Gear Expo '89

GEAR TECHNOLOGY

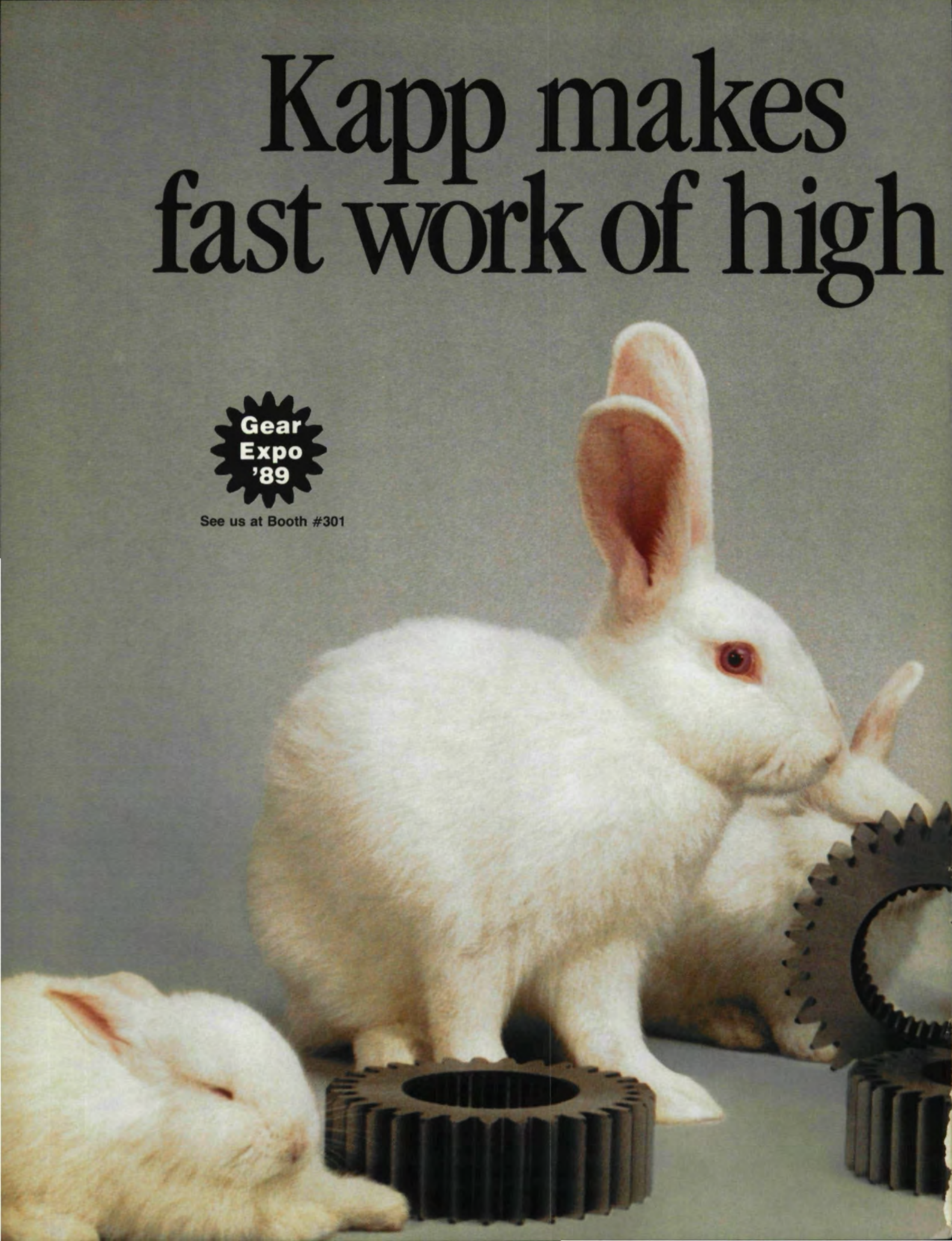
The Journal of Gear Manufacturing

NOVEMBER/DECEMBER 1989

Computer-Aided Spur Gear Tooth Design

Designing Plastic Spur and Helical Gears

Hard and Fine Finishing — Part II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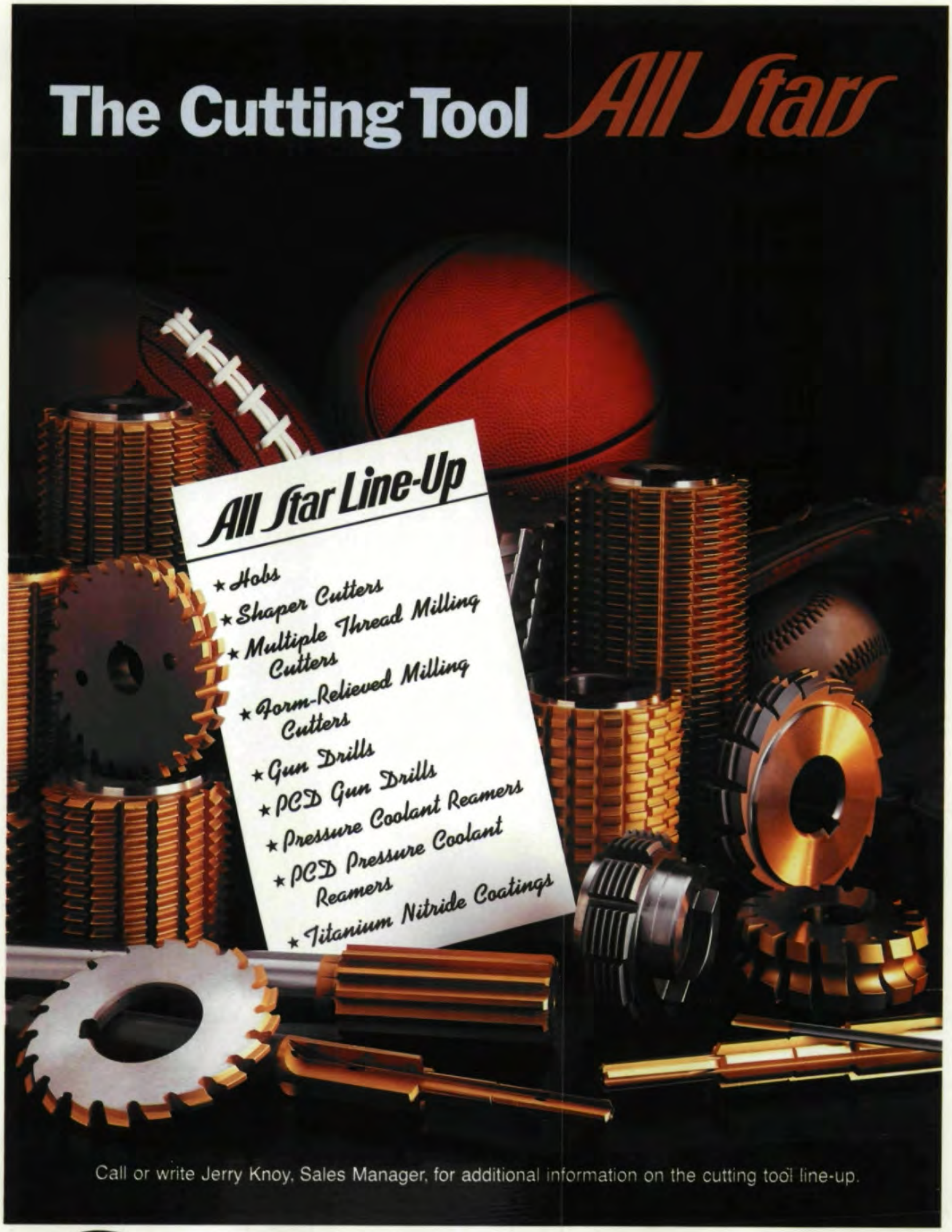
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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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MANUSCRIPTS: We are requesting technical papers with an educational emphasis for anyone having anything to do with the design, manufacture, testing or processing of gears. Subjects sought are solutions to specific problems, explanations of new technology, techniques, designs, processes, and alternative manufacturing methods. These can range from the "How to . . ." of gear cutting (BACK TO BASICS) to the most advanced technology. All manuscripts submitted will be carefully considered. However, the Publisher assumes no responsibility for the safety or return of manuscripts. Manuscripts must be accompanied by a self-addressed, self-stamped envelope, and be sent to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, P.O. Box 1426, Elk Grove, IL 60009, (708) 437-6604.

Editorial

SKILLED TRADES ARE NOT "SECOND-RATE" JOBS

The society which scorns excellence in plumbing because plumbing is a humble activity and tolerates shoddiness in philosophy because it is an exalted activity will have neither good plumbing nor good philosophy. Neither its pipes nor its theories will hold water.

John W. Gardner

The press release on my desk this morning said, "The (precision metal working) industry cannot attract enough qualified applicants. As many as 1,500 jobs a year (in the Chicago area alone) are going unfilled." So what else is new? That's just hard proof confirming the suspicion many of us have had for some time. Some of the best, most qualified and experienced people in our shops are reaching retirement age, and there's no one around to fill their spots. And, if the situation is bad in the metal working trades in general, it's even more critical in the gearing industry. Being small and highly specialized, gear manufacturing attracts even less attention and finds recruitment harder than the other precision metal trades.

Why is this? It's easy to point fingers and beat up on the education system. You know the litany by heart. "We don't do as well as the Japanese." "Our kids lack discipline and respect for hard work." "Nobody knows how to teach math and science anymore." "The government isn't spending enough money on education." But surely these are half-truths at best.

This same news release hit on something that may be closer to the mark. "Experts believe society has pushed the idea that to succeed, everyone must have a college education and has the notion the manufacturing jobs are second-class careers." This

time, at least, the experts are on to something.

If, as I do, you have late high school or college age people in your life, and you listen to them and their friends as they discuss the serious business of planning their careers, you will see some disturbing pictures emerge. Few, if any of them, even consider *not* going to college. This does not necessarily have anything to do with a burning desire for higher education on their part.

They go because, "All my friends are going," or "My parents will kill me if I don't," or "You can't earn any money if you don't go to college." They ask, "What would I do if I didn't go to college? Flip hamburgers?"

And what will they be studying at college? "I dunno. Marketing, I guess." "Business." "Pre-law, maybe."

Now, there's nothing wrong with learning marketing, business, or preparing for law school. But the fact is, the nation can use only so many marketing managers and business school grads, and it already has too many lawyers. (What are all these people going to market if there's no one around to manufacture anything?)

What is sad and disturbing about these young people is that many of them will go on to college, prepare for careers in which they are only moderately interested, perform indifferently, and graduate frankly ill-prepared to do much of anything useful. They and/or their parents will be saddled with thousands of dollars worth of debt, and the only jobs they'll qualify for are entry level office positions paying less than $20,000 per year.

These young people are not stupid or ill-disciplined, nor are they, to use the current teen-aged term, "burn-outs," dabbling in drugs or in trouble with the law, and with no use for or interest in honest work. They're bright, eager, hard-working young men and women who are being pressured by their peers, their parents, and their school guidance counselors to think that aspiring to anything less than a college education is to be a "failure," and that any job that doesn't require a college diploma is "second-rate".

This is a dangerous set of misperceptions. It undervalues many important jobs in our society and overvalues others and, in the process, skews in a negative way the number of people available to do the jobs society needs done.

Don't misunderstand me. I believe in higher education. Its value has been the subject of many of my editorials over the last six years. A college education provides opportunities that should be available to everyone who wants to take advantage of them. But it's not the only game in town. Contrary to what we, with the best of intentions, may have taught our children, not everyone has to go to college to have either a decent sense of self-worth or a decent job with a good salary and a chance for advancement.

College is a particular kind of higher education that has its uses, but it can't teach everyone everything. Not every

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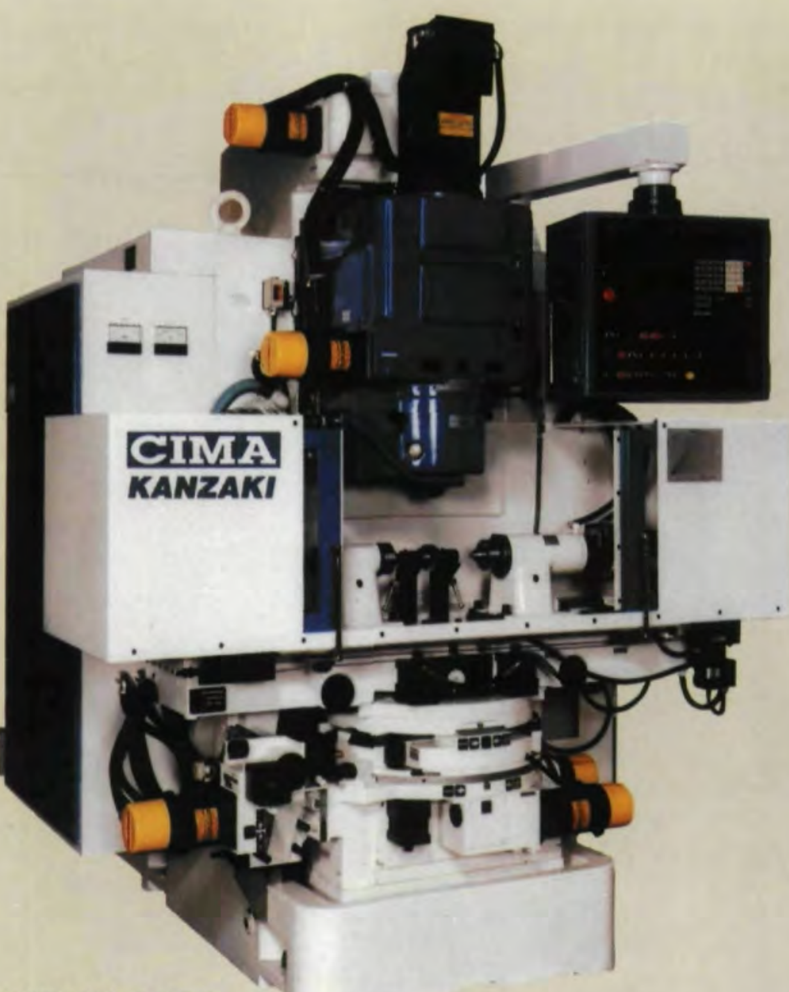

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CIRCLE A-4 ON READER REPLY CARD

TECHNICAL CALENDAR

NOVEMBER 6-8, 1989. AGMA Gear Expo '89, David Lawrence Convention Center, Pittsburgh, PA. Exhibition of gear machine tools, supplies, accessories and gear products. For more information, contact: Wendy Peidl, AGMA, 1500 King Street, Suite 201, Alexandria, VA, 22314. (703) 684-0211.

NOVEMBER 7-9, 1989, AGMA Fall Technical Meeting, Pittsburgh, PA. Seminars on a variety of gearing subjects held in conjunction with Gear Expo '89.

NOVEMBER 29 - DECEMBER 1. Fundamentals of Gear Design. Seminar, University of Wisconsin-Milwaukee. This course will cover basic design considerations in the development of a properly functioning gear system. It is planned with the designer, user, and beginning gear technologist in mind. For more information, contact: Richard G. Albers, Center for Continuing Engineering Education, University of Wisconsin-Milwaukee, 929 North 6th Street, Milwaukee, WI, 53203. Ph: (414) 227-3125.

The **International Trade Commission** is holding public hearings for the purpose of assessing the competitive position of the U.S. gear industry in U.S. and global markets. These hearings will be held on **November 1, 1989, at 9:30 a.m.** If you wish to be heard or submit a written statement, write to the Secretary U.S. International Trade Commission, 500 E. Street, S.W., Washington, D.C., 10436.

AGMA Technical Education Seminars, Series II. These one- or two-day seminars present the latest techniques and information on specific topics in a small group context. For more information regarding fees and registration contact AGMA, 1500 King St., Suite 201, Alexandria, VA 22314, (703) 684-0211.

December 5-6, 1989, Los Angeles, CA
Loose Gear Inspection from the Customer's Standpoint.
January 23, 1990, Cincinnati, OH
Gear Failure Analysis
March 6, 1990, Cincinnati, OH
Rational Gear Design and Lubrication
May 8, 1990, Los Angeles, CA
Controlling the Carburizing Process
June 5-6, 1990, Alexandria, VA
Gear System Design for Noise Control

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CIRCLE A-6 ON READER REPLY CARD

Guest Editorial

MISSION: COMPETING TO WIN

Like a lot of people, I grew up seeing the world as fairly flat and believing that everything of importance happened in Texas. As I grew older, my outlook grew to include the United States, Canada, and Mexico. The rest of the world did not seem very important, if it existed at all. Unfortunately, I was not alone in this very narrow view. Many others in the gear business shared this perception.

