# GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

JULY / AUGUST 1993

Gear Expo PRE-SHOW ISSUE Gear Expo

CNC BEVEL GEAR GENERATORS

CONTACT ANALYSIS OF GEARS USING COMBINED FE & SURFACE INTEGRAL METHODS

EUROPEAN "X" FACTOR SYSTEM FOR AMERICANS

SPECIAL GEAR EXPO '93 COVERAGE

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## RANDALL PUBLISHING, INC.

1425 Lunt Avenue
P.O. Box 1426
Elk Grove Village, IL 60007
(708) 437-6604 Phone
(708) 437-6618 Fax

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# Gear Expo '93 — A Wise Investment

Gear Expo '93 — another trade show, another plea to send people and/or equipment out of town, away from the office or plant. Another bid to spend time, money, and effort. Oh, please! Hasn't anybody heard that these are the "lean and mean" '90s?

But wait. Don't write this show off as just more hype to no good purpose. Part of the key to success in this fiercely competitive time is not so much spending less, but spending wisely; and a trip for you and some of your employees to Gear Expo '93 could be time, money, and effort very well spent indeed.

The organizers have been sensitive to cost. Hence, locations have been chosen in the Midwest, the heart of the gear market. Travel times are reduced. For many in the industry the visit to Detroit can be a day trip.

Making wise use of resources involves more than cutting travel time and expenses. "The World of Gearing" makes the best *use* of time spent as well. Devoted exclusively to the gear industry, the show focuses on what attenders want to see, without forcing them to hike through halls of things they don't.

The Gear Expo is also a kind of information clearinghouse for gear information — another saving of valuable time and effort. Many exhibitors have technical people at their booths as well as sales and marketing staff to share information and answer questions about their products. I am reminded of a visitor from South America to the '91 Gear Expo who told me he had more of his questions answered in 15 minutes with the technical personnel at one of his supplier's booths than he had in six months of extensive correspondence prior to the show. Sometimes there really is nothing like "hands-on" experience.

This kind of information exposure goes beyond the exhibition hall. Running concurrently with and immediately following Gear Expo '93 are The Gear Manufacturing Symposium and The Fall Technical Meeting. These educational seminars contain valuable information for everyone from your newest employee to your most experienced engineer.

## PUBLISHER'S PAGE

*Gear Technology* will be at the show too. We are gratified to know that you, our readers, value the information service we provide. But it is important not just to let us know that we provide a valuable service to you (although we love compliments as much as the next person), but to let our advertisers know as well.

Many of them will also be at the Gear Expo (See page 11 for a list), so please remember to mention your appreciation of their support of *Gear Technology* when you visit their booths. Please also visit us at Booth #518. We always enjoy hearing from you either by phone or letter and welcome meeting you in person.

In the end, of course, every company has to decide for itself whether attendance at trade shows is worth what it costs. But if value received for dollars spent is one of your criteria for determining a good buy, sending your employees to Gear Expo '93 is a very wise investment indeed.

Michael Goldstein

Michael Goldstein,
Publisher/Editor-in-Chief





# CALENDAR

**JULY 12-15**

University of Cincinnati Center for Industrial Heat Treating Processes workshops. Clarion International at O'Hare, Chicago, IL. Subject matter includes induction heat treating, gas carburizing, quenching, and part distortion and residual stress analysis. For more information call Dr. A. H. Soni at the Center, (513) 556-2710 or fax (513) 556-3390.

**AUGUST 24-25**

SME Fundamentals of Gear Design & Manufacture. Introduction to spur gears, gear finishing, broaching, inspection, and more.

**AUGUST 26**

SME Clinic on Heat Treating & Hardening Gears. Both clinics at Embassy Suites – Livonia, MI. For more information, contact Mike Traicoff, (313) 271-1500.

**SEPTEMBER 14-16**

Ohio State University course on gear noise. The course will cover causes of gear noise, noise reduction, dynamic modeling, signal analysis and problem diagnosis, housing dynamics, and housing noise radiation. For more information contact Carol J. Bird (614) 292-3204 or Dr. Donald Houser at (614) 292-5860.

**SEPTEMBER 21-23**

ASME 7th International Cogen Turbo Power Congress and Exposition. Bournemouth International Centre, Bournemouth, UK. For more information contact the International Gas Turbine Institute, (404) 847-0072 or fax (404) 847-0151.

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- ❑ Gear Honing
- ❑ Gear Lapping
- ❑ Gear Schools
- ❑ Gear Testing
- ❑ Heat Treating
- ❑ Import Agents
- ❑ Manufacturer of Gears – Custom-Made
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Send this form with your Visa/MasterCard/Amex number or a check for $700.00 for each display ad you are running, along with your copy and any logos you wish included to **Gear Technology, 1401 Lunt Avenue, P. O. Box 1426, Elk Grove Village, IL 60007.**
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**Return this form to us by September 10 to ensure a space in the directory.**

# *Gear Expo '93 Promises to be the Biggest All-Gearing Trade Show Yet*

**"World of Gearing" to open at Detroit's Cobo Hall.**

**Gear Expo '93, "The World of Gearing,"** is scheduled for October 10-13 at Cobo Conference & Exhibition Center, Detroit, MI. Hours are 9 a.m. to 5 p.m. on all four days of the show.

The show, sponsored and organized by the American Gear Manufacturers Association, is devoted exclusively to the gear and gear products industries, and will be the largest in terms of floor space since the exhibition was first organized in 1986.

As of June 1, 80 companies from around the world had reserved booth space for the show. This collection of the major players in the industry provides a forum for gear manufacturers and suppliers to exhibit their products, and gives visitors the chance to make comparisons and ask questions of technical representatives right at the show. Products and processes on display include broaching, custom gears, cutting tools, finishing, forging, grinding, heat treating, hobbing, inspection, lubrication, milling, shaping, shaving, and testing.

An index of *Gear Technology* advertisers who are exhibiting at this year's expo can be found on the adjoining page.

Once again the AGMA Fall Technical Meeting and the Gear Manufacturing Symposium (GMS) will be held in conjunction with the show. The FTM will include presentations by global experts in gear technology on a variety of gearing subjects, including gear tooth finishing, distortion control and heat treatment, undercutting in worm gears, gear noise, and bending fatigue. It will be held on October 14-15, the two days following Gear Expo '93.

The Gear Manufacturing Symposium, which will be held concurrently with the expo itself, is a program covering some of the basics of gearing for shop floor superintendents, foremen, factory managers, and new employees. It can serve as either an introduction to gear manufacturing or a refresher course.

Begun as a small tabletop exhibition, the Gear Expo is now a full-fledged trade show held every two years on a rotation schedule among several cities. Detroit, the oldest city in the Midwest, especially lends itself to this show. It is at the heart of America's manufacturing base and, as home to the "Big Three" U.S. automakers, produces millions of gearsets each year.

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

Transportation in the downtown area includes Detroit's elevated transportation system, The People Mover, plus more traditional buses, rental cars, taxis, and antique trolley cars. Cobo Center also has 2,200 parking spaces, plus an additional 3,000 at the Joe Louis Arena next door.

For more information about Gear Expo '93, the FTM or the GMS, contact AGMA Headquarters at (703) 684-0211. ■

**Booth #118**
**AGMA**
1500 King St., #201
Alexandria, VA 22314
(703) 684-0211

**Booth #621**
**American Metal Treating Co.**
1043 E. 62nd St.
Cleveland, OH 44103
(216) 431-4492

**Booth #601**
**American Pfauter L. P.**
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**Booth #345**
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San Diego, CA 92127-2139
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## *GEAR EXPO '93*
# *EXHIBITOR INDEX**

***Current as of May 25, 1993**

# AGMA Promises Bigger, Better Show in Detroit

**Gear Tech talks with AGMA Executive Director Joe Franklin about plans for Gear Expo '93 and beyond.**
**Nancy Bartels**

AGMA's Gear Expo '93 is expected to be at least 10% larger in terms of floor space than the '91 show, according to Joe Franklin, AGMA's executive director. "As of June 1, we have 80 exhibitors registered", he says.

But size is not the only thing to be improved at Gear Expo '93. "We've worked very hard, not just to get a booth sold, but to put a rational floor plan together," says Franklin. "In the past, the arrangement was like a pyramid. At the entrance were a lot of very large booths, and then as you moved toward the back and the food service area, the booths got smaller.

"What we've done is to get very large booths all over the floor. The result is rather like a retail shopping center with anchor stores. We will still have some major booths in the front, but there will also be some fairly large presentations at the back near the food service area. I think this reflects the recognition that at a show like this there are really no bad spaces. Attenders are going to visit all the booths."

AGMA is trying some other floor plan innovations as well. Franklin explains: "We have gone to exhibitors — say a hobbing company, a grinding company, and a measurement company — and said, 'You guys sell together in the marketplace; you mutually reinforce one another's equipment. Why don't you think about taking three booths next to each other at the show?' And many exhibitors have been very responsive to that idea. What you'll be able to see is not just a good set of equipment from one company, but a cross section of a whole system that can be installed. In a sense, we're trying to replicate the marketplace to some extent in a show environment."

