necessarily guarantee satisfactory performance. For example: if a positional accuracy is the prime concern, this information alone will not be enough. The AGMA Q number only specifies runout and pitch variation. Accumulated pitch and pitch variation are better measures. Also, profile and lead are controlled by K charts. These K charts give a very broad control of profile and lead. They have to be broad to allow for the effects of runout on the various teeth around a gear. See Fig. 13 for a description of a K chart.

Fig. 14 shows involute charts of a master gear with and without runout. This same thing can happen with tooth alignment or lead charts. In other words, involute isn't being evaluated separately. The K charts are evaluating a combination of involute and runout. ■

Editor's Note: *The second part of this article, covering diagnostics, measuring equipment and inspection practices among U.S. competitors, will appear in the next issue.*

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VIEWPOINT

Viewpoint— Our Readers Respond

I support Clem Miller (Viewpoint, May/June) in his skepticism of ISO 9000. The metrology of gears is important, but in the present state of the art, manufacture is more accurate than design.

Having encountered situations where gears were being made accurate to 2 μm, but the design specified was inaccurate by 20 μm, it seems more important to improve the design process than to spend time and effort on further "improving" the control of production.

Dr. J. D. Smith,
Cambridge University,
Cambridge, England

"Viewpoint" provides an opportunity for readers to respond to articles that appear in our pages. Letters should be addressed 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.*

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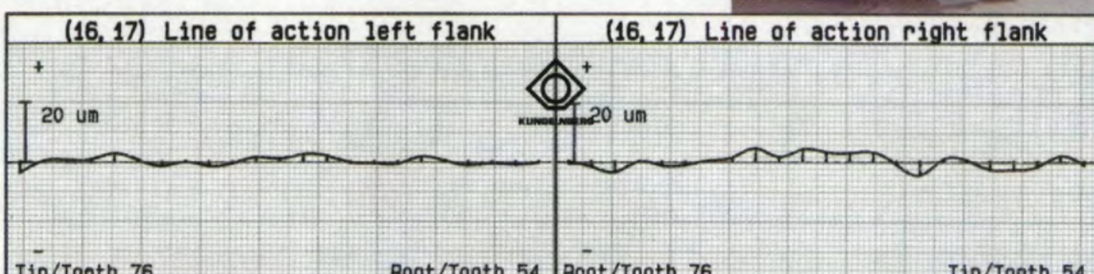

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Our Experts Discuss Electronic Gearboxes, Plus Backlash and What to Do about It

Dennis Richmond
William L. Janninck

Question: In the January/February issue of your magazine, we came across the term "electronic gearbox." We have seen this term used elsewhere as well. We understand that this EGB eliminates the change gear in the transmission line, but not how exactly this is done. Could you explain in more detail?
K. K. Patel, Coburg Powermotive Company, Gujarat, India

Dennis Richmond replies: The term "electronic gearbox" is, perhaps, a misnomer. It refers not to a particular object or collection of parts, but rather to the CNC software that automatically makes necessary adjustments in a machining operation. In a CNC machine, a programmed numerical control activates the machine's servos and spindle drives and controls the various operations.

For example, in a CNC gear grinding machine, the basic ratio between the grinding wheel and the workpiece axis is controlled by an electronic gearbox with an extremely high dynamic response. Modifications to the geometry (crowning, taper or lead) made during the machining sequence are controlled entirely by the CNC control. The advantage of this system is a uniform load condition on the machine's control system. Fluctuations are rapid but consistent. A disturbance simulator module takes advantage of this phenomenon, observing the transient load oscillations that occur through the fluctuations caused by the gear's teeth as they enter and exit the grinding worm's line of contact. The disturbance simulator module evaluates the very fast but uniform errors through a refined electronic circuit, which counteracts the minute deviations by assessing the signals and reducing the deviations by their mean values.

The transition from one load characteristic to another occurs relatively slowly; for example, asymmetric load-related errors take place when a helical gear is being ground. A one-sided load exists as the wheel enters the top face of the gear or exits the bottom face at the end of the pass. Since the disturbance simulator module corrects deviations based on the mean of a number of values received, even one-sided, load-related errors can be detected and offset by adjusting the module or diametral pitch.

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Address your gearing questions to our panel of experts. Write to them care of Shop Floor, Gear Technology, P. O. Box 1426, Elk Grove Village, IL 60009, or call our editorial staff at (708) 437-6604.

Dennis Richmond
is Sales Manager for Reishauer Corporation in Elgin, IL. He has been active in the gear industry for 22 years.

William L. Janninck
is a gear and tool design consultant. He is one of Gear Technology's *technical editors.*

Question: We manufacture electric motor traction drives for locomotives, including the driving pinion and its mating driven gear. We have experienced some problems related to the backlash in the gear set when put into final assembly. Can you examine our gear data and perhaps give us some idea what may be causing the problem? The pinion is 18 teeth and the gear is 64 teeth, 2.11667 DP (12 module), 20° PA, of spur configuration. The pinion and gear are both made of alloy steel, surface-hardened, fully ground on the entire form and operating on a center distance of 19.4094–19.4252" (493.0–493.4 mm). All the geometric gear data are included on the gear prints submitted. Also, how is backlash measured?