Today the world is a bigger and meaner place than it appeared when I was growing up. In traveling to other countries for my company and AGMA, I've seen our foreign competition in both the marketplace and their factories. Our European and Asian competitors are tough. They've done their homework, most of it in our own back yard.

These foreign companies have grown to be true competitors in every market. We see their footprints everywhere, as foreign shipments into the U.S. and Canada continue to rise. In some cases, these companies have decided to build, buy, or expand plants in our country. We now have to compete with them toe-to-toe or we stand a good chance of "losing the farm."

AGMA's mission is to assist its members to compete more effectively in today's global market. The Association is doing this in several ways. AGMA's reports keep us better informed about international economics and market trends that have a major impact on our members. Through standards-writing activities, we bring the best technical know-how to both national and international standards. The Fall Technical Meeting, the Gear Manufacturing Symposium, and the Technical Education Seminars help pass along this manufacturing expertise to the gear producers. The *AGMA Manufacturer's Self-Certification Program* enables us to upgrade our skills and helps assure customers that the best gear expertise in the world is put to work on their projects.

The week of meetings this year is Pittsburgh is where a lot of those efforts come together. The Fall Technical Meeting has grown to a session that truly has an international reputation for excellence. At this meeting, the best design and production gear engineers from around the world share their expertise.

JAMES R. PARTRIDGE, *current AGMA president, is Vice President of Lufkin Industries. He is a registered professional engineer and a member of the Texas Society of Professional Engineers. He is currently a Director of AGMA and has served as Treasurer and Senior Vice President of the association. Mr. Partridge is also an active member of the American Petroleum Institute and the Society of Petroleum Engineers. He is a graduate of Texas A&M University.*

Just as important is what we can learn on the floor of the Gear Expo in November. This event is not only the largest show dedicated to just the gear industry – it is also the best! There is more of the latest gear machinery and manufacturing knowledge per square foot at the exposition than anywhere else in the world.

Together these two important events in Pittsburgh offer us the chance to see "The Cutting Edge" of our technology – in design, application, and manufacturing. If you are ready to work in the competitive world of today, Gear Expo '89 is the place to obtain the necessary tools.

We can't expect someone else to shield our hides from the competition we face in the market. To furnish the best product and services at a competitive price, we must have the tools to compete, help set the rules of the game, play at our best, and keep track of the score. In Pittsburgh, we have the chance to "get down to it" and individually learn the best way to play to win.

The alternative is not acceptable!

James R. Partidge,
President, AGMA

Computer-Aided Spur Gear Tooth Design: An Application-Driven Approach

H. Zarefar
Portland State University, Portland, OR
T.J. Lawley
University of Texas at Arlington, Arlington, TX

Abstract:

This article discusses an application-driven approach to the computer-aided sizing of spur gear teeth. The methodology is based on the index of tooth loading and the environment of application of the gear. It employs handbook knowledge and empirical information to facilitate the design process for a novice. Results show that the approach is in agreement with the textbook data. However, this technique requires less expert knowledge to arrive at the conclusion. The methodology has been successfully implemented as a gear tooth sizing module of a parallel axis gear drive expert system.

AUTHORS:

DR. HORMOZ ZAREFAR *is Assistant Professor of Mechanical Engineering at Portland State University, where he teaches mechanical design and does research in the area of automation of the design of mechanical components. He received his Ph.D. in mechanical engineering from the University of Texas at Arlington, where he received the Carl W. Files Outstanding Teaching Associate Award. Dr. Zarefar is a member of ASME, ASEE, Sigma Xi (Scientific Research Society) and Pi Tau Sigma (Mechanical Engineering Scholastic Honor Society).*

DR. T.J. LAWLEY *has been a professor of mechanical engineering design since 1971. Before that he was employed as a mechanical engineering design engineer in industry for nine years. He received his B.A. and B.S.M.E. degrees from Rice University and his M.S.M.E. and Ph.D. degrees from Southern Methodist University. He is a member of ASME and a registered professional engineer.*

Introduction

The science of gear design is one that has been thoroughly investigated and much written about.[1,2] In transmission of motion via gears, two or more axes can be arranged in just about any orientation. One of the most common arrangements of input/output transmissions is the parallel axis type, in which the motion is transmitted from an input axis to a parallel output shaft. There are two widely used gear tooth configurations to achieve the parallel axis arrangement; namely, spur and helical gear tooth designations. Spur gear drives are the most economical way of transmission of motion between parallel axes.

Recently, with extensive utilization of digital computers in engineering design and analysis, interest has been shifted toward automation of gear design. Several attempts have been made in which various techniques for routine gear tooth design have been computerized.[1,4] However, these attempts have all been initiated by expert gear designers and have been based on the individual's own methodology for the gear tooth design process.

In this article a new technique for computer-aided sizing of the spur gear tooth is discussed. The approach is based on the requirements of the application environment of the gear, while the typical textbook approach [3] requires that the user be proficient in making key assumptions in the design process. Although limited to spur gears, a similar approach can be employed for the design of helical and other types of gear teeth as well.

The Approach

In general, design of a typical gearbox is influenced by the following factors:

Table 1

Application	Minimum hardness of steel gears: Pinion	Minimum hardness of steel gears: Gear	No. pinion cycles	Accuracy	K factor: N/mm^2	K factor: psi	Unit load: N/mm^2	Unit load: psi
Turbine driving a generator	225 HB 335 HB 59 HRC	210 HB 300 HB 58 HRC	10^{10} 10^{10} 10^{10}	High precision High precision High precision	0.69 1.04 2.76	100 150 400	45 59 83	6,500 8,500 12,000
Internal combustion engine driving a compressor	225 HB 335 HB 58 HRC	210 HB 300 HB 58 HRC	10^{9} 10^{9} 10^{9}	High precision High precision High precision	0.48 0.76 2.07	70 110 300	31 38 55	4,500 5,500 8,000
General-purpose industrial drives, helical (relatively uniform torque for both driving and driven units)	225 HB 335 HB 58 HRC	210 HB 300 HB 58 HRC	10^{8} 10^{8} 10^{8}	Medium high precision Medium high precision Medium high precision	1.38 2.07 5.52	200 300 800	38 48 69	5,500 7,000 10,000
Large industrial drives, spur— hoists, kilns, mills (moderate shock in driven units)	225 HB 335 HB 58 HRC	210 HB 300 HB 58 HRC	10^{8} 10^{8} 10^{8}	Medium precision Medium precision Medium precision	0.83 1.24 3.45	120 180 500	24 31 41	3,500 4,500 6,000
Aerospace, helical (single pair)	60 HRC	60 HRC	10^{9}	High precision	5.86	850	117	17,000
Aerospace, spur (epicyclic)	60 HRC	60 HRC	10^{9}	High precision	4.14	600	76	11,000
Vehicle transmission, helical	59 HRC	59 HRC	4×10^{7}	Medium high precision	6.20	900	124	18,000
Vehicle final drive, spur	59 HRC	59 HRC	4×10^{6}	Medium high precision	8.96	1300	124	18,000
Small commercial (pitch-line speed less than 5 m/s)	320 HB 320 HB	Phenolic laminate Nylon	4×10^{7} 10^{7}	Medium precision Medium precision	0.34 0.24	50 35	— —	— —
Small gadget (pitch-line speed less than 2.5 m/s)	200 HB 200 HB	Zinc alloy Brass or aluminum	10^{6} 10^{6}	Medium precision Medium precision	0.10 0.10	15 15	— —	— —

- Spatial arrangement of the input/output shafts,
- Input/output speed ratio,
- Power transmitted (torque or horsepower/speed combination).

The proposed spur gear tooth designer is an application-driven computer program in which the user identifies the nature of components, machines, or assemblies where the gear is to be used. It is essentially based on the index of tooth loading or K-factor.[1] This index is a basis for initial gear tooth design; hence, the preliminary stage of the design starts with selecting an initial K-factor from the user-defined application environment. Based upon the specified value of K-factor, a Brinell hardness number can be established (Table 1). From a table of Bhn versus tooth numbers, a preliminary estimate of the tooth number is established (Table 2). This step in turn leads to determination of a trial value for the pinion pitch diameter (d).

After determining the pitch diameter, a preliminary sizing of the gear can be undertaken by means of the Q-factor method.[1] The Q-factor method is one whereby the power, speed, and gear train ratio of the gear set are all combined into a single nondimensional value (Q). This number is representative of the "size" of the load the gear has to support. The Q-factor is defined by the formula:[1]

Table 2

Ratio u (m_G)	Long-life, high-speed gears: Brinell hardness 200	300	400	600	Vehicle gears, short life at maximum torque: Brinell hardness 200	300	400	600
1	80	50	39	35	50	37	29	26
1.5	67	45	32	30	45	30	24	22
2	60	42	28	27	42	27	21	20
3	53	37	25	25	37	24	18	18
4	49	34	24	24	34	23	17	17
5	47	32	23	23	32	22	17	17
7	45	31	22	22	31	21	16	16
10	43	30	21	21	30	20	16	16

$$Q = \frac{HP * (m + 1)}{n * m}$$

where

m = gear reduction (speed ratio)
HP = horsepower
n = pinion rpm

Note that by knowing the speed ratio, the initial value of the mating gear pitch diameter (D) can also be determined, and the pinion-gear center distance (C) can be evaluated. The face width can now be calculated using the formula: [1,2]

$$F = \frac{31500 * Q}{K * C^2}$$

where

C = center distance (inches)
F = face width (inches)
K = K-factor
Q = Q-factor

At this stage, the face width is checked

(continued on page 40)

Systematic Approach to Designing Plastic Spur and Helical Gears

Raymond M. Paquet
Plastics Gearing Technology, Inc.
Manchester, CT

Abstract:

Plastic gears are being used increasingly in applications, such as printers, cameras, small household appliances, small power tools, instruments, timers, counters and various other products. Because of the many variables involved, an engineer who designs gear trains on an occasional basis may find the design process to be somewhat overwhelming. This article outlines a systematic design approach for developing injection molded plastic spur and helical gears. The use of a computer program for designing plastic gears is introduced as an invaluable design tool for solving complex gearing equations.

Introduction

As one of man's oldest mechanical devices, gears have been in use for thousands of years. It is no surprise that design techniques, manufacturing procedures and standards have been established based on the commonly available materials of the time, namely wood and metal. When plastic materials were first used as gears in the 1950's and 1960's, it was common to design them using existing AGMA standards for metal gears. For example, if a designer was considering trying plastic gears for one of his applications, he would usually take an existing product drawing specifying steel, change the material to nylon and leave the gear data intact. There were no standards at the time to guide the designer to do otherwise. It was also economical to consider cutting plastic gears with the available stock hobs and cutters.

Today, there are still no official "standards" available, but some tooth profiles have been developed especially for plastic gears and used successfully in industry. If the injection molding process is utilized, then the designer is free from the constraints of using stock hobs and cutters. Since special hobs are required regardless, (to compensate for mold shrinkages and EDM spark gap), the designer might as well incorporate additional modifications, such as full fillet radius, to optimize the strength of the gear teeth.

The modifications to the gear teeth mentioned in this article are not new to the gearing industry. Engineers familiar with gear design will recognize that the recommended modifications for strengthening the plastic gear teeth are the same ones specified for heavily loaded metal gears in critical applications.

In summary, this article describes the manufacturing process used to produce plastic gears, and explains inspection procedures.

Design of Plastic Gears

A designer must consider the pros and cons of plastic materials before choosing it as a gear material. The advantages are

- relative low cost
- resistance to corrosion
- reduction in weight (about 15% the weight of steel)
- low inerita
- self-lubrication capacity
- potential for noise reduction
- design freedom to obtain additional features in one product such as posts, cams, cluster gears, trunnions, palls, sprockets, metal inserts and spring arms
- color coding capability for fast, error-free assembly.

The disadvantages include

- lower strength
- greater thermal expansion/contraction
- limited heat resistance
- size change with moisture absorption.

COMMON PLASTIC MATERIALS FOR GEARS. Even though many new materials have been introduced in the past decade, the majority of gear applications still call for acetals and nylons. In special cases, polycarbonates, thermoplastic polyesters and thermoplastic polyurethanes can also be considered.[1,5]

Acetal: Strong, stiff plastic with exceptional dimensional stability due to low moisture absorption, resistance to creep and vibration fatigue; low coefficient of friction; high resistance to abrasion and chemicals; retains most properties when immersed in hot water; low tendency to stress-crack.

Nylon: Family of resins having outstanding toughness and wear resistance, low coefficient of friction and excellent electrical properties and chemical resistance. Resins are hygroscopic; dimensional stability is poorer than that of most other plastics.

Polycarbonate: Highest impact resistance of any rigid plastic; excellent stability and resistance to creep under load; fair chemical resistance; stress cracks in hydrocarbons; usually used with addition of glass fiber einforcement and PTFE lubricant.

Thermoplastic polyester: Excellent dimensional stability, electrical properties, toughness and chemical resistance, except to strong acids or bases; notch sensitive; not suitable for outdoor use or for service in hot water. Material is relatively soft and has the potential for tooth damage.

Thermoplastic polyurethane: Tough, extremely abrasive and

impact resistant material; good electrical properties and chemical resistance. Difficult to injection mold small parts due to the material's elastic properties.
Polyester elastomer: Sound dampening; resistance to flex-fatigue and impact.

All of the above base materials can be formulated with fillers, such as glass fibers, for added strength, and PTFE, silicone and molybdenum disulphide for added lubricity.

ESTABLISHING DESIGN GOALS. Arriving at the specifications for the best possible gears given an application is a time-consuming operation when considering the following variables.

- Load carrying capacity. Strength equations are used to avoid tooth fracture at the root. They are derived from the Lewis equations for beam strength and have been modified for plastic gear teeth.
- Smoothness. The gear mesh should be designed so that binding will not occur at extreme environmental conditions.
- Quiet continuity of action. Ensure that the contact ratio is greater than 1.0, preferably 1.2, or simply stated, that at least one tooth is in contact at all times.
- Cost. If the gears are to be molded in plastic, the task becomes even more difficult to handle, especially since there are literally hundreds of formulations from which to choose. In addition, the use of non-standard hobs allows the design engineer to consider a larger choice of tooth forms, such as the six listed in Appendix A. Because of all the options open much of the engineer's work must necessarily be a process of substitution and elimination. This process involves numerous calculations, some of them quite complex.

SPUR AND HELICAL GEAR DESIGN – A SYSTEMATIC APPROACH.

Study of design parameters. The gear train design engineer is first required to make a study of all the information about the application available to him, such as the power to be transmitted, the ratio required, the speeds involved, the environment in which the gears will operate, the life expectancy and the space in which the gears are to be housed.

The designer then proceeds to make preliminary choices of numbers of teeth, diametral pitch, tooth form, type of material to be used and spur or helical design.

Determine load carrying capacity. The engineer now possesses all the information required to determine what power the gears can safely transmit. If the answer is too low, the effect of a coarser diametral pitch, a stronger tooth form, an increased face width, or plastic having higher allowable design stresses, singly or in any desired combination, must be investigated. The following is an example of an equation that can be used to calculate load capacity.[1]

$$HP = \frac{D \times F \times n \times J \times St \times K_T \times K_L}{126{,}000 \times P \times C_s \times K_R}$$

where:

HP = horsepower
D = operating pitch diameter
F = effective face width
n = speed, rpm
J = geometry factor
St = tensile strength of plastic
K_T = temperature factor
K_L = life factor
P = diametral pitch, or normal diametral pitch
C_s = service factor
K_R = factor of safety

If the calculations indicate that the gears would be pushed to their limits, developing prototype parts to perform actual life tests would be advisable.

Options available to optimize a gear mesh. In order to get the most out of the relatively low tensile strength of plastics, the tooth geometry can be altered several ways to be as strong as possible.