AGMA has worked to improve the show in other respects as well. To help attenders make the most of their visit to Detroit, The Gear Manufacturing Symposium will be held concurrently with the show, and the Fall Technical Meeting will be linked to the Gear Expo as well. It is scheduled for October 14-15, the two days immediately after the show.

In addition, AGMA is attempting to schedule several international standards meetings for Detroit that same week in order to attract more overseas visitors to the show and the FTM. "We'd like to let the international visitors spend some time at the show and the FTM, and then go to their meetings. With airfares the way they are now, people usually try to stay for the whole week."

Arrangements also have been made with the Cobo Hall management to greatly upgrade the food service at the show. Says Franklin, "The problem is in Detroit, there just aren't a lot of restaurants near Cobo Hall. We would like people to be able to eat at the hall if they want to."

Even though the '93 Gear Expo is still in the planning stages, AGMA is already looking ahead to future shows. The 1995 show will be in Indianapolis. After that, the show will probably alternate between Detroit and other cities.

"Sometimes," says Franklin, "there are good problems to have. Our good problem is that the expo is no longer a very small show. It's not a gigantic show — you don't get lost in it, but it's too big for a lot of places that we'd like to hold it."

Franklin sees three other possible areas for future expansion of the show. "One trend we have to recognize and respect is the emergence of alternative materials. Right now most of our show is dedicated to steel or other metal gears, but I anticipate that in the 1995 show we will have pavilions for both plastic and powder metal gears," he says.

Another area of expanded interest is in research and development. "We know that people are doing research in the gear industry all the time. What we would like to do is encourage the universities and the third party research centers to come into a pavilion at Gear Expo and strut their stuff," says Franklin. "This would be a marvelous opportunity for these institutions to tell the engineering community what their capabilities are."

The final area of expansion AGMA is considering is in further development of the expo as a product show. "We have some gear companies already at the show," says Franklin, "but we haven't developed the show specifically for them. But I think many gear companies are sensing the growing willingness to outsource for supplies. Ultimately, we'd like Gear Expo to be one-stop shopping for people manufacturing and using gears." ■

# Gear Tip Chamfer and Gear Noise; Surface Measurement of Spiral Bevel Gear Teeth

**William L. Janninck**
**Robert E. Smith**

**Could the tip chamfer that manufacturing people usually use on the tips of gear teeth be the cause of vibration in the gear set? The set in question is spur, of 2.25 DP, with 20° pressure angle. The pinion has 14 teeth and the mating gear, 63 teeth. The pinion turns at 535 rpm maximum. Could a chamfer a little over 1/64" cause a vibration problem?**

*Bill Janninck replies:* A 1/64" chamfer on the tips of gears that are as coarse as 2.25 DP would certainly not be considered a cause for gear vibrations. The chamfer is more likely to have a positive effect, such as the removal of sharp edges or burrs, than a negative one. There are other more likely causes for gear vibration or noise.

The usual first step followed in investigating any gear problem is to do some computations or computer modeling to assure that basic geometric requirements are met. The engineer checks for interferences, that suitable root clearances and backlash are present, and that the gear set has a contact ratio of 1.0 or more. The latter assures that there are proper conditions for a smooth transfer of contact from one gear tooth pair to the next. If the contact ratio were less than 1.0, the smooth passage of motion from pair to pair would be broken and could cause some impacting and vibration.

Since the details of the tooth depth system were not given, we investigated three cases.

- Fully standard gears using the 2.25" whole depth basic rack with 1.0" addendum.
- 25% long addendum pinion and 25% short addendum gear using the same basic rack as above.
- Fully standard stub depth gears with 1.8" whole depth and 0.8" addendum system.

Two sets of computations were made for each of these cases, one with no tip chamfer, and one with not a 1/64" tip chamfer, but rather a 1/16" tip chamfer, on both pinion and gear. The contact ratios resulting are shown in Table I.

In no case is the contact ratio below 1.0, even on the shallower depth stub teeth. Geometrically the design is good and, as mentioned before, the tip chamfer is not a likely cause for vibrations.

Another source of vibration is the gear and gear box support, where bend-

**Table I — Contact Ratio Comparison**

| | No Chamfer | 1/16" Chamfer |
|---|---|---|
| Standard Full Depth | 1.54 | 1.42 |
| 25% Long & Short Addendum | 1.56 | 1.36 |
| Standard Stub Depth | 1.34 | 1.13 |

## SHOP FLOOR

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

**William L. Janninck**
*is a gear and tool design consultant. He has been involved with gears and gear manufacturing for 45 years, 40 of them with Illinois Tools — ITW, Inc. He is the author of numerous articles on gear-related topics.*

**Robert E. Smith**
*is the principal in R. E. Smith & Co., Inc., gear consultants in Rochester, NY. He has over 40 years' experience in gearing methods, manufacture, metrology, and research.*

ing and torsional elasticity in shafting, the gear box, or bearings may be stimulated by gear tooth action.

A more likely cause for vibrations is in the functioning of the gear teeth themselves. The smooth engagement and transfer of contact and load to successive tooth pairs requires a certain degree of accuracy in tooth profile, tooth spacing, pitch line runout, and tooth parallelism. There is no easy way to predict the exact tolerances to use, but AGMA 390.03 gives some suggested values. With the pinion running at 535 rpm, the gear set pitch line velocity is 870 sfpm. AGMA Q8 or Q9 is suggested. Specific elemental tolerances can be taken from ANSI-AGMA 2000-A88. For the pinion,

| | |
|---|---|
| Profile | .0015 - .002 |
| Runout | .005 - .006 |
| Spacing | .001 - .0013 |
| Parallelism | .0008 - .0010 |

If the problem is serious enough, then actual inspections must be made to see just where the gear quality levels lie. Usually, the first thing checked is the profile, as it is more subject to manufacturing variations. A high profile near the tip is also cause for concern. The profile should preferably have a relieved tip and a high area near the pitch line. Next to be checked is spacing, where the errors can cause impacting, and then parallelism. Runout is the easiest to measure and is usually readily correctable.

At times it is worthwhile to examine the contact pattern occurring from running the gears together. Tip contact or any edge contact could indicate a quality problem.

**Is equipment available for measuring surface finish on spiral bevel gear teeth (Q12)? The unit we have works best on flat straight surfaces and measurement taken on spiral bevel gears is questionable. Also, I am looking for reports or documents on surface finish requirements. (Our 30Ra has been "upped" to 32Ra by the manufacturer, so visual comparator type inspection can be used. I feel 25-30 Ra should be maintained.)**

*Bob Smith replies:* First, there is nothing that relates surface finish requirements to a given AGMA quality level, such as Q12. The only relationship is that the finish must be good enough to prevent interference with the measurement of dimensional requirements of spacing, profile, etc. However, finish is related more to surface durability and scoring. You will find a discussion of this factor in AGMA 2001, *Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth, Appendix A.* The bevel gear rating standard, AGMA 2003, and the *Bevel Gear Design Manual*, AGMA 2005, vaguely discuss surface condition factor, $C_t$. Another reference is Dudley's *Gear Handbook*, Second Edition, 1991, page 15.21.

Measurement of surface finish on spiral bevel gear teeth can be difficult,

# SHOP FLOOR

but not impossible. It is somewhat limited by diametral pitch. As with all surfaces, it should be measured at right angles to the lay of the finish pattern. This means roughly tip to root rather than toe to heel. This limits the amount of stylus travel. Generally, the length of stylus travel should be five times the cutoff wavelength. The most typical cutoff wavelength is .030". The only AGMA document on surface finish, AGMA 118.01 (now undergoing revision), defines roughness wavelength as 1/20th of tooth length. This probably puts wavelength in the proper perspective to tooth size. Most instruments have a limited selection of cutoff wavelengths, such as .003", .010", .030", and .100".

Another problem when trying to assess surface texture in gear teeth is the large curvature of the tooth form. Form is not of interest for surface texture, so it must be removed by some electronic or mechanical means. This can be done with a "skid" type instrument. This, however, sets a limit on how fine a diametral pitch can be checked. Another technique is to use a smooth curved datum surface to guide the stylus. This might be an optical circular or elliptical curved element that is adjusted to approximate the curve of the gear tooth. The more modern measuring instruments have a stylus with a very long travel (.040") normal to the surface. This can move across surfaces with large form error, but still digitize the surface roughness data. With PC-based analysis of data, it is then possible to quantify the roughness data with the desired cutoff wavelength. This type of equipment is well-suited to measurement of gear tooth surfaces.

Another way to get around the problem of getting into small gear tooth slots is to use a replica material. This is a filled two-part epoxy material designed specifically for surface finish measurement. It can faithfully replicate a surface down to six micro-inches Ra. The replica is removed from the slot for measurement. The resulting measurement is a negative of the actual surface, but roughness data will be the same.