N. R. Krishnan, Crompton Greaves Ltd., Bombay, India

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William Janninck replies: The specific details of the backlash problem are not identified, so we will have to give some general answers.

Complaints concerning backlash in a gear set can be divided into two main categories—too little backlash and too much. Too little backlash is the most common complaint. A subcategory of this complaint is negative backlash, which causes a bind or pinch during the rotation of the set. This cannot be allowed, because it could cause severe loading on the gear teeth and the supporting structure, including the case, shafting and bearings. This can occur when the gears have excess runout, and while lash occurs at the low area of runout, a jam may occur when the gears are rotated to the high point of the runout.

In situations with too much backlash in the set, jams or binds do not occur, but noise or clatter may be introduced, and, depending on the application, room may be provided for impacting if loads are reversed. Excess lash can also be the cause of lowering the gear-tooth beam strength and reducing the gear wear life.

In investigating the first backlash problem, a study is made to see what the minimum lash in the set might be. This is done by modeling the set using the lowest allowed center distance based on the tolerance and the highest allowed gear and pinion tooth thicknesses. The resulting minimum backlash in the given set is .009" (.228 mm) by this modeling.

Modeling, however, is based on both gears in the set being perfect in all respects. If some allowance is made for the .003" (.077 mm) gear runout tolerance, plus some mounting errors, it is possible that the actual minimum lash might be reduced nearly to zero. This could only occur if the gear and pinion size were biased totally to the high limit, and the center distance were totally sitting on the low limit. Statistically this seems highly unlikely.

Since the tolerance on center distance of .016" (.40 mm) and the tolerance on tooth thickness of .009" (.228 mm) is liberal, it would be wise to get some inspection data on your production gears and boxes, especially on a set under rejection, to see where the sizes lie in the tolerance ranges and if they are biased as mentioned above,