- Full fillet radius modification. The tooth form shown in Fig. 1 has a full fillet radius at the root to eliminate sharp corners that could result in high stress points. This modification can add up to 20% to the tooth strength. Teeth of heavily loaded metal gears are also designed with this feature.
- Tip relief modification. Tip relief starting halfway up the addendum should be another modification specified for plastic gears. When a tooth deflects under load, it can get in the way of the oncoming tooth of the mating gear. This can cause noise, wear and loss of uniform action. Tip relief is added to improve this situation. (See Fig. 2.)
- Elimination of undercut condition. If a standard gear has a small number of teeth, the lower portion on the tooth will be undercut as illustrated on the left in Fig. 3. This condition is undesirable in that it weakens the tooth, causes wear and inhibits continuity of action.

To eliminate undercutting, the circular tooth thickness should be increased over standard as shown in Fig. 3 on the right.

- Balance circular tooth thickness. If the gear teeth in a mesh are designed at standard, they will be the "weak link" in a power drive. The pinion tooth illustrated in Fig. 4 clearly shows the problem. (All dimensions are in inches.)

To design the gear teeth for equal beam strength, it is essential that the thickness at the root be equal, as illustrated in Fig. 5.

With the use of available formulas, the tooth thickness of the pinion was increased from a standard of 0.0491 to 0.0640, and the gear tooth thickness was decreased from 0.0491 to 0.0480 to get an equal thickness of 0.0660 at the root.

Expansion and error considerations. Once the tooth thickness of each gear has been established, the next step would be the calculation required to determine the increase in center distance above standard to cater to all the factors that could otherwise cause the gears to bind in operation. The designer considers the following equation:

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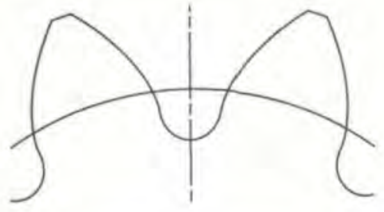

Fig. 1

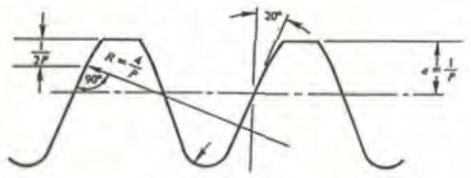

Fig. 2

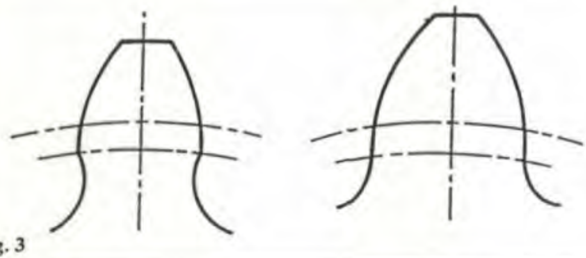

Fig. 3

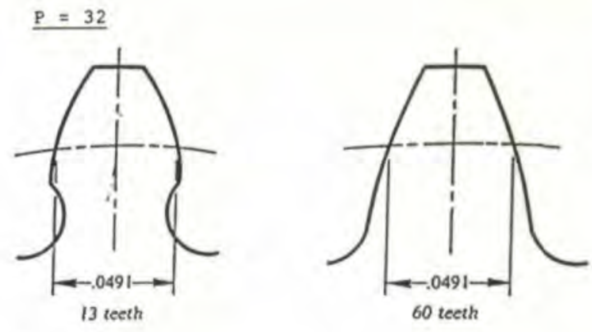

Fig. 4

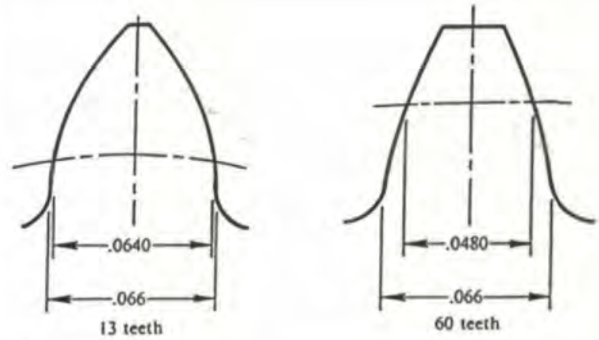

Fig. 5

$$\Delta_c = \frac{TCT_1 + TCT_2}{2} + C\left[(T-70)\left(\frac{COEF_1 \times N_1}{N_1+N_2} + \frac{COEF_2 \times N_2}{N_1+N_2} - COEF_H\right) + \left(\frac{M_1 \times N_1}{N_1+N_2} + \frac{M_2 \times N_2}{N_1+N_2} - M_H\right)\right] + \frac{TIR_1 + TIR_2}{2}$$

where:

Δ_c = required increase in center distance
TCT_1 = maximum total composite tolerance of 1st gear
TCT_2 = maximum total composite tolerance of 2nd gear
C = close mesh center distance
T = maximum temperature to which gear will be subjected (°F)
$COEF_1$ = coefficient of linear thermal expansion of material of 1st gear (in/in/°F)
$COEF_2$ = coefficient of linear thermal expansion of material of 2nd gear (in/in/°F)
$COEF_H$ = coefficient of linear thermal expansion of material of housing (in/in/°F)
N_1 = number of teeth in 1st gear
N_2 = number of teeth in 2nd gear
M_1 = expansion due to moisture pick-up of material of 1st gear (in/in)
M_2 = expansion due to moisture pick-up of material of 2nd gear (in/in)
M_H = expansion due to moisture pick-up of material of the housing (in/in)
TIR_1 = maximum allowable runout of bearing of 1st gear
TIR_2 = maximum allowable runout of bearing of 2nd gear

The calculations enable the designer to specify tooth thicknesses and operational center distance.[1]

Calculate contact ratio. If one of the mating spur gears has a relatively small number of teeth, it is essential to check the pair's contact ratio, particularly if the gears are plastic. Plastics, in general, have high coefficients of linear thermal expansion. When the expansion differential between the gears and the housing in which they are mounted is great, and when the maximum environmental temperature is high, the center distance requires a significant amount of increase to prevent binding at those high temperatures. However when the same gears later contract at the lower temperatures, they may be out of mesh to an extent that the contact ratio becomes less than adquate.

One option available to get around this problem is to use a tooth form with longer teeth. (See Appendix A). For continuity of action the contact ratio must never be less than 1.0. It should preferably be at least 1.20.

The variables required to calculate the minimum contact ratio are the numbers of teeth for each gear, the diametral pitch, the minimum outside diameters, and the maximum operating center distance. If it is less than adequate, the designer can review the following available alternatives:

- tooth forms with longer teeth,
- non-standard diametral pitches ($P = 48.5526$, for example),
- materials with lower coefficients of linear thermal expansion,
- higher quality gears,
- tighter center distance tolerances,
- changes in the numbers of teeth in the gears.

Establish Drawing Specifications and Tolerances. The final step in designing a gear is preparation of a drawing which lists the gear data. It must be specified so that there can be no possibility of misinterpretation. This may seem obvious, but ambiguity in writing gear specifications has often resulted in costly and time-consuming changes to expensive molding dies. Appendix E illustrates the data that should appear on a drawing of a spur gear.

Gear Design Example

In this example we will suppose that the design engineer is assigned a project to work on a gear mesh for a special type of chart recorder.

Where does the engineer begin?

What procedure should be followed?

How should the analysis be done?

For this project the designer has decided to use the step by step approach described so far. Correctly used, the system will provide a design for gears having an adequate contact ratio, tooth profiles with no undercut, and sufficient clearance to insure that they will not bind when subjected to the extremes of the environment in which they will operate. Furthermore, the design will be presented in such a form that there can be no misinterpretation of the engineer's requirements, either by the molder of the gear or the inspector responsible for insuring that the requirements have been met.

In other hands this system could well produce designs that would differ, but be just as valid.

The following constraints were outlined by the manager:

- costs to be kept as low as possible
- material of housing to be polycarbonate with 20% glass fiber
- the gears will be subjected to a maximum operating temperature of 130° Fahrenheit (F)
- ratio required is 2 to 1
- bearings must have a maximum allowable runout of 0.001 inches
- load is light; material strength is not a factor.
- life requirement is 260 hours.
- input torque is 4 inch-ounces.
- input speed is 50 revolutions per minute (rpm).

Begin Analysis

(All dimensions are in inches.)

Step 1. Study of design parameters. Some preliminary assumptions need to be made to begin the analysis: 1) material of pinion and gear to be unfilled acetal; 2) spur gears will be used; 3) gears to have an accuracy equivalent to AGMA Quality No. Q6; 4) start with the PGT-1 tooth form to take advantage of the full fillet radius and the tip relief.

Step 2. Settle upon the numbers of teeth and diametral pitch. Given the spacing of the housing and the required ratio, the following have been chosen:

	PINION	GEAR
Number of Teeth	15	30
Diametral Pitch	48	48

Step 3. Determine the load carrying capacity. Although the loads are light, the designer performs the calculation to determine the load capacity. The required tensile strength of the material was calculated at 1,900 pounds per square inch (psi). Therefore, the unfilled acetal with a tensile strength rating of approximately 9,000 psi is a good choice.

Step 4. Determine the minimum circular tooth thickness (CTT) at the pitch diameter of the pinion to avoid undercut. The calculated minimum CTT = 0.0353. Assign a +0.000, −0.001 tolerance to the pinion CTT.

The preliminary data established so far are shown in Table 1.

Table 1

	PINION	GEAR
Number of Teeth	15	30
Diametral Pitch	48	48
Pressure Angle	20°	20°
PGT Tooth Form	PGT-1	PGT-1
Stand. Pitch Circle Dia.	0.3125	0.6250
Outside Diameter	0.3640	0.6620
	0.3610	0.6580
Circular Tooth Thickness		
(at pitch circular dia.)	0.0363	0.0310
	0.0353	0.0300
AGMA Quality No.	Q6	Q6
Max Total Composite Tol	0.0036	0.0036
Max Tooth-To-Tooth Tol	0.0022	0.0020

Step 5. Expansion and error considerations (to determine the operating center distance). If perfect gears could be manufactured, the operating center distance would be 0.4712. However, the thermal expansion of the gears, gear runout and bearing runout must be considered to prevent the mesh from binding. The designer has calculated that the center distance must be separated by 0.0061 to allow for the possible variations in the system. Assuming a tolerance of +0.003, −0.000, the operating center distance (OCD) is specified at 0.4773 − 0.4803.

Step 6. Calculate the contact ratio (CR) for the established values. The following six input variables are used to calculate the contact ratio.

number of teeth	N1	15
number of teeth	N2	30
diametral pitch	P	48
outside diameter	Do1	0.3610"
outside diameter	Do2	0.6580"
operating center distance	OCD	0.4803"
contact ratio	CR	1.01

A contact ratio of 1.01 is too low. The designer must now attempt to raise it to 1.20. He knows that the molding department will have no trouble if AGMA Quality No. 7 gears were specified instead of No. 6, so he decides to find out what improvement this change might effect. The maximum total composite tolerance is now .0026 for both the pinion and gear. As a consequence, the maximum operating center distance is reduced from 0.4803 to 0.4793.

operating center distance	OCD	0.4793"
contact ratio	CR	1.05

He now tries the effect of reducing the operating center distance tolerance from +0.003, −0.000 to +0.001, −0.000.

operating center distance	OCD	0.4773"
contact ratio	CR	1.13

He tries even more accurate gears – AGMA Quality No. 8, having maximum total composite tolerances of 0.0018.

operating center distance	OCD	0.4765"
contact ratio	CR	1.16

At this point, he decides to return to the beginning and determine what contact ratio he can achieve by switching the plastic he had first chosen from an acetal having a coefficient of linear thermal expansion of 6.8 x 10-5 in./in./deg. F, to a glass-filled variety having a coefficient of 1.5 x 10-5 in./in./deg. F. The operating center distance is now 0.4845-0.4855.

operating center distance	OCD	0.4750"
contact	CR	1.23

The designer has reached his goal, but only by switching to a more costly plastic, and has reduced the center distance tolerance to less than is desirable.

But he still has another option open. He can use a tooth form having a longer addendum. One such tooth form used successfully in plastics gearing is the tooth form with an addendum of 1.35/P shown in Appendix A. This tooth form will add little to the overall tool cost.

To eliminate undercut, the pinion will now be required to have a circular tooth thickness of 0.0424 – 0.0434. The minimum outside diameters of the pinion and gear will be 0.3796 and 0.6725, respectively. Returning to the cheaper unfilled acetal, a center distance tolerance of +0.003, −0.000 and AGMA Quality No. 7 gears, the designer tries again. The maximum center distance is now 0.4882

outside diameter	Do1	0.3796"
outside diameter	Do2	0.6725"
operating center distance	OCD	0.4882"
contact ratio	CR	1.20

Step 7. Check the load carrying capacity. Even though the loads in this application are light, the designer checks the allowable horsepower to determine what difference the new tooth form has made. If any change to the gear design is necessitated, it will amount to no more than a slight increase to the face width.

Step 8. Establish manufacturing and inspection data. Finally, the designer arrives at the manufacturing and inspection data to go on the production drawings. The complete data for the 15-tooth and 30-tooth gears used in this example are shown in Appendix E.

Manufacturing of Plastic Gears

Description of Molding Dies and the Injection Molding Process. A molded gear, of course, can be no more accurate that the molding die which produces it. Basic considerations for gear molding dies are the same as for any precision molded product. Since molded gears are usually small in relation to the size of average moldings, they lend themselves admirably to being molded in small, high speed automatic injection molding machines (20 - 80 ton range). This allows for compact dies, usually with one to eight cavities.

Several factors can be built in to a molding die to produce the most accurate gear possible.

- The increased strength of case-hardened die frames will enable them to withstand the abuses of the molding process and can maintain their accuracy throughout extended useful life.
- The elimination of wear bushings allowed by hardened steel surfaces in the bores removes the additional errors resulting from less than perfect concentricity of these items.
- The mold should have a balanced runner system to provide equal pressure drops into all cavities.
- Adequate venting in the cavities will allow air to be displaced by the flow of the plastic.
- Adequate ejection systems ensure minimum distortion of the product when ejected from the die.
- Adequate interlocks between the die halves remove the misalignment of running fits provided in the guide system.

Figs. 6 and 7 are cross sections of a gear molding die.

When the molding die is closed, the melted plastic resin is injected under high pressure into the gear cavities. When the resin solidifies, the molding die opens, and the plastic gear is ejected from the die. Cycle times usually range from 10 to 30 seconds, depending on the part geometry and the plastic resin used.

Several factors in a molding die can contribute to the gear's runout:

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GEAR TRAINING PROGRAM

1. BASIC FUNDAMENTALS

1. **Gear History**
   - A. Cycloidal Teeth
   - B. Involute Teeth
   - C. Gear Cutting Machines
   - D. Gear Cutting Tools
2. **Gear Types**
   - A. Parallel Axis
   - B. Intersecting Axis
   - C. Skew Axis
3. **Gear Ratios**
4. **Involute Gear Geometry**
   - A. Nomenclature
   - B. Involutometry — Contract Ratio, etc.
   - C. Helical Gears — Lead — Helical Overlap
5. **Gear Tooth Systems**
   - A. Full Depth
   - B. Full Fillet
   - C. Stub Depth
6. **General Formulae**
7. **Mathematics — (I.T.W. Trig Book)**

2. HI SPEED STEELS

- A. Common Types
- B. Special Types
- C. Heat Treatment — Metallurgy
- D. Controls
- E. Surface Treatments
- F. Special Cases

3. CUTTING THE GEAR

1. **Forming**
   - A. Milling
   - B. Broaching
   - C. Shear Cutting
2. **Generating**
   - A. Shaping
     - a) Rack Type
     - b) Circular Type
     - c) Machine Types and Manufacturers
     - d) Schematic — Principles
     - e) Speeds — Feeds
     - f) Machine Cutting Conditions
   - B. Hobbing
     - a) The Hobbing Machine
     - b) Types and Manufacturers
     - c) Schematic — Differential and Non-Differential
     - d) Speeds — Feeds
     - e) Climb Cut — Conventional Cut
     - f) Shifting — Types
3. **The Hob as a Cutting Tool**
   - A. How It Cuts
   - B. Tolerances and Classes
   - C. Multiple Threads
   - D. Hob Sharpening and Control
   - E. The Effect of Hob and Mounting Errors on the Gear
4. **The Shaper Cutter as a Cutting Tool**
   - A. Know Your Shaper Cutters
   - B. Design Limitations
   - C. Sharpening
   - D. The Effect of Cutter Mounting and Errors on the Gear
   - E. Manufacturing Methods
5. **Tool Tolerance Vs. Gear Tolerance**
   - A. Machining Tolerances
   - B. Gear Blank Accuracy and Design Limitations

4. FINISHING THE GEAR

1. **Gear Finishing — Before Hardening**
   - A. Shaving
     - a) The Shaving Cutter
     - b) Types of Shaving — Conventional, Underpass, Diagonal
     - c) Crown Shaving
     - d) Shaving Cutter Modifications
     - e) Co-ordinating Tool Design — The Shaver and Pre-Shave Tool
     - f) Re-Sharpening
     - g) Machines
   - B. Rolling
2. **Gear Finishing after Heat Treat**
   - A. Honing
   - B. Lapping
   - C. Grinding
     - a) Methods — Formed Wheel-Generating — Threaded Wheel
     - b) Machine Types

5. GEAR INSPECTION

1. **Functional**
   - A. Gear Rollers
   - B. Gear Charters
     - a) Reading the Chart
     - b) Tooth-to-Tooth Composite Error
     - c) Total Composite Error
   - C. Master Gears
     - a) Tolerances
     - b) Designs
     - c) Special Types
2. **Analytical**
   - A. Size — Tooth Thickness
   - B. Runout
   - C. Spacing
   - D. Lead
   - E. Involute
3. **Automatic and Semi-Automatic**
   - A. How They Work
   - B. What Can Be Checked
   - C. How Fast
4. **Chart Interpretation — Analytical and Functional**
   - A. Reading the Charts
   - B. Which Errors Affect Other Elements
   - C. How to Correct the Error the Chart Shows

6. INDIVIDUAL INSTRUCTION AND SPECIFIC PROBLEMS OR PLANT TOUR

This Program is flexible and changed to meet the needs of each group.