Finally, you show concern that a tolerance limit was changed from 30 to 32Ra. The measurement of surface finish is not very exacting. Do two people measure at exactly the same spot? A difference of two micro-inches out of thirty is not very alarming and shouldn't make any difference in the functionality of the gear. As an example of this, the parameter for measurement was changed many years ago from RMS to Ra. Most people used the same numerical tolerances without conversion. Some people still use the parameter terms interchangeably, but with the same numerical values. Many pieces of current literature still use RMS, even though it is not equivalent to a surface with the same value for Ra. ■

# CNC Bevel Gear Generators and Flared Cup Gear Grinding

**Theodore Krenzer**
**The Gleason Works, Rochester, NY**

New freedom of motion available with CNC generators make possible improved tooth contact on bevel and hypoid gears. Mechanical machines by their nature are inflexible and require a special mechanism for every desired motion. These mechanisms are generally exotic and expensive. As a result, it was not until the introduction of CNC generators that engineers started exploring motion possibilities and their effect on tooth contact.

This article covers the exploitation of new motion freedoms to improve tooth contact patterns on gear sets manufactured by a face-milling duplex process, a manufacturing method where both flanks are completed in a single operation.

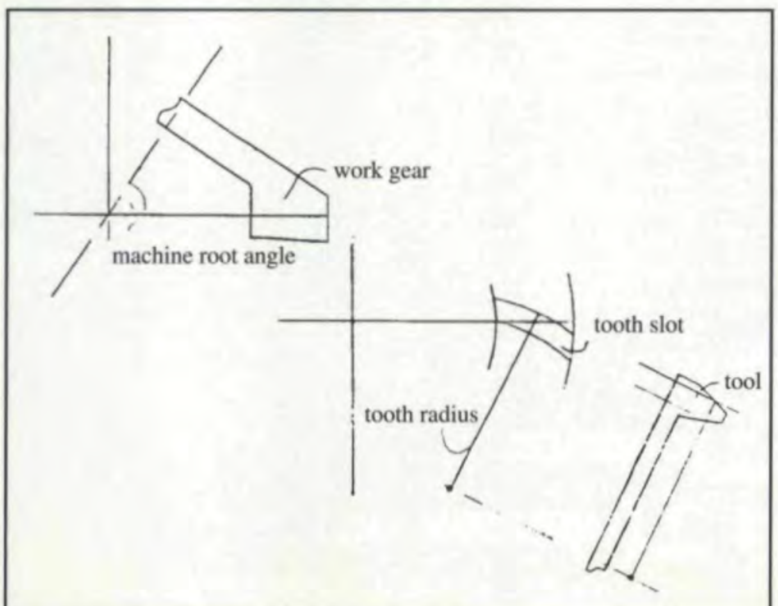


**Fig. 1 — Conventional face milling duplex setup.**

Starting with a brief background of the flared cup process, the article proceeds to describe the possible linear and angular motion variations and their effect on the gear tooth surface. The article concludes with the use of Tooth Contact Analysis (TCA) to evaluate the enhancement of this duplex process made possible by applying these motions.

**Flared Cup Process**

When cutting face milled gears using this operation, the cutter is positioned relative to the gear blank, so that the correct spiral and pressure angles will be produced. The gear blank is held stationary, and a tooth slot is form-cut by infeeding the cutter. The part is indexed one pitch, and the process is repeated. See Fig. 1. When the cutter is replaced by a grinding wheel, contact exists over the entire length and depth of the tooth surface. Heat buildup results, causing a tendency for surface damage because ofburning. The flared cup grinding process is used to overcome this problem.

The flared cup process uses a wheel which is tilted out of the work (30°s of tilt is commonly used). The outside wheel surface has a normal radius of curvature less than the conventional tool, and the inside wheel surface has a normal radius of curvature greater than the conventional tool. Line contact exists between the work and the wheel. The wheel is positioned relative to the gear blank so that at the calculated mean position on the gear tooth surface, the correct spiral and pressure angles are produced. The tilted axis of the wheel is in a plane normal to the tooth surface. In achieving this setup the tilted axis is offset from the conventional tool axis. The tooth is swept out by rotating the flared wheel about the axis of the conventional tool axis. See Fig. 2.

As the wheel is dressed its radii change, which requires compensating machine changes to maintain the proper tooth geometry. Making these setting changes manually on mechanical machines is a problem. Tooth geometry often varies from part to part. Full CNC machines, where wheel radius can be accurately determined, are programmed to automatically compensate for wheel size changes resulting from dressing.

Wheel life is a function of the radius change which occurs as a result of dressing. Over the useful life of a wheel, the relative curvature decreases between the convex tooth surface and the inside wheel surface and increases between the concave tooth surface and the outside wheel surface. Although the final tooth surface is produced by line contact, at any instant surface contact exists between the wheel and work in proportion to the depth of grind. The contact area is dependent on the relative curvature between the wheel and work and the variation in the contact area between the two tooth sides is used to determine wheel life. As a rule of thumb, good results are obtained when the difference in contact area between the two sides of the tool does not exceed the ratio of 2:1. Fig. 3 shows sections of a tooth and grinding wheel at three stages of wheel life — ideal, new wheel, and spent wheel.

### New Freedoms

Three angular and three linear motions define the relative motions that can exist between any two bodies, in this case between the flared cup tool and the work gear. One of the angular freedoms is used to sweep out the tooth surfaces; therefore, effectively only two angular freedoms are available for contact pattern control.

At any instant in sweeping out the tooth surfaces, the CNC generator has the capability to change the relative orientation between the contact line and the gear tooth. Motions to achieve a change could be defined in any number of reference systems. For this case all of the motions are defined based on the instantaneous radial plane; that is, the plane containing the conventional tool axis and the radial line to the mid-height point on the contact line.

The freedoms are defined as follows:

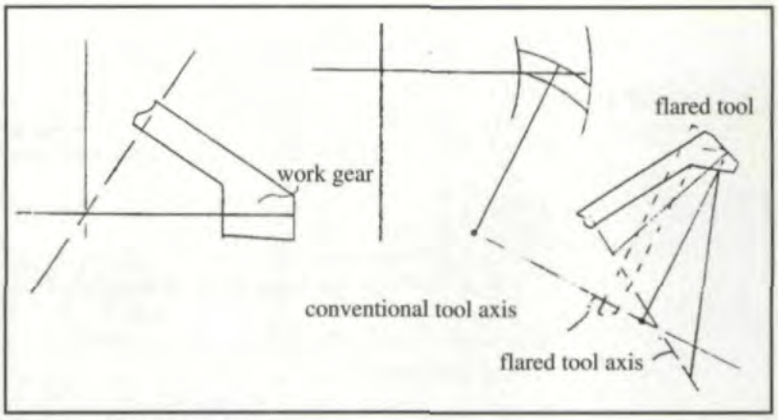


**Fig. 2 — Flared cup setup.**

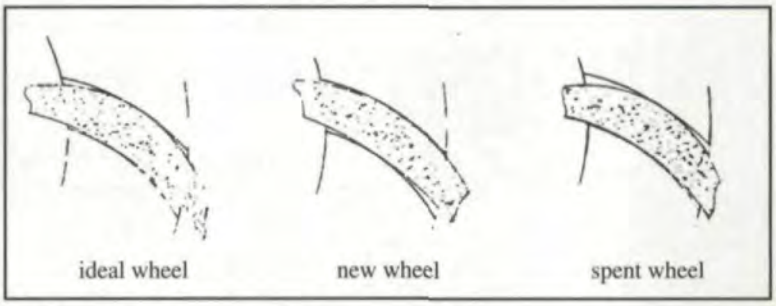


**Fig. 3 — Contact area variation.**

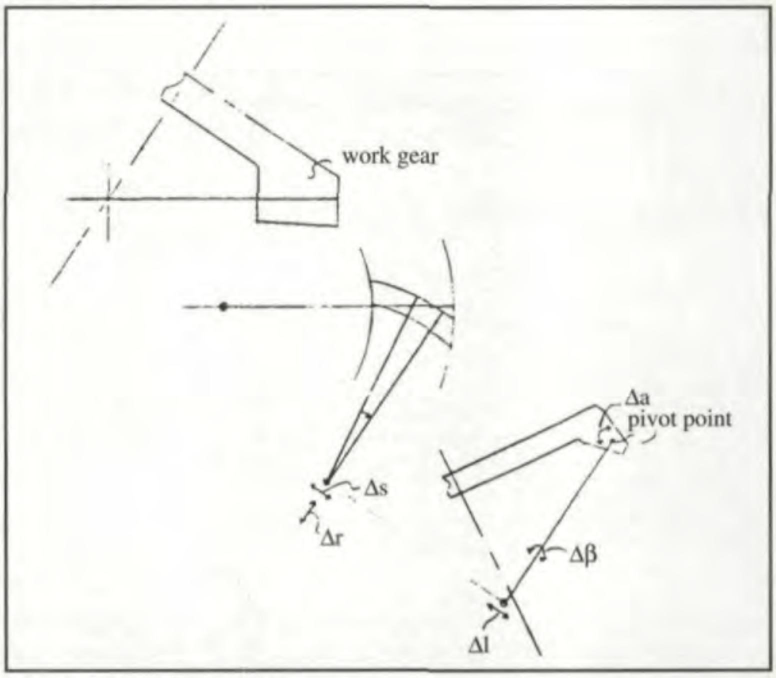


**Fig. 4 — Flared cup motions.**

1. Rotational motion in the instantaneous radial plane;

2. Rotational motion about the instantaneous radial line;

3. Linear motion along the conventional tool axis;

4. Linear motion along the instantaneous radial line;

5. Linear motion perpendicular to the instantaneous radial line.

Fig. 4 is a sketch of a flared cup setup

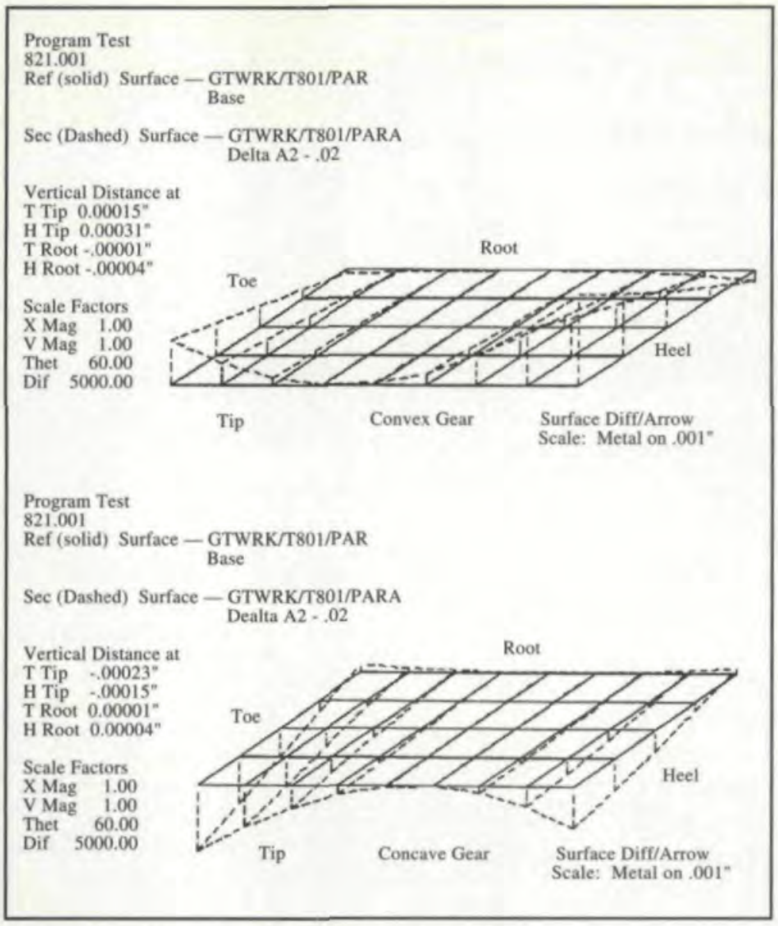


**Fig. 5 — Topology graph of second order $\Delta a$ change.**

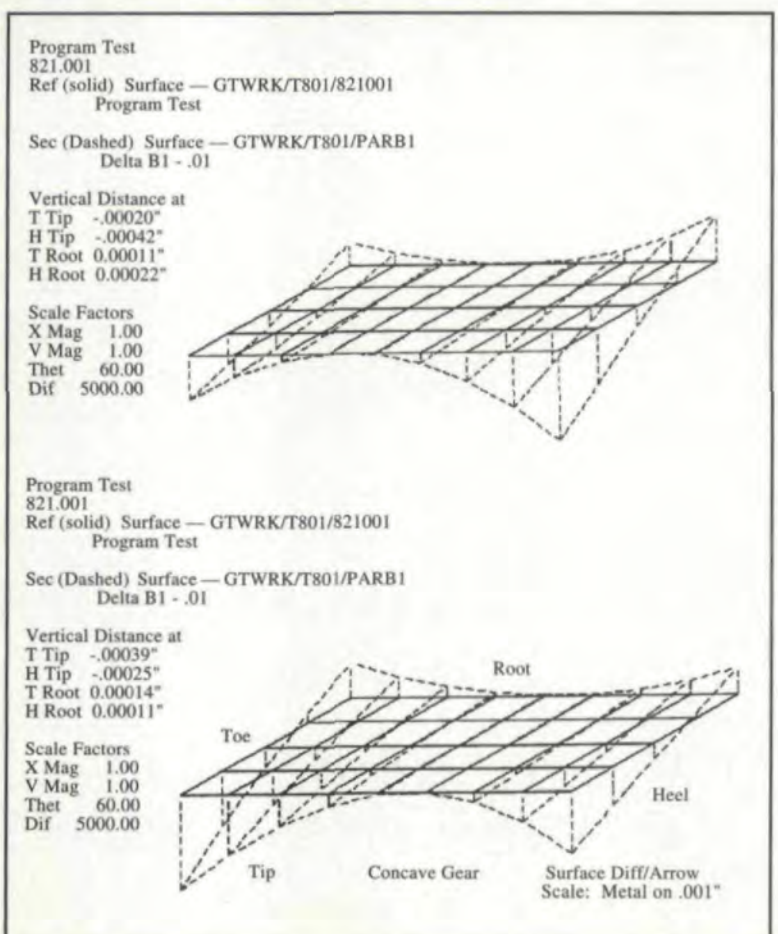


**Fig. 6 — Topology graph of second order $\Delta\beta$ change.**

showing these motions. The angular motions pivot about a point at mid-tooth depth and mid-slot width. A timed relationship exists between the motions and the angular position of the wheel as it is swept through the tooth slot. Although a number of functions could be used to define the relationships, polynomial expressions were selected.

***Radial Tilt***. This angular motion is a tilting of the tool in the instantaneous radial plane as the tooth is swept out. The effect is to change the pressure angle on both flanks of the tooth as the grind line moves from the tooth center section. The change increases the pressure angle on one flank, and decreases the pressure angle on the other flank as compared with the conventional tooth. At any tool phase angle position designed by $\Delta a$, the radial tilt of the tool is given by:

$$\Delta a = A_1\,\Delta\alpha + A_2\,\Delta\alpha^2 + A_3\,\Delta\alpha^3 + A_4\,\Delta\alpha^4$$

where $A_1$, $A_2$, $A_3$, $A_4$ are the coefficients that control the motion.

Fig. 5 schematically illustrates the change in surface topology on the convex and concave flanks of a gear tooth. The solid lines represent the baseline surface, and the dashed lines represent the surface resulting from a second order change in $\Delta a$. It can be seen that metal is removed on each side of the center section at the bottom of the convex flank and at the top of the concave flank of the gear teeth. The opposite effect occurs at the top of the convex side and at the bottom of the concave side.

The $A_1$ coefficient produces a velocity in the normal direction at $\Delta\alpha = 0$, the setup must be altered to accommodate the velocity when this coefficient is used.

***Tangential Tilt.*** This angular motion is a tilting of the tool around the instantaneous radial line as the tooth is swept out. Again the effect is to change the pressure angle on both flanks of the tooth as the grind line moves from the tooth center section. In this case the pressure angle is increased or decreased on both flanks, as compared with the conventional tooth. At any tool phase angle position designated by $\Delta$, the tangential tilt of the tool is given by:

$$\Delta_\beta = B_1\,\Delta\alpha + B_2\,\Delta\alpha^2 + B_3\,\Delta\alpha^3 + B_4\,\Delta\alpha^4$$

where $B_1$, $B_2$, $B_3$, $B_4$ are the coefficients that control the motion.

Fig. 6 schematically illustrates the change in the surface topology due to a second order change in $\Delta\beta$. It can be seen that metal is removed on each side of the center section at the top of both tooth flanks. The opposite effect occurs at the bottom of both tooth flanks.

***Axial Motion***. This linear motion is one that has been described previously in Refs. 1 and 2. It is an advance or retreat of the work along the conventional tool axis. The effect is to modify the surface topology in the same manner on both tooth flanks. More metal is either removed or left on the tooth flanks in the lengthwise direction than in conventionally formed teeth. At any tool phase angle position designated by $\Delta\alpha$, the change in position of the tool relative to the work in the direction of the cutter axis is given by:

$$\Delta l = L_1\,\Delta\alpha + L_2\,\Delta\alpha^2 + L_3\,\Delta\alpha^3 + L_4\,\Delta\alpha^4$$

where $L_1$, $L_2$, $L_3$, $L_4$ are the coefficients that control the motion.

Since the $L_1$ coefficient produces a velocity in the normal direction at $\Delta\alpha = 0$, the setup must be altered to hold spiral and pressure angle at the mean point when this coefficient is used.

Fig. 7 schematically illustrates the change in surface topology due to a second order change in $\Delta l$. It can be seen that metal is left on at the inside and outside of both tooth flanks.

***Radial Motion.*** This motion is a movement of the tool along the instantaneous grind radius between the tool and the work gear. The effect is to modify the surface topology in the opposite manner on the tooth flanks. Metal is removed on one flank and metal is left on the other, unlike the case of conventionally formed teeth. At any tool phase angle position designated by $\Delta\alpha$, the change in position of the tool relative to the work in the instantaneous radial direction is given by

$$\Delta r = R_1\,\Delta\alpha + R_2\,\Delta\alpha^2 + R_3\,\Delta\alpha^3 + R_4\,\Delta\alpha^4$$

where $R_1$, $R_2$, $R_3$, $R_4$ are the coefficients that control the motion.