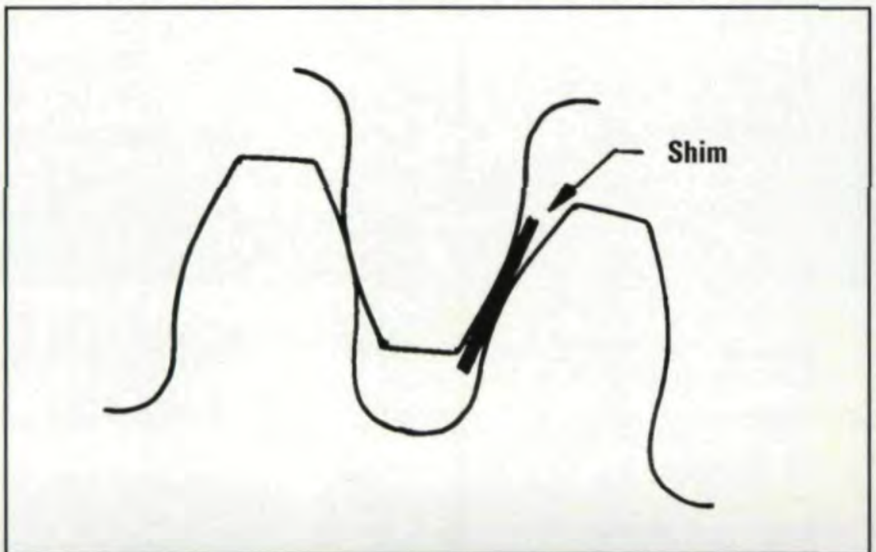

Fig. 1

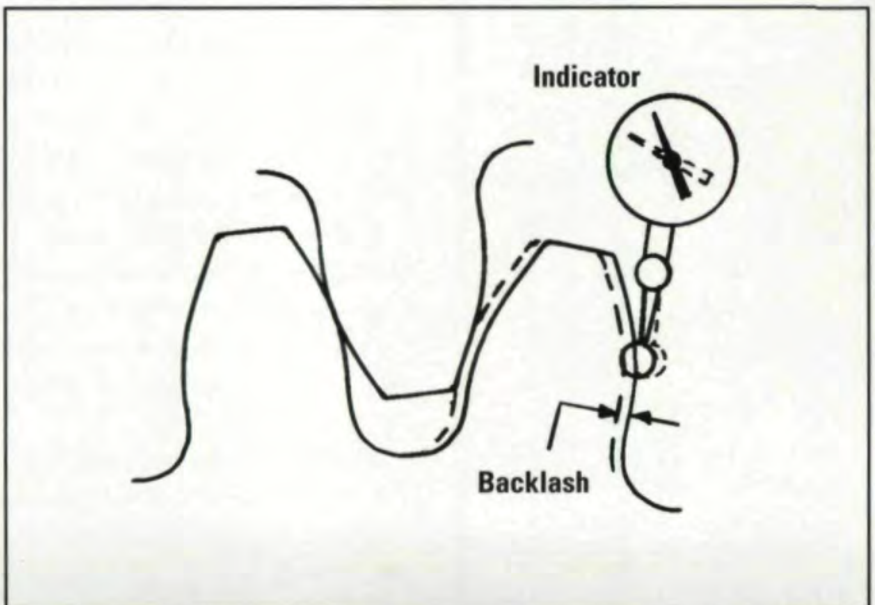

Fig. 2

or even if they are out of limits. This will surely expose the source of the problem. It is difficult to speculate without knowing anything more specific, but one possible solution is to reduce the high tooth thickness by .004–.005" (.102–.127 mm). This would give more lash allowance, and it might eliminate the problem.

By similar model computation, the maximum lash was .041" (1.04 mm). This maximum amount seems reasonable for this particular gear set.

There are several ways to measure backlash. The first and quickest is to mount up the gears and spin them through mesh, turning enough times to assure that all teeth of pinion and gear eventually engage. If they bind somewhere, then there is zero or negative lash, and the set is rejected. If they spin free, then some measure of the actual lash can be obtained by

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LASH CAN BE MEASURED BY SHIMS, FEELER GAGES OR A DIAL INDICATOR.

using shims or feeler gages (see Fig. 1). The highest and lowest shims that fit are a measure of the lash.

A small dial indicator can also be used to put a value on the lash. While the indicator is touching against a gear-tooth flank and the mate is locked on rotation, the gear is jogged back and forth, and the indicator is read for the lash with that tooth pair in mesh (see Fig. 2). As with the shims, this has to be done for a number of different mesh positions, and a high and low value will be found.

In establishing the minimum backlash, you must include allowances for the runout to be expected in each gear, as well as some provision for other factors, such as operating temperature and lubricating films. ■

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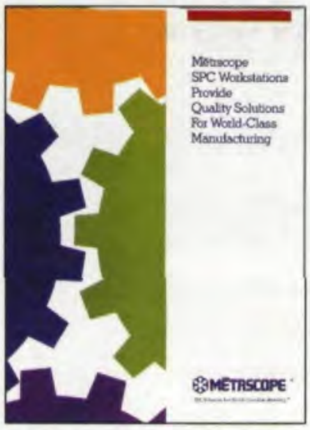

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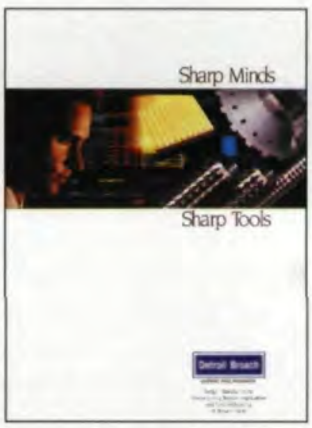

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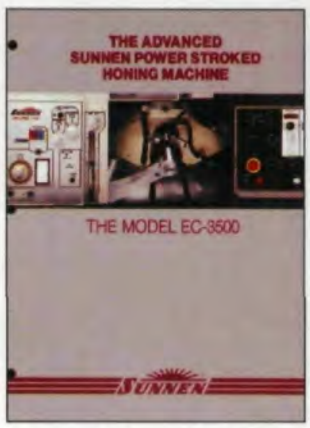

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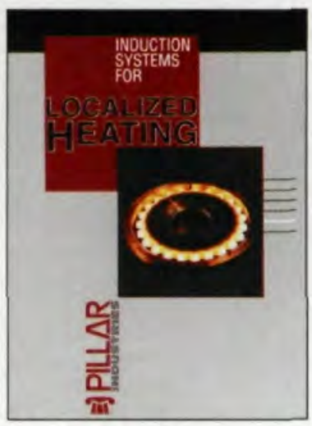

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SEPT. 14–16
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SEPT. 28
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OCT. 2-6
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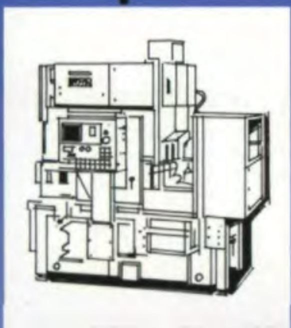

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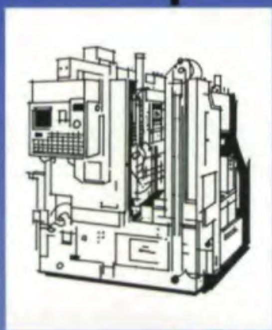

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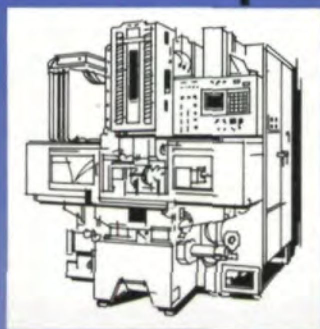

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