The 4-day program consists of a coordinated series of lectures given by the Engineering, Production and Inspection staffs of Illinois Tool Works Inc. They represent over sixty years of experience in gear and tool specialization.

The sessions are conducted with a technique and approach that leads to interesing discussion and participation. Starting with the basic fundamentals, the program is directed into those areas of greatest interest and value to the people attending the particular session. The groups are kept small so that this object is readily accomplished.

As mentioned, the planned program lasts four days. One-half of the fourth day is for individual discussion of specific problems in a detailed manner with members of the Illinois Tool Works' staff.

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All those attending are assigned to the same hotel. This promotes friendly contact and discussion of mutual problems and interests. Tuition for the course includes transportation from the hotel to ITW and back, one group dinner, all continental breakfasts and all lunches.

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CIRCLE A-8 ON READER REPLY CARD

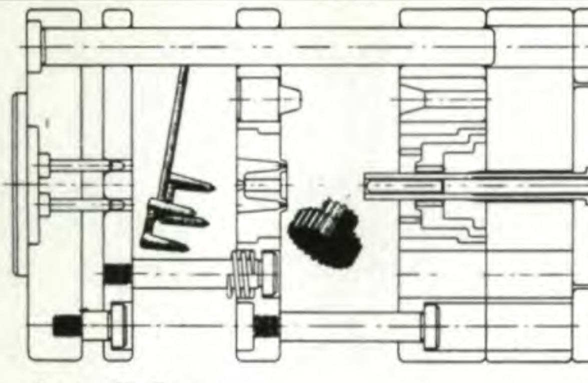

Fig. 6 Die Open

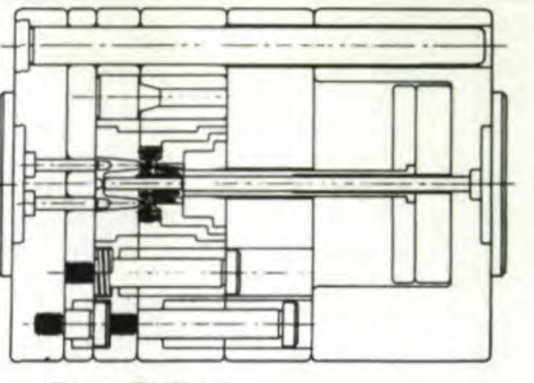

Fig. 7 Die Closed

• The .0001/.0002" running fit required between the core pin and the stationary half of the die;
• An additional .0001 to .0002" running fit will be present if sleeve ejection is required around the core pin because the solid anchor is lost;
• Because the gear cavities themselves are a series of concentric rings, the potential for runout error exists.
• Uneven plastic flow resulting from the off-center gating position on the part will introduce additional runout in the gear.

The selection of a molder should be given careful consideration. True, given a correctly designed and accurately made molding die, any competent molder can produce gears; however, not all molding companies can be classified as gear manufacturers.

A good plastic gear manufacturer should have the following capabilities:
• experience in molding a wide range of gears;
• ability to advise on product design and selection of materials;
• availability of molding presses most suitable for gear molding;
• availability of inspection equipment necessary to maintain proper quality control (such as gear checking equipment).

Gear Cavities. Popular wisdom holds that molded gears are not as accurate as machined gears. This is inaccurate. A molded gear held to the tolerance of AGMA Quality Number Q8 is just as accurate as a machined gear of the same quality number. It is true that gears have not yet been molded to the highest precision obtainable by machining, but the number of gears requiring such precision represent only a small percentage of all gears made. In general, plastic gears are usually specified AGMA quality Q6 or Q7.

The idea that molded plastic gears need not be as accurate as metal gears is based upon the contention that, because of the yielding nature of plastic, runout and tooth-to-tooth errors do not have the same ill effects. This is incorrect because plastic gear teeth that are flexing to an excessive degree because of inaccuracies will fail through fatigue and wear much earlier than accurate teeth.

All plastics shrink when changing from a molten to a solid state. As a consequence, all mold cavities must be made larger than the product specification. For example, if a molded gear is to have an outside diameter of 1.200", and the plastic has a mold shrinkage of .025 in/in, then the outside diameter of the cavity will be required to be 1.2308".

In making a gear cavity, however, it is not sufficient to take a generating hob and machine an oversize gear. Compensation

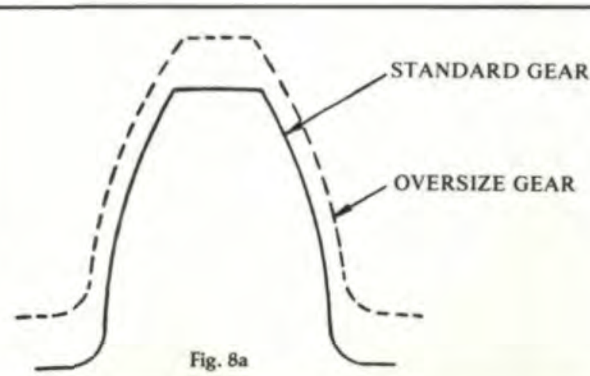

Fig. 8a

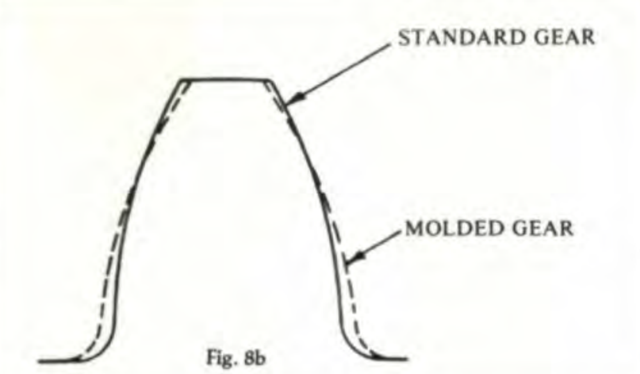

Fig. 8b

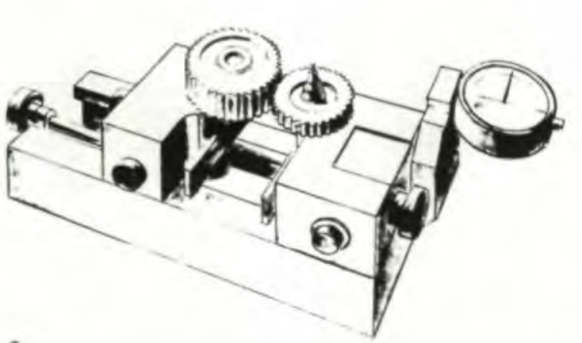
Fig. 9

must also be made for the pressure angle of the hob. If it isn't, the result will be a molded gear with a serious profile error. It will have a larger than acceptable tooth-to-tooth error. For example, in Fig. 8a an enlargement of a 32 D.P., 20 degree P.A., gear tooth is shown, and superimposed upon it is the profile of an oversize gear tooth cut with a standard 32 D.P., 20 degree P.A. hob. Fig. 8b shows the standard gear tooth, and this time superimposed upon it is the profile of the molded tooth (after shrinkage) that would be obtained from the oversize cavity.

Note that the tooth of the molded gear departs considerably from standard. It is thicker at the root and thinner at the tip. (It has a pressure angle much in excess of 20°.)

The formula used to calculate the correct pressure angle is:

$$\cos\phi_2 = \frac{D\cos\phi_1}{D(1+S)}$$

where:

D = Pitch circle diameter
ϕ_1 = Pressure angle of hob
ϕ_2 = Pressure angle of molded gear
S = Shrinkage

The teeth in the cavity must be carefully compensated for shrinkage so that, when the molded gear solidifies and becomes stable, the teeth will have the correct profile. This design work is further complicated in the case of the helical gear, because the axial shrinkage is usually different from the shrinkage across the diameter.

Compensating correctly for shrinkage in a gear requires that the mold designer have a thorough understanding of gear geometery, plus considerable experience in the shrinkage

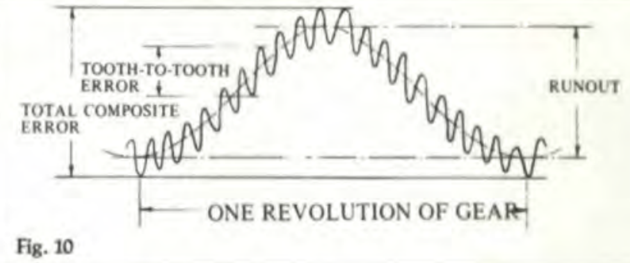

Fig. 10

behavior of all types and grades of plastics.

The actual manufacture of accurate gear cavities is accomplished by using conventional or wire EDM machines.

Inspection of Plastic Gears

One practical method of checking gear accuracy is to rotate the manufactured gear in close mesh with a master gear using a center distance measuring instrument. The model shown in Fig. 9 is equipped with a dial indicator and requires that the operator note the radial displacements as the gear is rotated manually through one complete revolution.

The more sophisticated models trace the radial displacements on a moving chart (See Fig. 10.) through an electronic device.

The errors present in a gear are runout, lateral runout (wobble), pitch error, and profile error.

The pitch error plus the profile error add up to the tooth-to-tooth composite error, and this, plus the total runout, is known

(continued on page 23)

EDITORIAL
(continued from page 5)

job vital to our economy requires four years of college. And, contrary to popular opinion, college does not provide the advanced training necessary to qualify for many highly skilled jobs. Furthermore, lack of interest in the traditional scholarly and professional training that colleges provide best does not make for a "second-class" employee.

But that's not the message we give our young people.

In the meantime, the metal working industries are short some 1500 workers every year in the Chicago metropolitan area alone, and we have no reason to think the situation is much better anywhere else in the country or that it will improve in the near future.

I'm not sure how we got to this place in our thinking about work. For my father's generation, high school was the end of the educational line for most people. Many were unable to remain in school even that long. Those who qualified for skilled apprenticeship programs in the "trades" were to be envied. No one thought that a young man or woman learning a skill that would provide economic security for a lifetime might be a "failure" because that skill was acquired on the job instead of on the college campus. "Trades" were not thought of as second choices for people too poor or unambitious to go to college. They were honest jobs requiring just as much hard work and training as any other. No one was embarrassed to say, "I'm a precision gear maker."

These jobs haven't changed. They still require intelligence and hard work. They are demanding and challenging; they provide a decent living and plenty of opportunity for advancement. No one needs to be embarrassed to admit to working in a gear shop.

It's our thinking on the subject that's changed. Because precision metal working or gear manufacturing are not necessarily "glamorous" or "professional," the highly qualified young people who might enter these jobs are not encouraged to even consider them. Maybe if somebody did a t.v. series called "LA Gear Cutting" and cast it with beautiful people in tailor-made coveralls, the job would be more appealing.

A more practical suggestion is to begin with our own attitudes. Those of us familiar with the gear business, whether we are managers, foremen, or operators, can encourage the young people we know to consider the possibilities of entering a "trade." We can inform them of the realities of work in skilled jobs like gear manufacturing: That during the last 40 years the U.S. has never been able to produce enough qualified precision metal workers to fill industry needs; that these jobs can pay as much as $35,000 to $50,000 a year; that even during their first year of training, precision metal working apprentices can expect to earn nearly $20,000 annually; that training is usually subsidized by the employers; that at age 26, a young tool and die maker can earn about $40,000 in contrast to the mechanical engineer of the same age, who will earn around $35,000.

Talk to the guidance counselors at your local schools. Do they know these facts? What are their attitudes about non-college job opportunities? Do they consider such careers "second-class" choices? Are they aware of the jobs available locally in gear manufacturing? Do our high schools provide any kind

(continued on page 22)

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EDITORIAL
(continued from page 20)

of shop education that might begin to steer qualified young people toward the skilled trades we desperately need?

We can also recruit capable young people to our own training programs. We can encourage organizations like the Tooling and Manufacturing Association, which has a program that actively works to spread the word about skilled precision metal working jobs to high school guidance counselors.

We can become involved in the efforts of local community colleges, many of whom are in the forefront of providing advanced manufacturing training. Working in conjunction with local industries, these colleges are providing classes, not for degree programs, but to support and supplement the on-the-job training that local manufacturers provide. Such school programs can be life-savers for the medium-to-small manufacturer who does not have the resources to provide an elaborate training program of his own. The offer of used machinery or time and expertise to such programs can help begin to close the gap between our need for trained workers and their availability. Discussions with school administrators about the need for such programs can be an important first step.

Those thousands of vacancies that go unfilled every year are vital to the success of our industry. Every year they remain vacant weakens our competitive position. It is not enough to complain about "inadequate training" or an "unskilled work force." We have a responsibility to get out the word about the opportunities in our field. We know that, far from being "second-rate", the skilled workers on our shop floors are talented specialists essential to the survival of our businesses. We have to let the rest of the world know that too.

Excellence in gear manufacturing is as important to our society as excellence in either plumbing or philosophy. If we allow gearing jobs to go unfilled because of some mistaken notion that they are "humble" or not worthwhile, not only will both our pipes and our theories fail to hold water, but our machinery, our businesses, and our economy will also suffer. Ultimately, it will be our society that ends up as "second-class".

Michael Goldstein

Michael Goldstein
Editor/Publisher

SYSTEMATIC APPROACH . . .

(continued from page 19)

as the total composite error.

As mentioned previously, the AGMA quality number is used to classify the quality of metal gears as well as plastic gears. The number specifies the gear accuracy in terms of maximum tooth-to-tooth and total composite tolerances allowed. The AGMA quality numbers and the corresponding maximum tolerances are listed in the AGMA "Gear Manual 390.03".

Summary

The injection molding process allows the gear designer to freely modify the tooth profile to obtain maximum tooth strength. These modifications will add little to the final tooling cost.

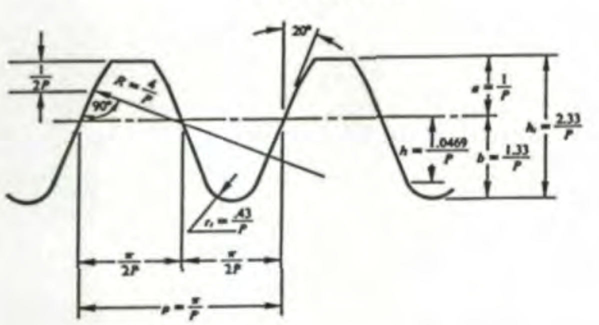
The gear design example was chosen to point out the many variables involved when designing plastic gears. The use of a systematic design approach allows the overall design problem to be broken down into smaller blocks which can be analyzed individually. Given another design with different constraints (fixed operating center distance, for example), the sequence of events in the design process would differ, but the problem would still be broken down into smaller blocks and analyzed one step at a time.

Correctly used, the systematic approach will provide a design for plastic gears having load carrying capacity, adequate contact, as close to balance tooth thickness as possible, and sufficient clearance so that no binding will occur at the extreme operating conditions.