Since the $R_1$ coefficient produces a velocity in the normal direction at $\Delta\alpha = 0$, the setup must be altered to accommodate the velocity when this coefficient is used.

Fig. 8 schematically illustrates the change in surface topology, due to a second order change in $\Delta r$. It can be seen that metal is removed on the convex side and left on the concave side on each side of the tooth center section.

***Tangential Motion***. The tool can also be moved in a direction perpendicular to the instantaneous grind radius. The effect is to modify the surface topology in the opposite manner

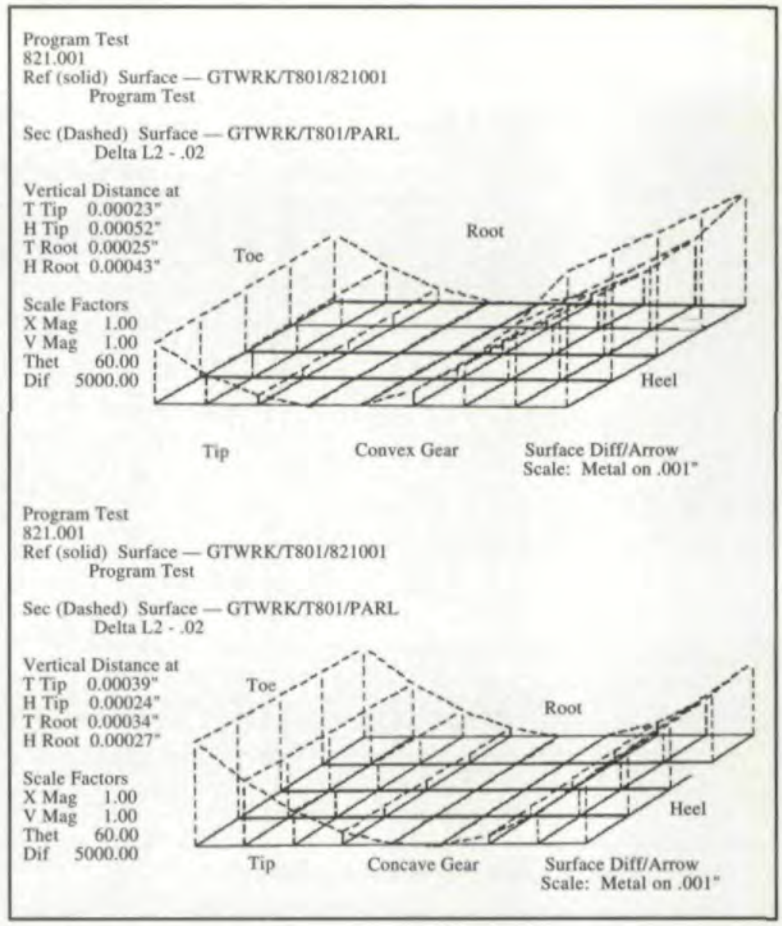


**Fig. 7 — Topology graph of second order $\Delta l$ change.**

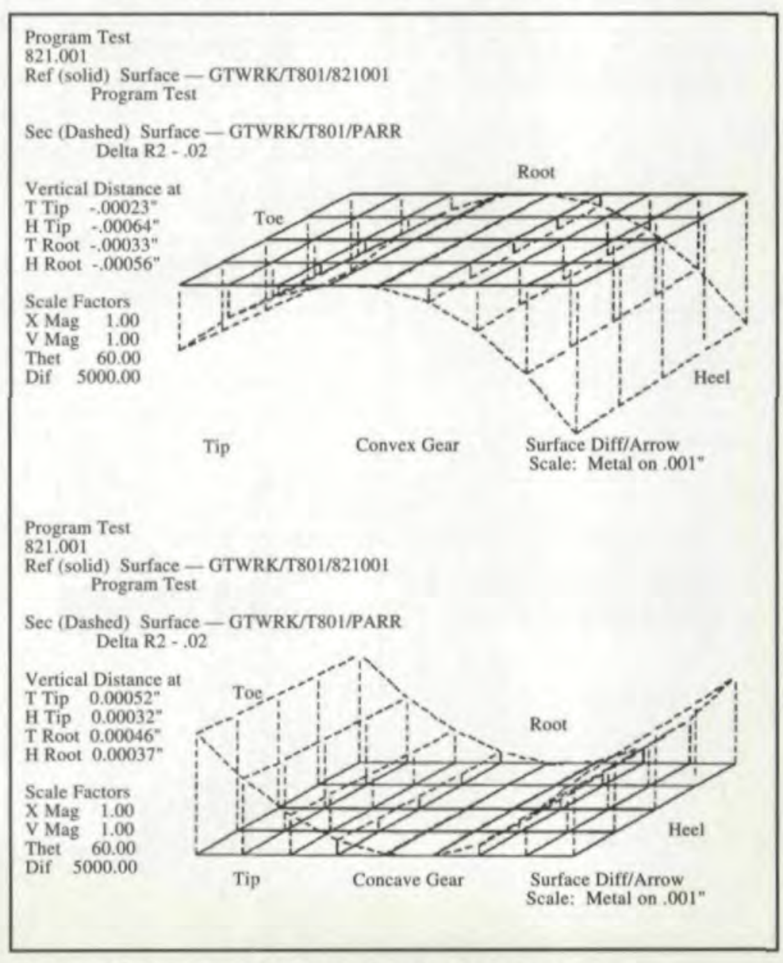


**Fig. 8 — Topology graph of second order $\Delta r$ change.**

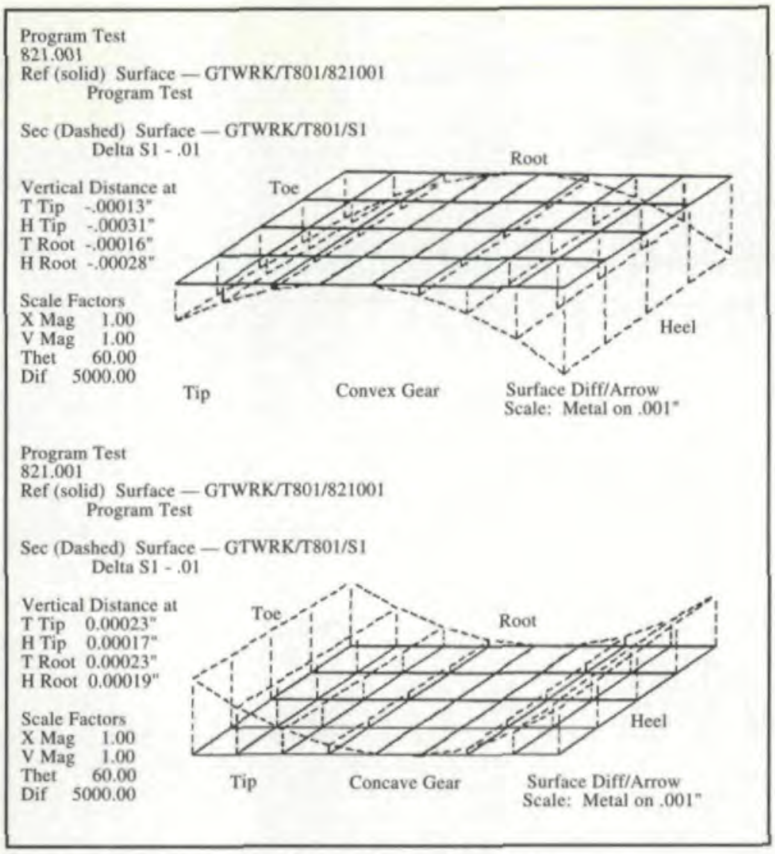


**Fig. 9 — Topology graph of second order Δs change.**

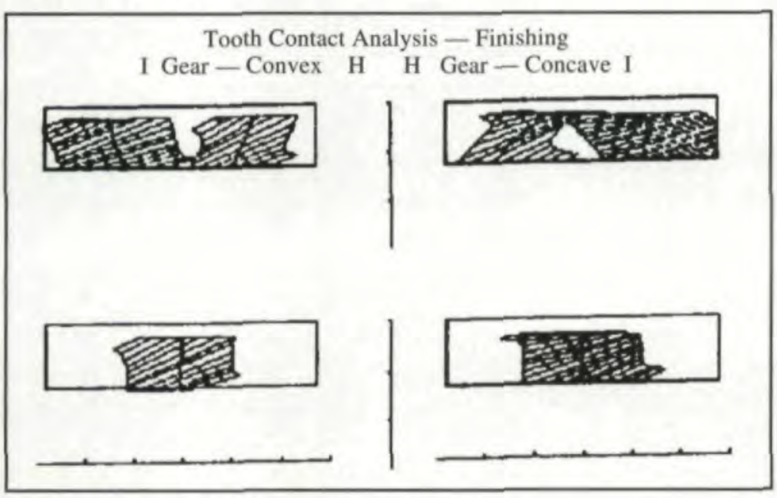


**Fig. 10 — Baseline TCA.**

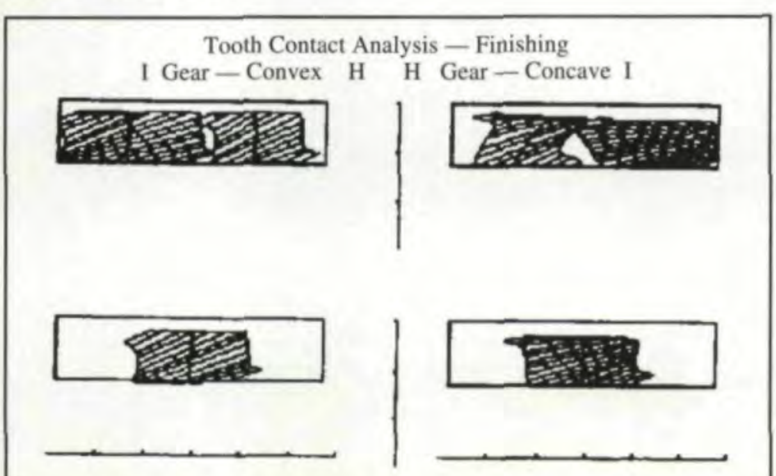


**Fig. 11 — TCA of second order Δα change.**

on the tooth flanks. It has a effect similar to the effect described above for a change in radial motion. At any cutter phase angle position designated by $\Delta\alpha$, the change in position of the tool relative to the work, in a direction perpendicular to the instantaneous radial is given by:

$$\Delta s = S_1\,\Delta\alpha + S_2\,\Delta\alpha^2 + S_3\,\Delta\alpha^3 + S_4\,\Delta_\alpha\alpha^4$$

where $S_1$, $S_2$, $S_3$, $S_4$ are the coefficients that control the motion.

Fig. 9 schematically illustrates the change in surface topology due to a second order change in Δs.

### Tooth Contact Analysis (TCA)

A second illustration of the five motion freedoms is shown using TCA. To aid in the comparison, the same job is used, and the effect of second order changes of the same magnitude are evaluated. Fig. 10 is the baseline TCA. It represents a conjugate gear set with only lengthwise mismatch. Fig. 11 is the TCA of the radial tilt change. Fig. 12 is the tangential tilt change. Fig. 13 is the axial motion. Fig. 14 is the radial motion. Fig. 15 is the tangential motion.

### Duplex Enhancement

The face milling duplex process is successfully used in many applications; in particular on fine pitch jobs and jobs where the contact pattern is enhanced by lapping. When grinding is the final finishing operation, the desired contact pattern length needed for most automotive applications cannot be achieved easily. Often a contact pattern, where the contact length varies from top to bottom, called a diamond pattern, results. This has been a factor in limiting the success of grinding as the final operation in the manufacture of hypoid gear sets for land applications. Enhancement of the duplex process by exploiting the new motion freedoms should make grinding more attractive as a final finishing process because of the ability to develop contact patterns with a wide range of characteristics.

Typically, gear sets used in automotive and

| TABLE I — BLANK DATA | SMALL AUTOMOTIVE | | |
|---|---|---|---|
| | Inch | | MM |
| Pitch/Module | 6.154 | | 4.13 |
| Pitch Diameter | 7.638 | | 194.01 |
| Number of Teeth | | 12/47 | |
| Face Width | 1.300 | | 33.02 |
| Offset | 1.250 | | 31.75 |
| Spiral Angle | | 48° | |
| Cutter Radius | 3.000 | | 76.20 |

truck applications cover the range from two to seven diametral pitch. Three sets within this range were designed using the standard duplex method plus the new motions. Blank data for the sets is given in Table I.

***Small Automotive*** — When the diametral pitch is six or higher, the need for added flared cup motions can be questioned. Fig. 16 is a TCA comparison of the duplex job designed without added motions on the left and with added motions on the right. The jobs were designed with very little transmission motion variation. Both designs are similar. Substantial pattern length was obtained without the introduction of flared cup motions. However, even for this case, greater pattern length was achieved on center, while limiting the length when the contact moves to the inside or outside of the blank. The long center contact is beneficial when gear noise is a concern.

***Large Automotive*** — For coarser diametral pitches, the benefits that can be achieved are more easily seen. Fig. 17 is a TCA comparison of the large automotive set designed without added motions on the left and with added motions on the right. The standard development is reasonably acceptable, and enhancement of the contact pattern as a result of the lapping could result in a set that would be very acceptable relative to noise quality. However, if the development is for a final finish grind, the contact pattern has more than the desired lengthwise mismatch.

Lengthening the pattern by conventional duplex methods would result in an unacceptable diamond pattern.

The introduction of additional motions can substantially increase the on-center pattern length while controlling the diamond condition. Also, note that pattern length at the toe and heel is held to a reasonable length to maintain adjustability.

***Large Truck*** — For diametral pitches in the range of two, the benefits are dramatic. Fig. 18

| LARGE AUTOMOTIVE | | | TRUCK | | |
|---|---|---|---|---|---|
| Inch | | MM | Inch | | MM |
| 4.178 | | 6.08 | 2.226 | | 11.41 |
| 10.500 | | 266.70 | 16.625 | | 422.28 |
| | 11/41 | | | 7/37 | |
| 1.600 | | 40.64 | 2.000 | | 50.80 |
| 1.500 | | 38.10 | 1.750 | | 44.45 |
| | 48° | | | 47° | |
| 3.750 | | 95.25 | 6.000 | | 152.40 |