APPENDICES

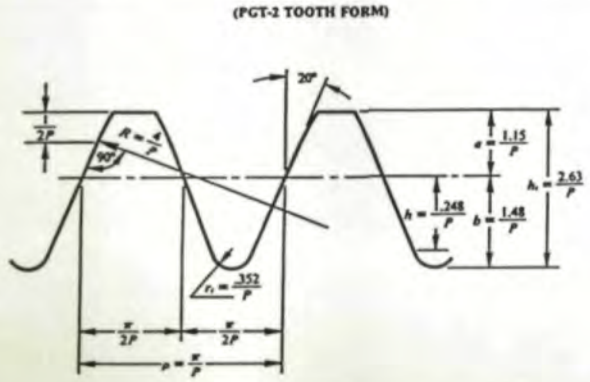
Appendix A — Tooth Forms for Plastic Gears.

(PGT-1 TOOTH FORM)

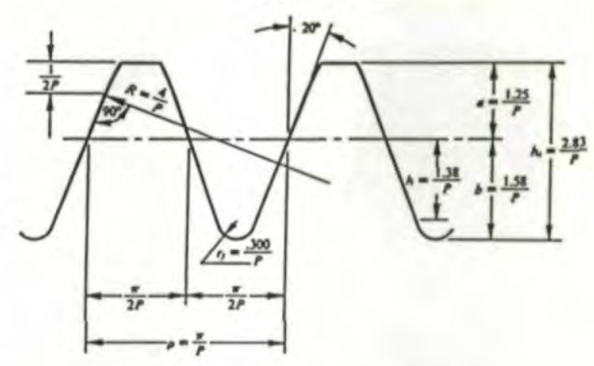


P = diametral pitch p = circular pitch 20° = pressure angle
a = addendum b = dedendum h_t = whole depth r_f = root radius
h = depth of straight portion of dedendum to point of tangency with root radius.

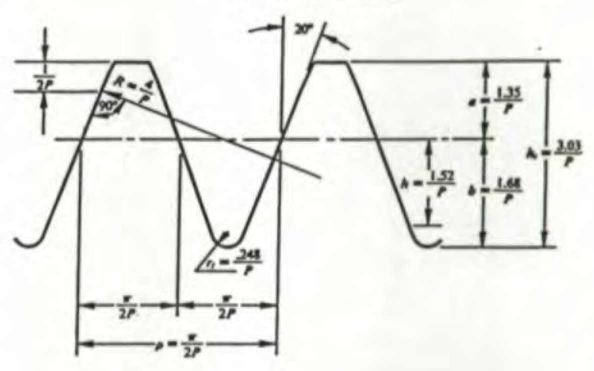
(PGT-2 TOOTH FORM)



(PGT-3 TOOTH FORM)



(PGT-4 TOOTH FORM)



(continued on page 26)

GEARMAKING!

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(ISO 53-1974 (E) Tooth Form)

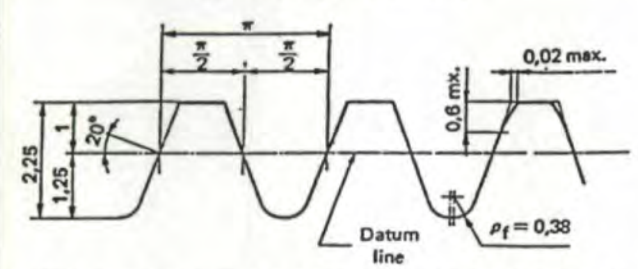
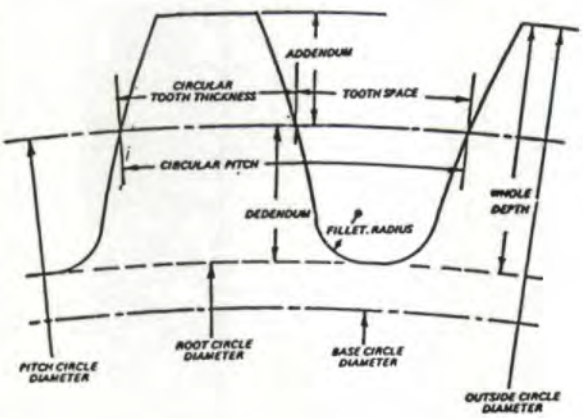


FIGURE — Representation of the profile of module $m = 1$ or of diametral pitch $P = 1$

(AGMA Tooth Form)



	Coarse-Pitch 19.99 and Coarser	Fine-Pitch 20.00 and Finer
Addendum	$\frac{1.000}{P}$	$\frac{1.000}{P}$
Dedendum	$\frac{1.250}{P}$	$\frac{1.20}{P} + .002$
Whole Depth	$\frac{2.250}{P}$	$\frac{2.20}{P} + .002$
Circular Tooth Thickness	$\frac{\pi}{2P}$	$\frac{\pi}{2P}$

P = Diametral Pitch

Appendix B – Computer Software.

If the design engineer has a personal computer and a gear software program, the time involved in designing a train of gears is reduced from many hours to a matter of minutes. It can prove to be an invaluable tool for a designer who wants to try many options in obtaining optimal gear meshes. Appendices B, C, and D describe one of the commercially available software programs for plastic gears.

Because mathematical and interpolation errors are eliminated, the designer is free to spend more time designing gears and less time solving mathematical equations.

Summary of Computer Program Features[6]

1) Program runs on IBM PCs and compatibles.
2) Design can be for either a spur or helical gear.
3) Designer can choose either English or Metric Units.
4) One gear can be designed alone, or two meshing gears can be designed simultaneously.
5) Gear data can be saved, retrieved, printed or deleted.
6) When an input value is entered in one of the sub-program selections, it is also entered in the remaining sub-programs. For example, if the diametral pitch of 48 is entered in sub-program "A", the user does not have to enter it again in sub-program "G"; it is already displayed there.
7) The calculations in the program have been designed similar to a spreadsheet. The user can enact "what if" variations by changing input values. The output values are immediately recalculated and displayed.

The program is a series of sub-programs which perform various calculations used in the process of designing spur and helical gears. (See Appendix C for a listing of the 12 sub-programs available in the "Main Menu"). Recognizable descriptions and symbols are employed throughout the program. The operational procedure can be mastered in less than an hour.

Appendix C — Main Menu Screen

A:	SME_15T	SME_30T	English	Spur	Mode 1	New
(dsk)	(gr #1)	(gr #2)	(unit)	(type)		

Plastics Gearing Technology, Inc. — Gear Design Program
****Main Menu****

A) Oper Phi, stand t, stand a, ht, and PD
B) Outside dia, root dia, and Domax (given t)
C) Minimum circular tooth thickness to avoid undercutting
D) Circular tooth thicknesses for balanced tooth strengths
E) CMCD, minimum operating center distance
F) Delta C
G) t1 + t2
H) Contact ratio, RA, AA, %RA
I) Gear testing radius (min and max)
J) Measurement over pins (min and max)
K) Horsepower ratings
L) Check and/or print gear data specified on drawing
M) Exit

Enter menu selection —>> M

F1	Help	F2	Type	F3	Mode	F4		F5	Dir	F6	ID1	F7	New	F8	Clr1	F9	Ret1	F10	Sav1
			Unit		Mon		Disk		Swap		ID2		Del		Clr2		Ret2		Sav2

Description of abbrevations used in the above Main Menu

Oper Phi	operating pressure angle	CMCD	close mesh center distance
stand t	standard circular tooth thickness	Delta C	required increase in center distance
stand a	standard addendum	t1 + t2	sum of the tooth thicknesses for a given CMCD
ht	whole depth	RA	recess action
PD	pitch diameter	AA	approach action
Domax	maximum allowable outside diameter	%RA	percent recess action
t	circular tooth thickness		

Appendix D — Example of a Sub-program Screen (For Contact Ratio Calculation).[6]

C:	SME_15T	SME_30T	English	Spur	Mode 1	Calc	Page 1 of 1

Contact ratio, RA, AA, %RA

Parameter Description	Symbol	Gr#	Value	Units
number of teeth	N	1	15	
number of teeth	N	2	30	
diametral pitch	P		48.0000	
outside diameter	Do	1	0.3796	inches
outside diameter	Do	2	0.6725	inches
operating center distance	OCD		0.4882	inches
recess action	RA		0.0501	
approach action	AA		0.0235	
contact ratio	CR		1.196	
percent recess action	%RA		68.11	

User Note:
Gear 1 is the drive gear and
Gear 2 is the driven gear.

F1	Help	F2	F3	Mode	F4	Calc	F5	Dir	F6	ID1	PgUp	Prev page Esc Exit
	Equa			Mon				Swap		ID1	PgDn	Next Page

Appendix E — Manufacturing and Inspection Data For The Pinion and Gear

PINION DATA

Basic Specifications	
number of teeth	15
diametral pitch	48.0000
pressure angle	20.00°
standard pitch diameter	0.3125"
PGT tooth form	4
standard addendum	0.0281"
standard whole depth	0.0631"
circ tooth thickness max at PCD	0.04340"
circ tooth thickness min at PCD	0.04240"
Manufacturing and Inspection	
gear testing radius max	0.1716"
gear testing radius min	0.1677"
AGMA quality number	Q7
max total composite tol of gear	0.0026"
max tooth-to-tooth comp tolerance	0.0016"
number of teeth in master gear	96
circ tooth thickness of master gear	0.03270"
testing pressure	8.0 oz
diameter of measuring pins	0.0400"
measurement over two pins (max)	0.3920"
measurement over two pins (min)	0.3903"
max allowable outside diameter	0.3826"
outside diameter tol (+ tol)	0.0000"
outside diameter tol (− tol)	0.0030"
max root diameter	0.2718"
Engineering References	
mating gear part number	30T gear
number of teeth in mating gear	30
max operating center distance	0.4882"
min operating center distance	0.4852"

GEAR DATA

Basic Specifications	
number of teeth	30
diametral pitch	48.0000
pressure angle	20.00°
standard pitch diameter	0.6250"
PGT tooth form	4
standard addendum	0.0281"
standard whole depth	0.0631"
circ tooth thickness max at PCD	0.03100"
circ tooth thickness min at PCD	0.03000"
Manufacturing and Inspection	
gear testing radius max	0.3114"
gear testing radius min	0.3074"
AGMA quality number	Q7
max total composite tol of gear	0.0026"
max tooth-to-tooth comp tolerance	0.0014"
number of teeth in master gear	96
circ tooth thickness of master gear	0.03270"
testing pressure	8.0 oz
diameter of measuring pins	0.0400"
measurement over two pins (max)	0.6851"
measurement over two pins (min)	0.6828"
outside diameter	0.6765"
outside diameter tol (+ tol)	0.0000"
outside diameter tol (− tol)	0.0040"
max root diameter	0.5503"
Engineering References	
mating gear part number	15T GEAR
number of teeth in mating gear	15
max operating center distance	0.4882"
min operating center distance	0.4852"

References

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Acknowledgements: *Presented at the SME Gear Processing and Manufacturing Clinic, November 1-3, 1988, and at the American Gear Manufacturers Association 17th Annual Gear Manufacturing Symposium, April 9-11, 1989. Reprinted with permission of both organizations.*

Hard Finishing and Fine Finishing Part 2

Dr. -Ing. H. Schriefer
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Editor's Note: Part I of this article dealing with the geometry of gear flanks, kinematics, and drive systems, tools, and computer assistance for hard and fine finishing, appeared in our last issue.

Operating Sequences and Machining Results

Fig. 1 shows the possible operating sequences for fine finishing and for hard finishing, integrated into the traditional sequences. It clearly follows from this table that the aims of fine finishing differ from those of hard finishing.

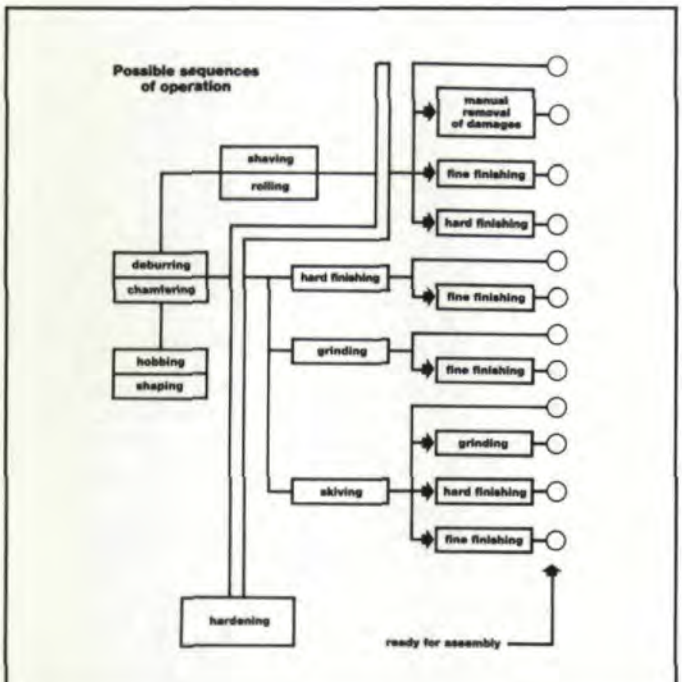

Fig. 1 – Operating sequences for fine finishing and for hard finishing.

AUTHOR:

Dr. – ING. HERBERT SCHRIEFER *is with Hurth Company, Munich, where he is technical manager for the Division of Machines and Systems. He studied at the Technische Universität in Berlin and worked at the Laboratory for Machine Tools (WZL) at the Technische Hochschule Aachen where he developed the geometry and kinematic part of a program sequence for bevel and hypoid gears. This program system represents the most extensive and universal computer analysis of the running and stress conditions of complex gear drives today.*

After shaping or hobbing, the tooth flanks must be either chamfered or deburred. Here it is paramount that the secondary burr produced will not be formed into the flank, but to the face of the gear, because during hardening, the secondary burr will straighten up and, due to its extreme hardness, will lead to excessive tool wear.

It is characteristic of all further operating sequences represented that the tooth gearings produced are ready for immediate assembly. Which of the operating sequences shown is optimal will depend on the quality required, on economic considerations, and on the circumstances of the sequences available.

Now we will show some examples of fine finishing and hard finishing. Along with conventional evaluations using the measurements of profiles, tooth traces, and pitch, statistical evaluations will be made to document the uniform output. To this end, workpieces are taken from manufacturing lines, and the results of profile and tooth trace measurements are plotted by using a point of reference. Thus, broom diagrams, whose range of tolerance is limited by preset windows, are set up. Further evaluations are obtained by means of the double flank total composite error test and by noise measurements.

Results of fine finishing are shown in Fig. 2. Damages shown on the double flank total composite error diagram have been removed, and a distinct reduction on the broom diagrams can be seen.

After removing the damages and improving the flank geometry, fine finishing is followed by a marked reduction of the generated noise.

Noise analyses are carried out in an unloaded condition. Noise emissions of car gear transmissions are particularly annoying in the lower load range, since they tend to override those of the engine.

As you will note in Fig. 1, another aim of fine finishing is to dispense with the noise test and the manual removal of damages after hardening of shaved or rolled gears. This may be illustrated by a hard finishing mode where work gears occur with damages at the flank sides of up to 350μm. The task is essentially to remove the defects in the marginal area of the flanks, while taking into account relatively large admissible radial deviations and wobbling of the gears. On a typical customer sample this procedure resulted in a tool life of a finishing tool in excess of 8000 parts and a cycle time of about 15 seconds per part. A considerable optimization of quality and cost for finished gears in mass production can be reached because of low production costs for green shaving or rolling, the elimination of the noise test and removal of defects, and the little effort required for fine finishing. The use of simple work conveyors also contributes to higher quality and lower costs, although these conveyors are not entirely safe from causing damage themselves.

An indispensable prerequisite is, however, that the heat treatment will not lead to excessive incalculable variations of the tooth geometry.

The fine finishing process also resulted in the percentage of gear boxes returned for reassembly dropping to a range of 0.3 per thousand, though the fine finishing was not followed by a noise test. The reason is that manual removal of defects involves many subjective imponderabilities, while fine finishing ensures that even the slightest damages will be definitely removed.

Inadmissible deviations of the teeth after hardening, such as

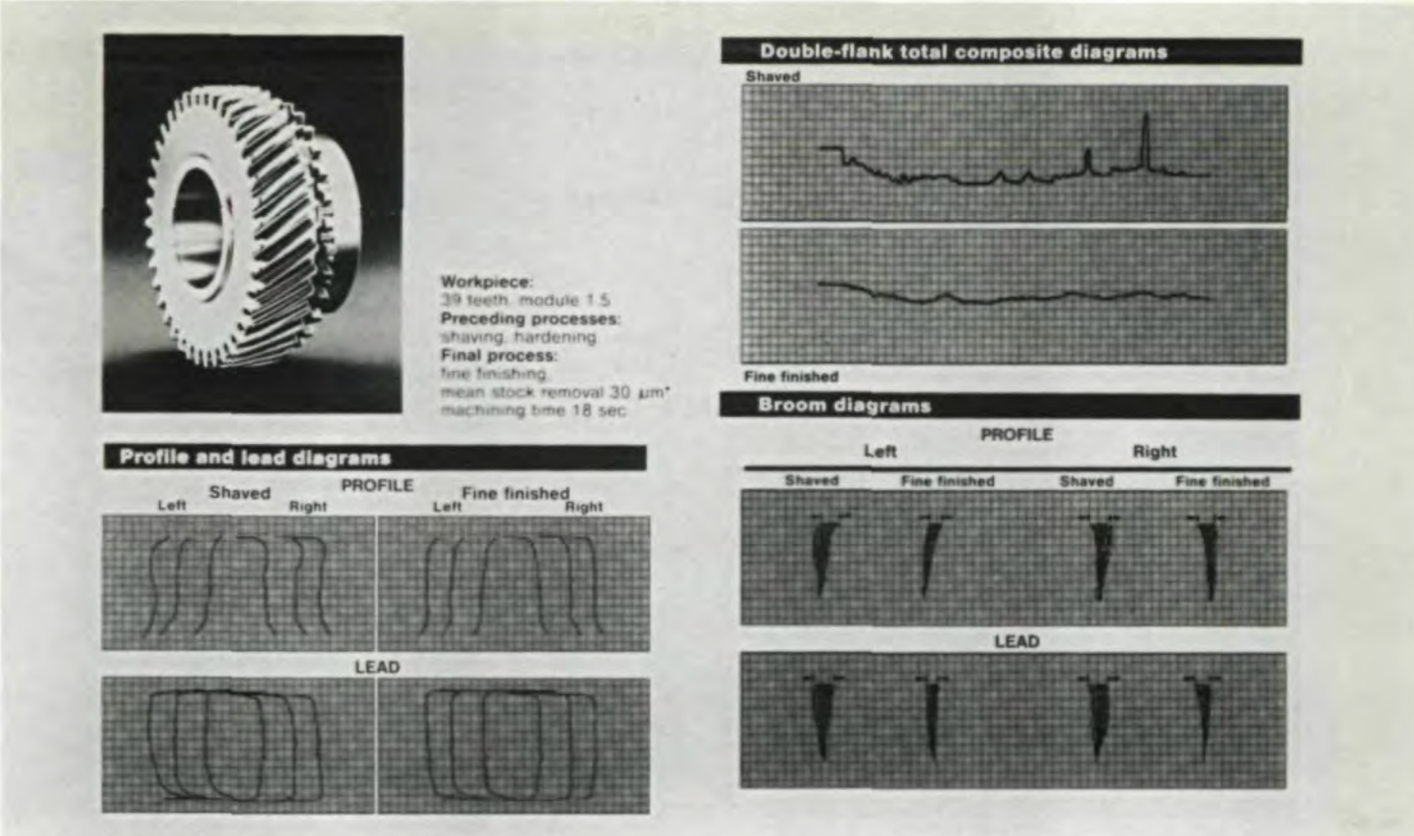

Fig. 2 – Working example of fine finishing a gear.

bent teeth, need, however, to be removed before fine finishing by means of an automatic two-flank rolling test.

Fig. 3 shows the macrogeometrical results before and after hard finishing with a CBN-coated tool. It can be seen that hard finishing will produce an entirely definite geometry of tooth flanks.

The desired gear flank has a slight profile bearing and face crowning, and it has been slightly modified at the addendum area. In this case, a stock removal per flank of 50µm will be sufficient.

Since any highly accurate hard machining process is always the most complicated and costly operation in the work flow of gear production, this production step should not be used in improper extension. A minimization of the required stock removal from the hardened flank is also appropriate for the necessary case-depth (1/10th mm of case-depth thickness will require a cementation time of about one hour) and in view of the reduction of hardness caused by the removal of excessive layers at the marginal areas.

It should be further noted that, for reasons of load capacity, the machining step in the root fillet should be as small as possible, or precutting should be carried out with the protuberance entering tangentially into the hard-finished flank.

The aim of minimizing the case depth also results from the requirement *not* to core-harden the synchro mesh gearing nor the tooth tips of addendum modified teeth with an extremely high tip and, consequently, a minimum thickness at the tip.

For the example shown the machining time was 48 seconds. As an average, quality grade 5 acc. DIN 3962 was obtained. For the roughness of the flanks we have $R_{ZD} = 3\ \mu m$, $R_a = 0.45\ \mu m$. The variation of the measurement over flanks is less than 20 µm.

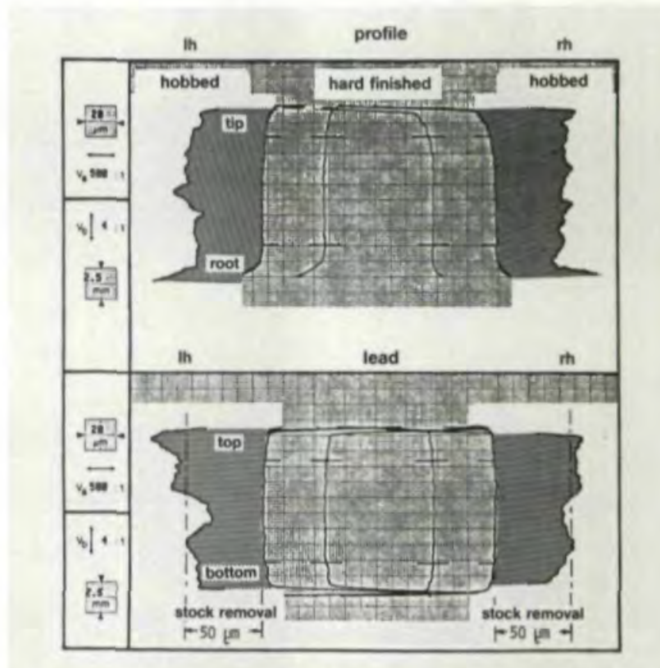

Fig. 3 – Profile and lead diagrams before and after hard finishing.

Results from hard finishing are presented in Tables 1 and 2 together with profile, lead, and pitch measurement records. (See Figs. 4-12.)

In the middle of the profile and lead records the required modifications are given.

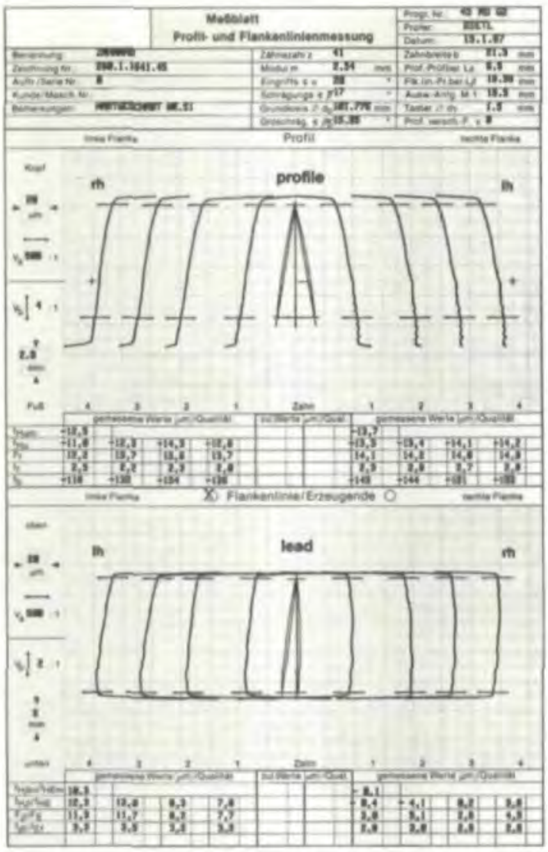

Fig. 4 – Machining example of hard finishing No. 1.

Table 1
Machining Examples of Hard Finishing

	Fig. 14/15 Example 1	Fig. 16/17 Example 2	Fig. 18/19 Example 3	Fig. 20/21/22 Example 4
Number of teeth	41	32	51	31
Module	2.54	1.75	2	1.75
Pressure angle	20°	15°	15°	17.5°
Helix angle	−17°	−25.75°	29.5°	−30.5°
Face width	21.5 mm	16.7 mm	17 mm	16.5 mm
Application	1st speed	3rd speed/ collar	1st speed	5th speed
Modification of pressure angle	lh/rh 12μm slanting	lh tip relief rh slanting by 4 μm	lh/rh tip relief	lh/rh tip relief
Profile crowning	lh/rh 2 μm	lh/rh 3 μm	lh/rh 3 μm	lh/rh 3 μm
Modification of helix angle	lh 10 μm rh 0 μm	lh 25 μm rh 12 μm	lh 8 μm rh face mod.	lh 7 μm rh 0 μm
Lead crowning	lh/rh 2 μm	lh/rh 4 μm	lh 4 μm rh 2 μm	lh/rh 5 μm
Cutting material	corundum	CBN	corundum	CBN
Bonding + blank	ceramic wheel	nickel + steel wheel	ceramic wheel	nickel + steel wheel
Premachining	pre-shaved	shaped	hobbed	hobbed
stock removal/flank	55 μm	45 μm	50 μm	50 μm
Quality acc. DIN 3962	5 to 6	5	6	6
Surface roughness R_{ZD}	2 μm	7 μm with subsequent fine finish.	3 μm	4 μm
Machining time	67 sec.	69 sec.	71 sec.	49 sec.
Tool life/per dressing cycle	30 off	total 8,000 off	25 off	total 7,300 off
Number of dressing cycles	35 off	–	30 off	–
Number of recoatings per tool	–	5	–	5

Experience with successful modifications for noise reduction in wide load ranges demonstrate that what matters is correctly chosen modifications of the pressure angle and helix angle with the least possible crowning. As a rule, the drive flank and the coast flank should be modified differently. Flank corrections in the lead-in and lead-out area are carried out degressively. In

Table 2
Main Characteristics of Fine Finishing and Hard Finishing

	Fine finishing	Hard finishing
Machine concept	• development from modular system of green shaving machine	• same
	• connected load 12 kVA	• connected load 15 kVA
	• part is driven and brake force applied to tool	• part and tool are kinematically locked to a master gear transmission
	• stored-program control	• CNC controlled, two axes
	• no additional motions during machining	• radial infeed
Tool	• external gear, outside diameter 230 mm	• same
	• corundum, synthetic resin bonded	• corundum, ceramic bonding or CBN coated
	• repeated dressing with diamond dresser is possible	• same
	• number of pieces per tool life depends on stock removed, number of work gear teeth, hardening distortion, radial deviation, quality of premachining, requirements on quality grade to be obtained	• same
	• example of tool life: m = 2.2, Z = 35 stock removal 2 ÷ 15 μm dressing cycle 100 ÷ 400 parts number of gears per tool life up to 8000 pcs.	• same stock removal 40 ÷ 150 μm dressing cycles 15 ÷ 60 parts number of gears per ceramic tool, up to 1200 pcs. number of gears per CBN tool, up to 10,000 pcs.
Feed and infeed	• no longitudinal feed no infeed constant distance between center lines	• same radial infeed
Meshing conditions	• continuous motion without traces of generated cut and feed	• same
	• crossed-axes angle 10 - 45°	• crossed-axes angle 6 - 20°
	• single-flank machining by reversing sense of rotation	• same
Part	• module 1.2 ÷ 6 mm	• 1 ÷ 4 mm
	• outside diameter 20 ÷ 280 mm	• 45 ÷ 250 mm
	• Width of tooth flank up to 50 mm	• up to 35 mm
	• helix angle 0° and ± 40°	• same
Tooth geometry	• depending on quality of precutting	• on an average better than quality grade 6 acc. DIN 3962
	• R_{ZD} smaller than 2 μm possible	as required: • R_{ZD} = 2 to 5 μm (ceramic) • R_{ZD} = 4 to 8 μm (CBN)
Machining time	• *about 0.5 min.*	• *depending on machining criteria, about 1 min.*

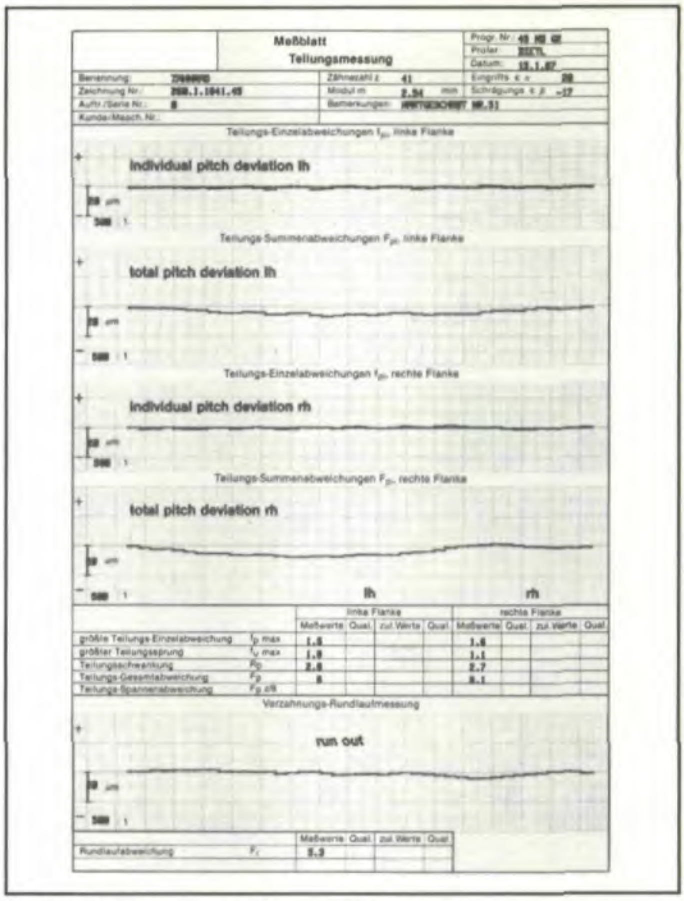

Fig. 5 – Machining example of hard finishing No. 1.

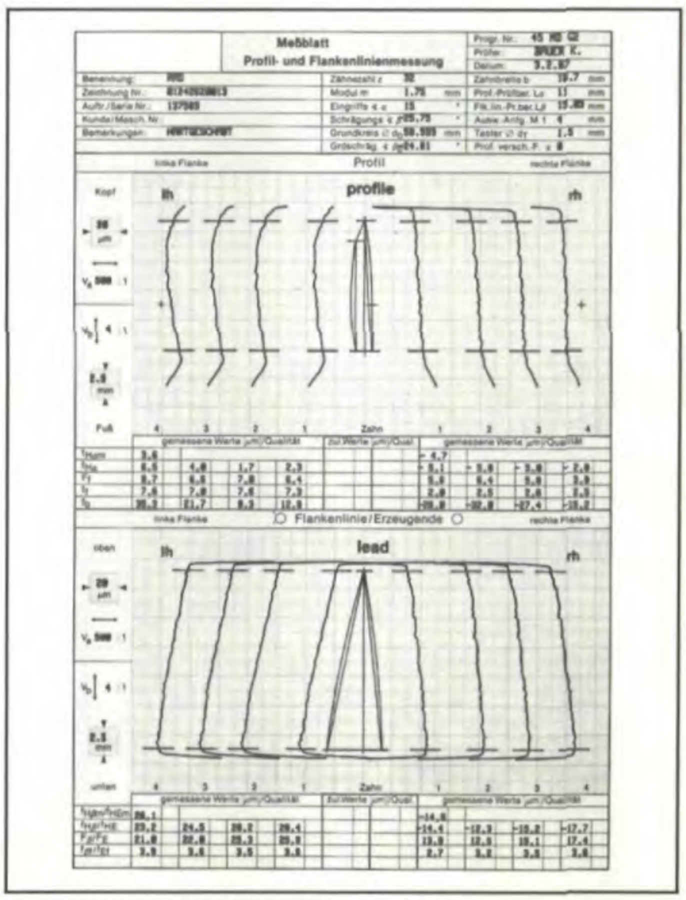

Fig. 6 – Machining example of hard finishing No. 2.