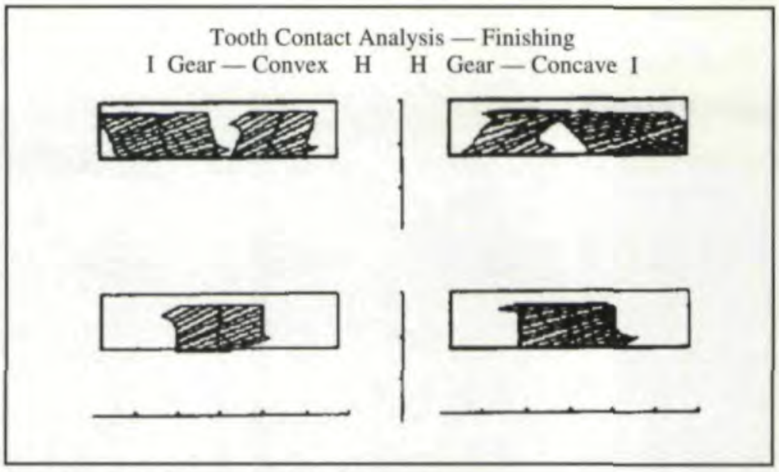


**Fig. 12 — TCA of second order Δβ change.**

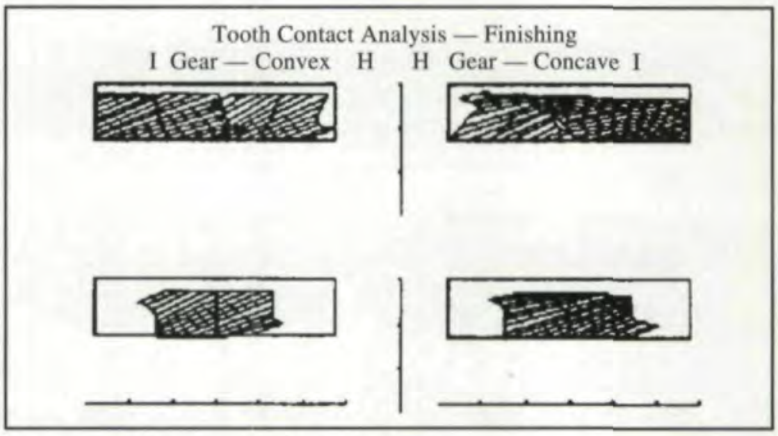


**Fig. 13 — TCA of second order Δ*l* change.**

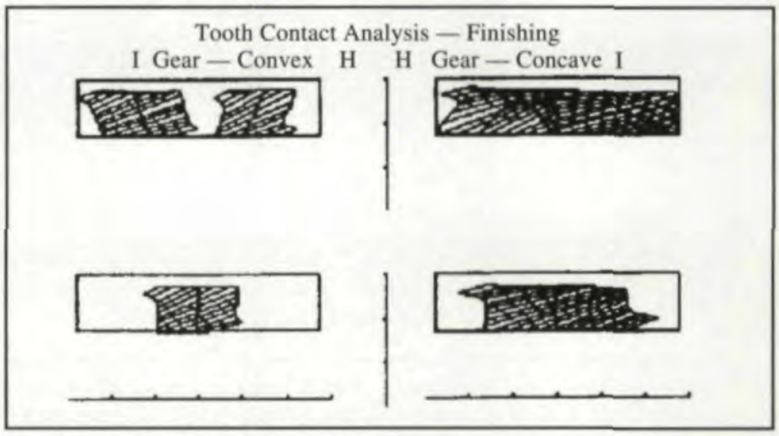


**Fig. 14 — TCA of second order Δr change.**

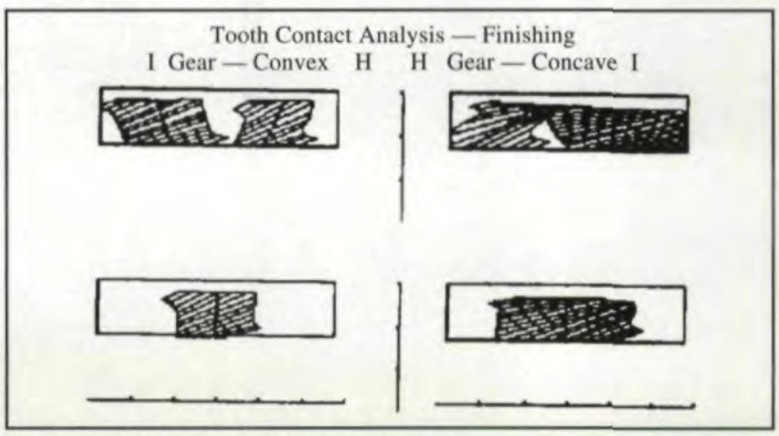


**Fig. 15 — TCA of second order Δs change.**

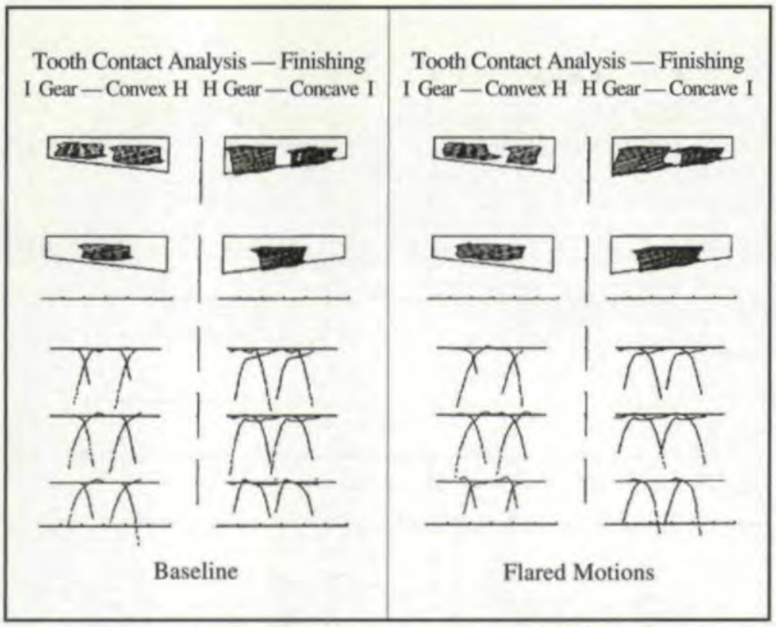


**Fig. 16 — TCA of comparison of small automotive duplex design.**

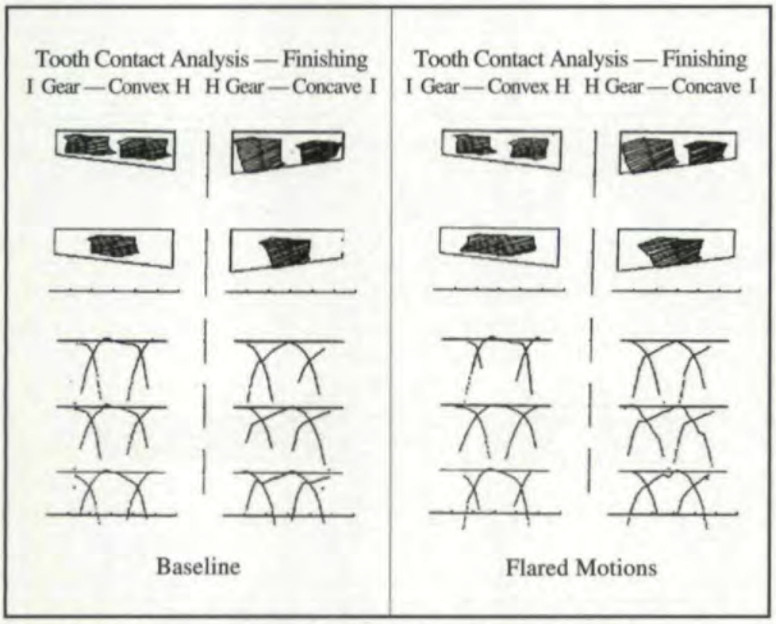


**Fig. 17 — TCA comparison of large automotive duplex design.**

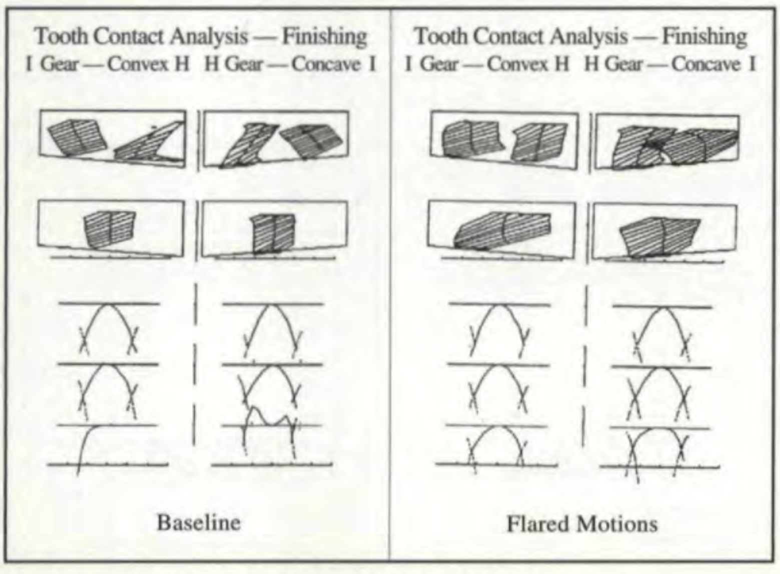


**Fig. 18 — TCA comparison of large truck duplex design.**

is a TCA comparison of the large truck job without added motions on the left and with added motions on the right. Pattern length on the standard development is short. Typical bias-in at the toe and bias-out at the heel contact patterns exist. Any further attempt to lengthen the pattern by conventional means would cause severe diamond problems. Note that more transmission motion variation was introduced into this design on the assumption that the set is more highly loaded and, as a result, requires increased adjustability.

With the added motions, the lengthwise pattern was increased at the central position, the diamond patterns at the toe and heel were controlled, and the lengthwise mismatch at the toe and heel were held.

## Summation

The article presented a theoretical description of the freedoms available on full CNC generators and their application to the flared cup gear grinding process. Surface topology and TCA were used to graphically define the effects of motion variation on the tooth surfaces. Finally, the application of the motion freedoms to enhance the flared cup duplex process was demonstrated using TCA.

In automotive and truck applications the amount and distribution of the mismatch between mating surfaces has a critical effect on sound quality. Theoretically, the flared cup process combined with a full CNC hypoid generator offers the motion freedoms that provide the necessary mismatch control for the duplex process allowing both pinion and gear members to be finished ground in one operation. ■

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# Contact Analysis of Gears Using a Combined Finite Element and Surface Integral Method

**S. M. Vijayakar**
**Advanced Numerical Solutions, Columbus, OH**

**Donald R. Houser,**
**Ohio State University, Columbus, OH**

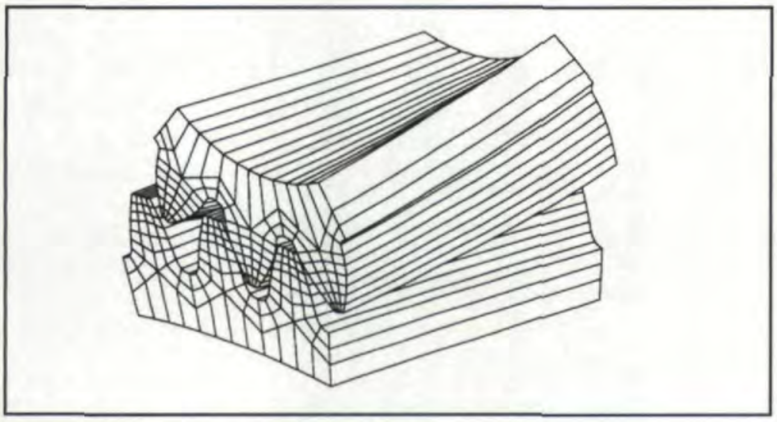


**Fig. 1 — Contact analysis of helical gears.**

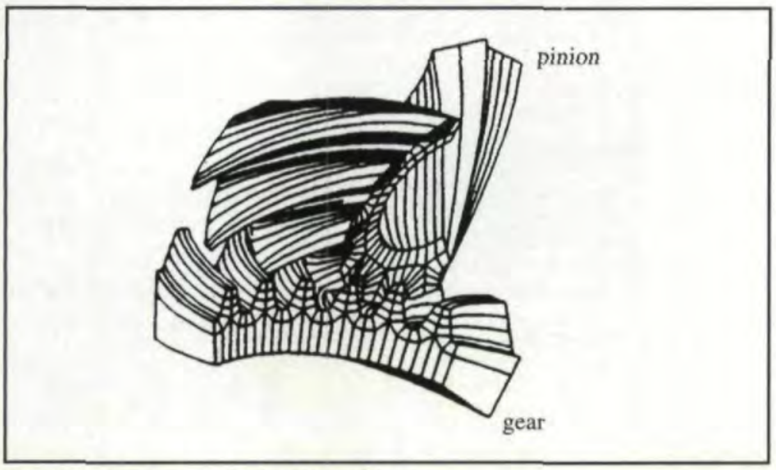



**Fig. 2 — Contact analysis of hypoid gears.**

**Introduction**

The complete and accurate solution to the contact problem of three-dimensional gears has been, for the past several decades, one of the more sought after, albeit elusive goals in the engineering community. Even the arrival on the scene in the mid-seventies of finite element techniques failed to produce the solution to any but the most simple gear contact problems.