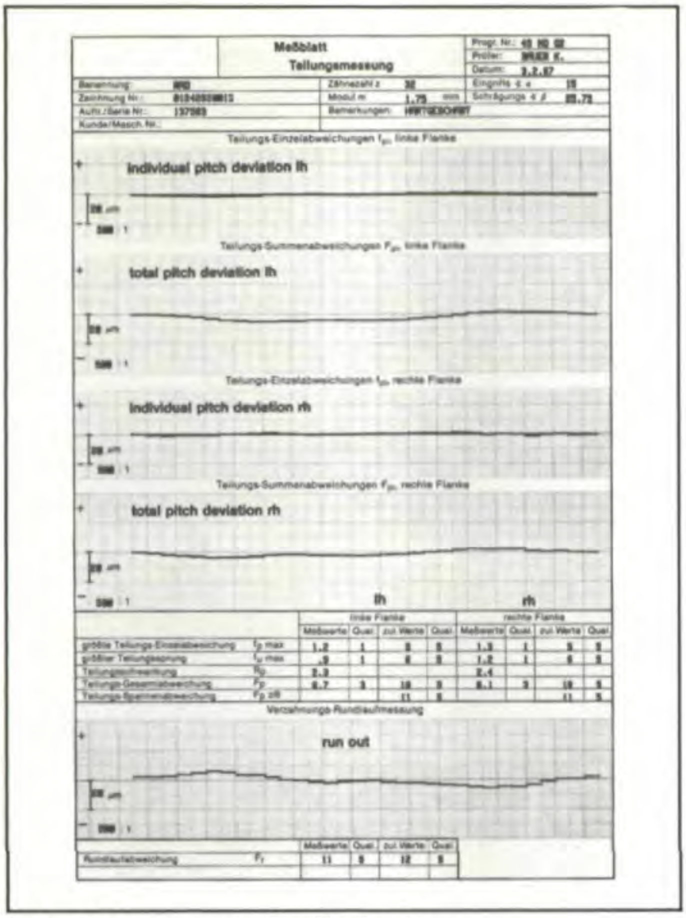

Fig. 7 – Machining example of hard finishing No. 2.

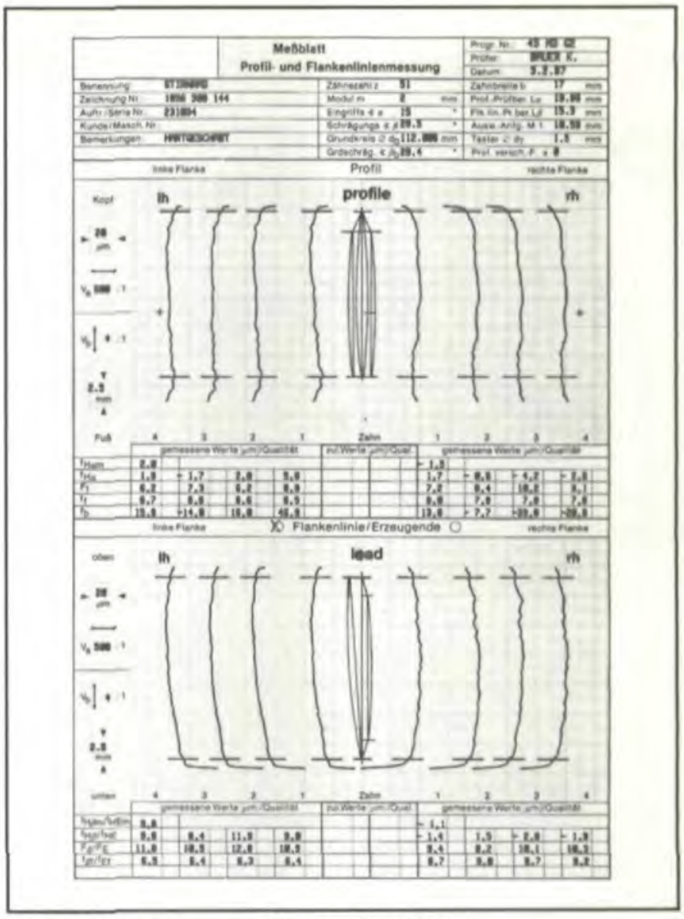

Fig. 8 – Machining example of hard finishing No. 3.

order to be able to use the least amount of crowning, the admissible pitch errors must be kept small. This will permit at the same time, the best possible distributions of contact and root fillet stresses in mesh for critically high load stages with the least possible influence of additional dynamic forces. In other words, apart from improvements on the noise sector, considerable increases of the load capacity will be obtained.

Gear Noise with Hard Finished and Fine Finished Gears

Fig. 13 schematically shows the possibilities offered by the hard machining processes in order to counter the noise generated.

A reduction of the gear noise will start with the implementation of tooth gearing data (face width, helix and pressure angle, addendum modification factor, and high tooth design, etc.).[3,6] The aim is to obtain a maximum contact ratio for smoothing the modulation of rigidity and to choose a contact ratio of an integral number of teeth, taking into account the contact conditions

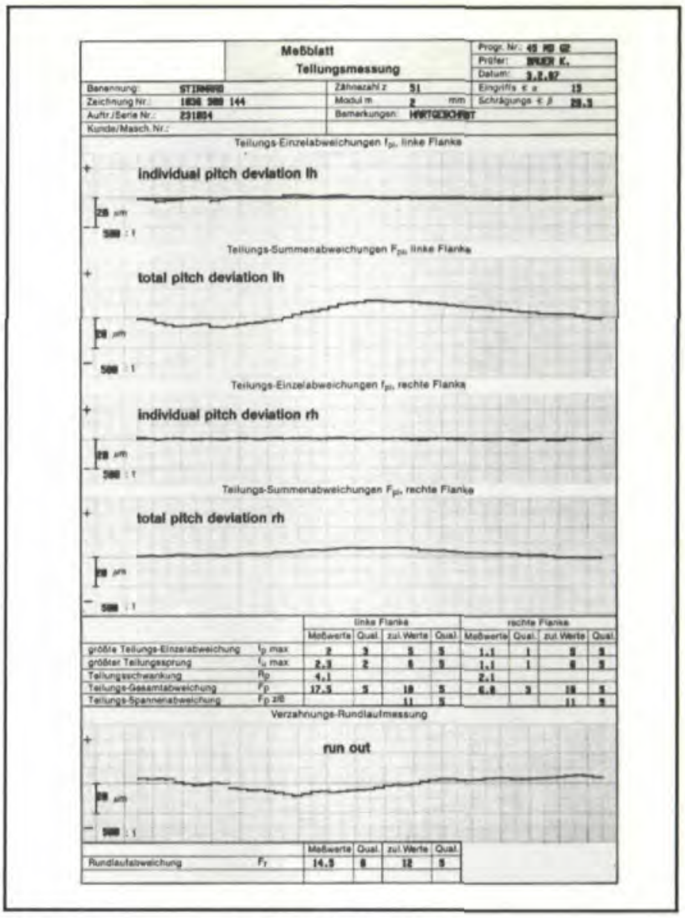

Fig. 9 – Machining example of hard finishing No. 3.

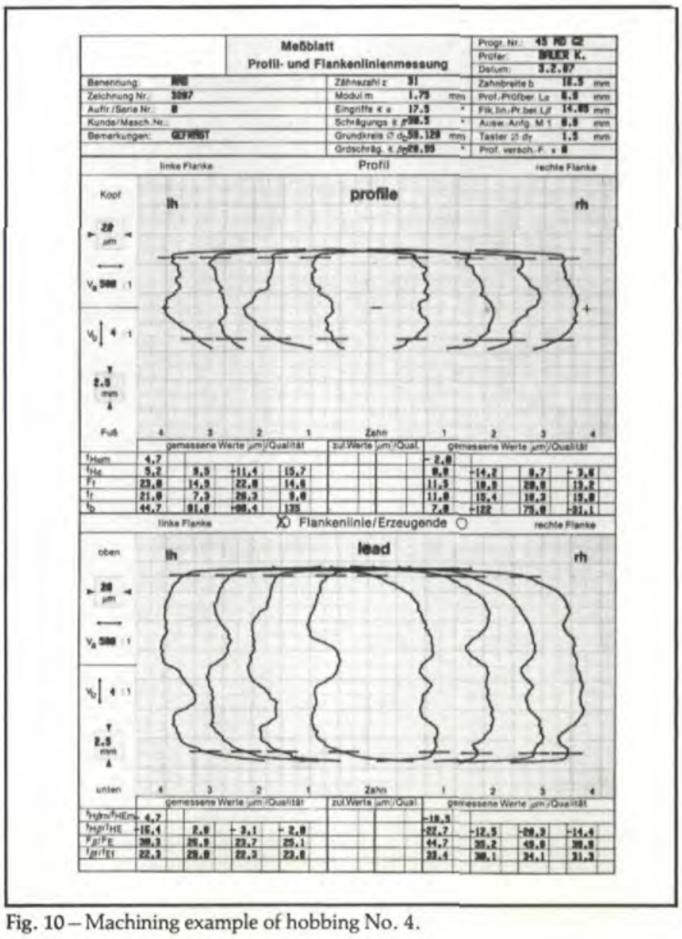

Fig. 10 – Machining example of hobbing No. 4.

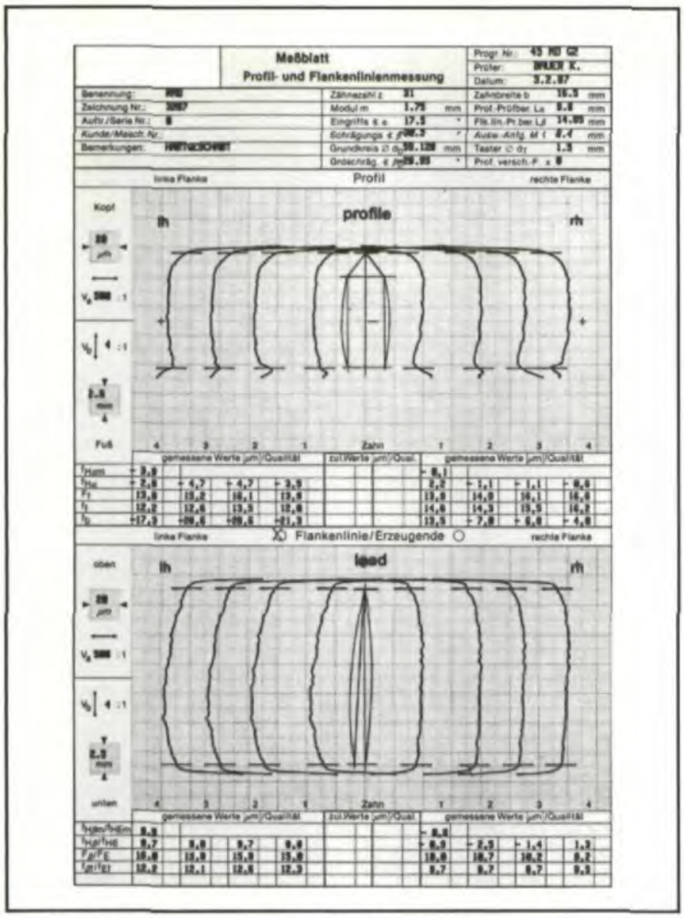

Fig. 11 — Machining example of hard finishing No. 4.

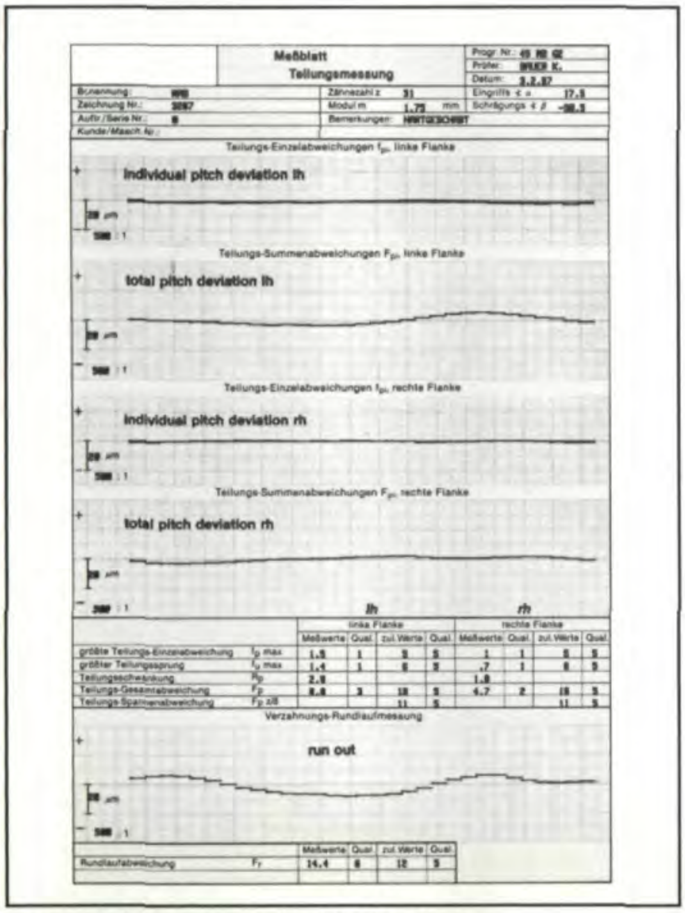

Fig. 12 — Machining example of hard finishing No. 4.

under load. The modification band width should permit a configuration of the tooth pattern that will allow no erratic variations of rigidity. Only a defined tooth bearing development will permit a modification of flanks for a wide-band load range. Otherwise heavy noise will be generated mainly in the partial load range, since the rather unyielding partial-load tooth pattern will result in a contact ratio of a non-integral number.

At the same time, the tooth contact pattern should have such an ease-off geometry that neutralizes deviations caused by manufacturing and faulty meshing due to the load is minimal.

A conclusion drawn from these circumstances is that only a tooth flank design specifically adapted to the possibility of a defined tooth pattern will be able to make full use of the essential advantages of hard finishing.

The practical procedure for defining the modifications can be ascertained by checking the tooth pattern, as with copper-plated gear flanks, with the gear unit in actual operation.

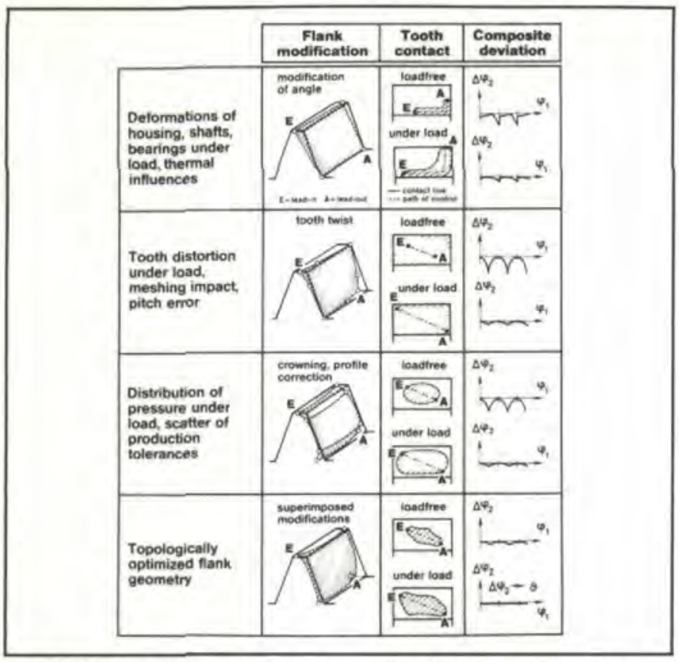

Fig. 13 – Systematic diagram of superimposed modifications of flanks for noise reduction.

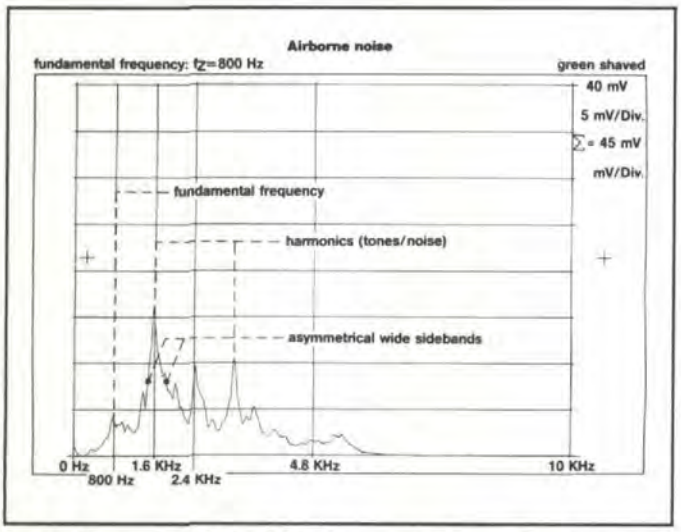

Fig. 14 – Frequency analysis of the airborne noise of green shaved gears.

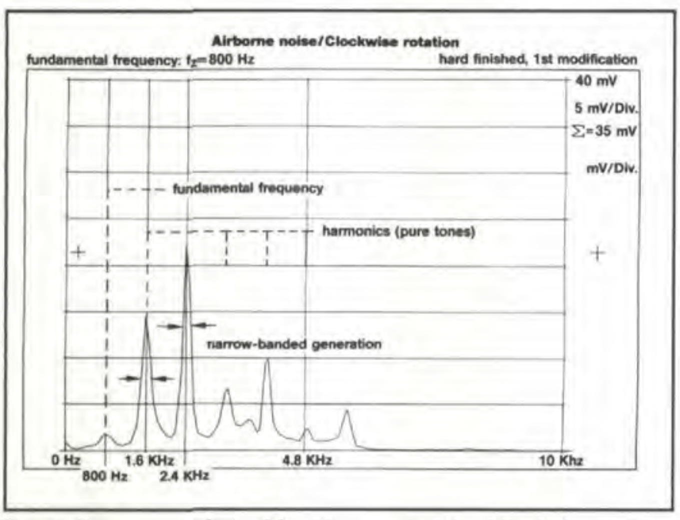

Fig. 15 – Frequency analysis of the airborne noise of hard-finished gears (first design of modifications).

The first step is to define the modifications of the helix and pressure angles.

These modifications differ for the drive and coast flank geometry according to the deformation behavior of the system of shafts and bearings, as well as of the gear housing.

Angle modifications are defined for a characteristic load stage of the gear set by evaluating the tooth patterns. According to a recognizable trend, a flexible grinding process is used to provide gears with modifications that will cover the estimated area of modification. These tests will be continued until satisfactory contact pattern (no noise behavior) is achieved.

The second step is to define modifications which will lay down the band width of the admissible load range and which will neutralize a scatter of deviations caused by manufacturing on account of production tolerances. Mainly, this would concern profile corrections in the lead-in and lead-out areas or addendum and dedendum areas and superimposed crownings.

The procedure is the same as explained above, though in this case the preset load range will be investigated step by step.

Effective modifications for noise reduction are characterized by an unerring choice of angle modifications and by the least possible superpositions of profile corrections and crownings in the range of a few μm.

The effectiveness of these modifications will essentially depend on the very close scatter of production tolerances of the gears. For instance, pitch, profile, and helix angle deviations could reduce and even neutralize the effect of such modifications.

In the following, a noise reduction on a test rig will be described by way of example. A frequency analysis of the airborne noise signals was carried out.

A green shaved pairing of gears is used as reference (Fig. 14).

With the frequency analysis, the partially stochastic distribution of amplitudes in Fig. 14 is striking. Though the first harmonic dominates, it is not noticed subjectively as such in the noise.

In the following, the gears were hard-finished with two different modifications.

The first design of modifications supplied the distribution of amplitudes shown in Fig. 15. The modifications were established from these green shaved gears running particularly smoothly. The amplitude with fundamental frequency (800 Hz) diminished markedly.

With the harmonics, however, a momentous narrow-banded rise of amplitudes occurred.

This narrow-banded generation, especially of the first harmonic, is subjectively disagreeable. The narrow-banded generation came into being due to an accurate gear, geometry (especially the very small pitch error) with unsuitable modifications.

So far, the transfer of modifications from green shaved gears has not been successful in all cases, since the necessary crownings required by hard finishing are much smaller than for green shaving, because for hard finishing correctly selected modifications of the helix and pressure angles with minimal crownings are of prime importance for getting best possible contact ratio. The second design of modifications will have the distribution of amplitudes shown in Fig. 16.

The marked drops of amplitude with only slight relative protruding amplitudes, and the consequential wide-band quality

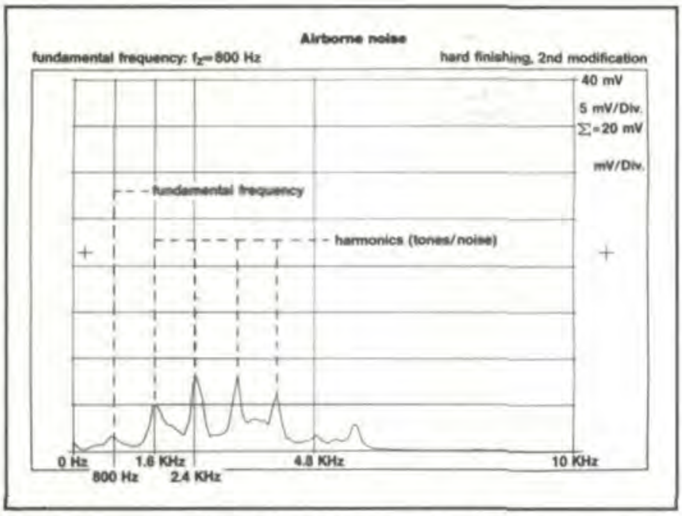

Fig. 16 – Frequency analysis of the airborne noise of hard-finished gears (second design of modifications).

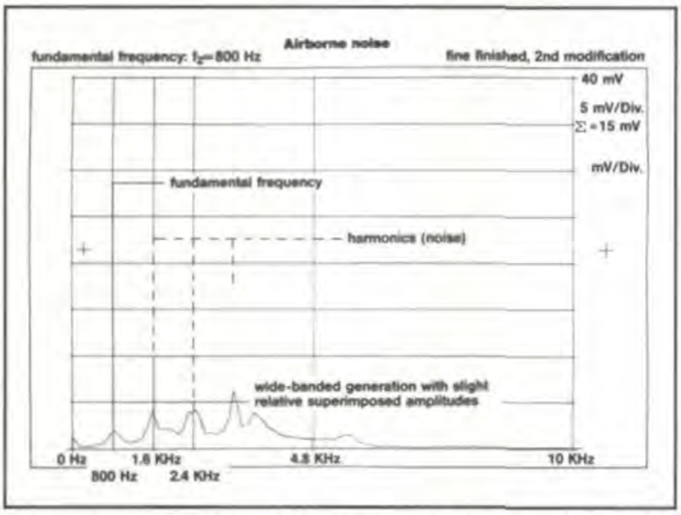

Fig. 17 – Frequency analysis of the airborne noise of hard-finished and fine finished gears.

provide a good subjective noise impression, since no individual tone made itself markedly heard.

In the next step of improvement the microstructure of flanks, with a roughness value $R_{ZD} \approx 3\ \mu m$ was reduced to $R_{ZD} \approx 1\ \mu m$ by fine finishing without changing the geometry. Fig. 17 shows the resulting frequency analysis.

The peaks of amplitudes with reference to Fig. 16 were further levelled. The subjective noise impression was especially improved for the well audible frequencies. If the noise relative to Fig. 16 still produced a sound impression, the noise relative to Fig. 17 increasingly merged into a noise similar to wide-banded rustle. The noise improvement, however, should not be primarily attributed to a reduction of surface roughness, but to the removal of microgeometrical meshing faults.

When investigating the deflections of gears on the test rig, marked acoustical differences were obtained in specific deflection and speed stages, requiring additional selective modifications of the flanks or damping methods for the natural frequencies of the gear blanks.

The strategy of defining modifications also consists in providing the results obtained under ideal testing conditions in such a way with a modification band width so that good results are obtained even in actual operation of the gear unit under various load stages. The modifications ultimately found and specific tooth gear quality parameters must be frequently kept at closer tolerances than required in the average quality specifications in order to obtain optimum noise results. Likewise, high demands must be made on the quality of the gear housing and to the systems of shafts and bearings if the noise reduction is to take a maximum effect.

Summary

Starting from the geometrical and qualitative requirements made on hard finishing methods of gears using geometrically undefined cutting edges, a brief survey was given on suitable tool configurations, the kinematical conditions, and possible drive modes for hard finishing and for fine finishing. (See Part I.)

The importance of tool manufacturing and the part played by computer assistance for tool design were dealt with. Machining results demonstrated the efficiency of the hard finishing methods of gears. Finally, the basic procedure to be followed for establishing tooth flank modifications for noise reduction were discussed.

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against the allowable values. If the check is successful, then the preliminary stage of the design is declared complete. It is suggested[2] that the face width not exceed $5p$ and be at least $3p$ where p is the circular pitch. Dudley[1] suggests that the face width should not exceed the value of the pinion diameter. So, if the calculated face width meets both of the above criteria, then the preliminary design is complete; otherwise, the initial pinion tooth number selection is revised, and the process is repeated until a satisfactory value of face width is found. Notice that if the minimum tooth number criterion is not met, a new class of materials based on a revised Bhn value is selected, and the design process is reinitiated.

Design for Strength and Durability

After the preliminary design, the tooth must be checked for strength and durability. Calculations for tooth strength and surface durability are based on the unit load and the K-factor.[1] Unit load is a factor which is a measure of strength of the tooth while the K-factor is a measure of surface durability or resistance to pitting.

The unit load is calculated based on the given information about the horsepower (HP):

$$U_l = \frac{HP * P * 126050}{n * d * F}$$

where d = pinion pitch diameter (inches).

The calculated unit load is now incorporated in the formula for tooth strength evaluation. The formula groups all the geometric design variables into essentially three factors:[1]

$$S_t = K_t\, U_l\, K_d \qquad \text{(psi)}$$

where
K_t = a dimensionless geometry factor,
U_l = an index of load intensity (unit load),
K_d = overall derating factor for bending strength (psi).

The geometry factor is a measure of shape of the tooth and its effect on the bending stress. Factors such as the depth of the tooth, pressure angle, and addendum/dedendum ratio are incorporated in assigning the geometry factor. Index of load intensity is designated by the unit load. The derating factor is an index to account for the non-uniform load distribution across the face width. It also takes into account dynamic overloads due to spacing error and the effect of the masses of the pinion and the gear. It also compensates for other quality related concepts, such as surface finish effect, overloads due to non-steady power, and metallurgical variations between small and large size gears.

Overall derating factor is defined by the following formula:[1]

$$K_d = \frac{K_m K_a K_s}{K_v}$$

where
K_a = application factor
K_m = load distribution factor
K_s = size factor
K_v = dynamic load factor

A similar derating factor can be defined for the tooth surface durability:[1]

$$C_d = \frac{C_a C_m C_s}{C_v}$$

where
C_a = application factor
C_m = load distribution factor
C_s = size factor
C_v = dynamic factor

It is assumed that all the factors constituting durability and strength factors are the same except the size factor (C_s). From the standpoint of surface durability, face width is probably the best way to evaluate the effect of size.[1] The size factor for durability is generally intended for derating of gears based on metallurgical discrepancies between large and smaller gears.

The rules of thumb employed for the size factors are[1]

1) For face widths up to 5", the size factor is set to 1.
2) For face widths greater than 5" and up to 16", C_s is set to 1.3.

The level of reliability chosen in this particular approach is L_1 (expected life with 99% reliability). However, one can define several reliability factors, depending upon the nature of the design needs. Based on the level of reliability and the number of required life cycles, the allowable values of strength and durability can then be obtained from the material

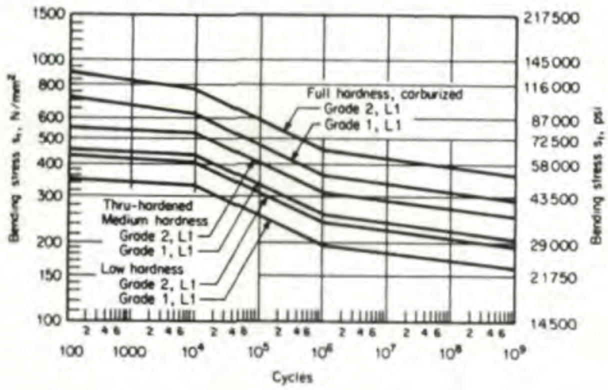

Fig. 1 – Bending strength versus life cycles[1].

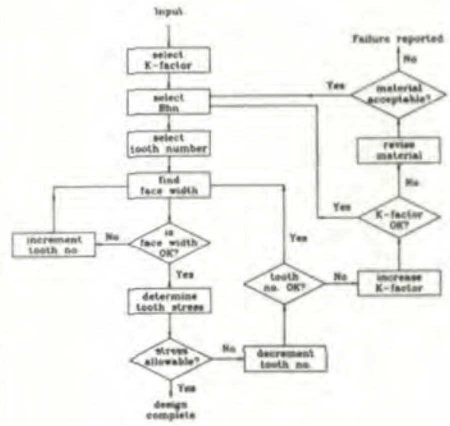

Fig. 2 – Flow chart of the spur gear tooth design program.

database (Fig. 1). The calculated bending strength and surface durability are checked against the allowable values. If the check is successful, the design is accepted; otherwise, the selected number of teeth is decreased until a stronger gear design is achieved, or the minimum allowable number of teeth is exhausted. If the latter occurs, then a stronger material is selected, and the design process gets underway. However, if the material data base is depleted, then failure of the design is reported. At this point the user is advised to consider an alternative gear drive system, such as helical or epicyclic arrangements. The flow of information in the spur gear design methodology is shown in Fig 2.

Table 3

Parameter	Reference [3]	Application Driven
Pitch (P)	4	4
Diameter (d) [Inches]	4.5	4.5
Face width (F) [Inches]	3.41	3.38

Example and Discussion

The example is a comparison between the textbook approach described in Reference 2 and the application-driven program proposed in this article. The example essentially states the following:

> "A pair of 4:1 reduction spur gears is desired for a 100-hp 1120-rpm motor. The gears are to be 20 full-depth with a clearance of 0.250/P and made of UNS G-10400 steel, heat-treated and drawn to 1000° F. Make an estimate of the required gear size."

In the above example, the authors[2] use a factor of safety of four and the material is predetermined. In contrast, the application-driven approach requires the user to identify the application environment only. For this particular example, a general class of high-speed drives is specified as the application environment for the spur gear design. Results for both approaches are shown in Table 3.

As can be seen from the table, both methods yield similar results. However, there are marked differences in the approach. Shigley and Mitchell[2] assume a minimum pinion tooth number of 18 and a pre-established type of material. In the application-driven approach, the minimum tooth number is initially set to 15, and the program establishes 18 as a viable tooth number for the pinion. Also, the material is selected from a data base of available through-hardened steel. Note that this data base can be updated if necessary, and new types of material can

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(continued from page 42)

be added for any particular application. Hence, selection of proper material is an integral part of the gear tooth design program. Also, the gear tooth data obtained via the application-driven approach incorporates the strength and durability factors. These factors inherently carry a measure of safety to the gear tooth design process, and, therefore, introduction of a user-defined factor of safety becomes unnecessary.

This approach incorporates the expertise of the gear designers into the design routine. The expertise is obtained from handbook information and can be regularly updated if necessary. Consequently, no preconceived information about the type of material or the factor of safety is required to arrive at an acceptable design decision.

Conclusion

A new technique for computer-aided sizing of spur gears has been proposed. The method has successfully been implemented in a comprehensive parallel axis transmission design expert system.[5] The gear tooth design is application-drive and is based on the index of tooth loading (K-factor). Results show that the technique could lend itself to modular expansion to include other types of gear teeth designs or different application environments. Once the K-factor has been established for any new application environment, it can be stored as the basis for the gear tooth design for that particular application. This in turn, suggests applying a rule-based expert system architecture to continually update the knowledge base of application or K-factor values. An intelligent material data base search routine can also improve the options available for gear material selection.

This article presents a first step toward gear tooth design automation incorporating the expertise and experience of those whose endeavors led to the now established AGMA standards for the gear design. A logical next step in this process would be the inclusion of helical gear design and the expertise in determining whether a set of spur or helical gears are to be employed for a particular design environment. The goal is to merge the gear design techniques with the state of the art in computer-aided design methods, which include the expert knowledge as well as routine computational methods.

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