The reasons for this are manifold. When gears are brought into contact, the width of the contact zone is typically an order of magnitude smaller than the other dimensions of the gears. This gives rise to the need for a very highly refined finite element mesh near the contact zone. But given the fact that the contact zone moves over the surface of the gear, one would need a very highly refined mesh all over the contacting surface. Finite element models refined to this extent cannot be accommodated on even the largest of today's computers. Compounding this difficulty is the fact that the contact conditions are very sensitive to the geometry of the contacting surfaces. General purpose finite element models cannot provide the required level of geometric accuracy. Finally, the difficulties of generating an optimal three-dimensional mesh that can accurately model the stress gradients in the critical regions, while minimizing the number of degrees of freedom of the model have kept the finite element method from being widely used to solve the complete gear contact problem.

Research in the mid- and late eighties showed that the gear contact problem was not unsurmountable, but required an approach that combined the strengths of the finite element method with those of other techniques, such as the boundary element and surface integral methods. Concepts from mathematical programming could be used to advantage in solving the contact equations. An innovative approach to the formulation of the finite elements themselves could go a long way towards solving the mesh generation and geometric accuracy problems. With the idea of incorporating the best of these and other technologies in mind, we began development of what is now CAPP (Contact Analysis Program Package) four years ago. It has evolved into a powerful collection of computer programs that provide the gear designer with an insight into the state of stress in gears that has thus far never been possible. Some of the features that CAPP supports are friction, sub-surface stress calculation, stress contours, transmission error, contact pressure distributions, and load distribution calculation.

Figs. 1-5 show examples of gear sets for which this process has been successfully used.

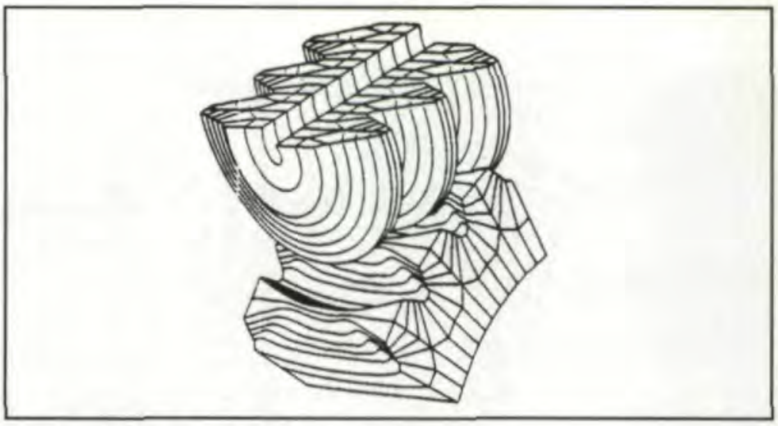


**Fig. 3 — Contact analysis of worm gears.**

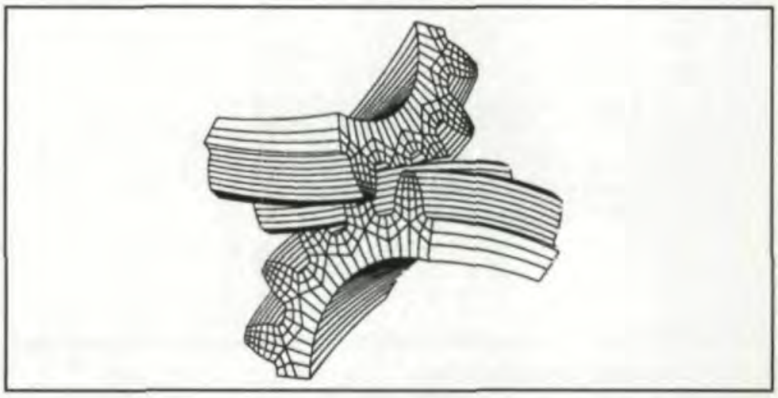


**Fig. 4 — Contact analysis of a 90° crossed axis external helical gear set.**

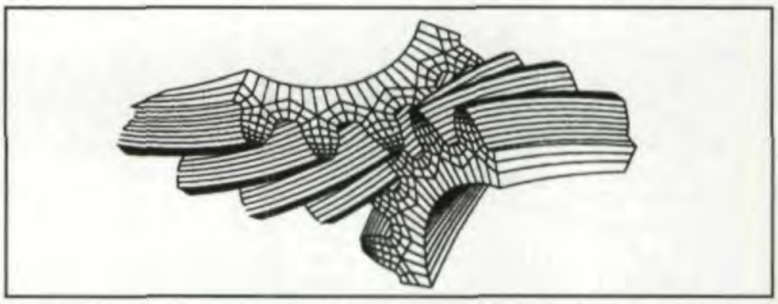


**Fig. 5 — Contact analysis of a 90° crossed axis external helical gear set.**

### Contact Analysis

In earlier studies of contact modeling (See Refs. 1, 2, 11, 12), a pure finite element approach was used to obtain compliance terms relating traction at one location of a body to the normal displacement at another location on the contacting body. It became apparent that in order to obtain sufficient resolution in the contact area, the size of the finite element model would have to be inordinately large. A finite element mesh that is locally refined around the contact region cannot be used when the contact zone travels over the surfaces of the two bodies.

Other researchers working in the tribology area (Refs. 3, 7, 9) have obtained compliance relationships in surface integral form by integrating the Greens function for a point load on the surface of a half space (the Bousinesq solution) over the areas of individual cells demarcated on the contact zone. This method works well as long as the extent of the contacting bodies is much larger than the dimensions of the contact zone, and the contact zone is far enough from the other surface boundaries so that the two contacting bodies may be treated as elastic half spaces. These conditions are, however, not satisfied by gears.

The approach that is described here is based on the assumption that beyond a certain distance from the contact zone, the finite element model predicts deformations well. The elastic half space model is accurate in predicting *relative* displacements of points near the contact zone. Under these assumptions, it is possible to make predictions of surface displacements that make use of the advantages of both the finite element method as well as the surface integral approach.

This method is related to asymptotic matching methods that are commonly used to solve singular perturbation problems. Schwartz and


**Dr. S. M. Vijayakar**
*is the principal at Advanced Numerical Systems, an engineering consulting firm specializing in special purpose finite element and other modeling techniques for the gearing industry.*

**Dr. D. R. Houser**
*is Professor of Mechanical Engineering at Ohio State University and the Director of the Gear Dynamics and Gear Noise Research Laboratory at the university.*

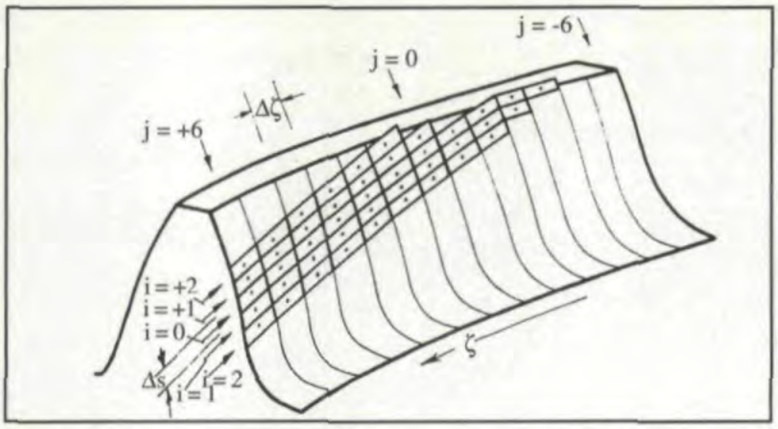


**Fig. 6 — Computational grid in the contact zone of the gears.**

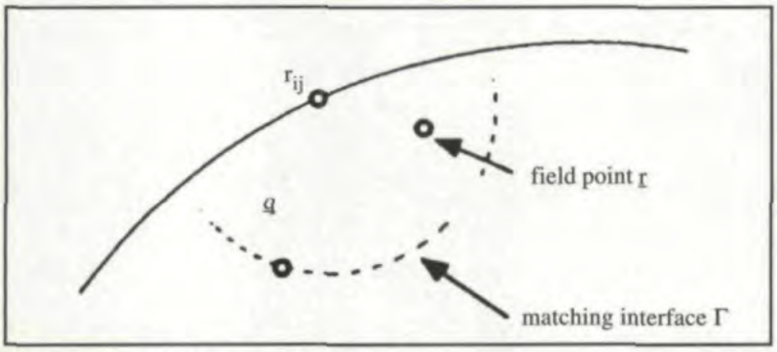


**Fig. 7 — The matching interface.**

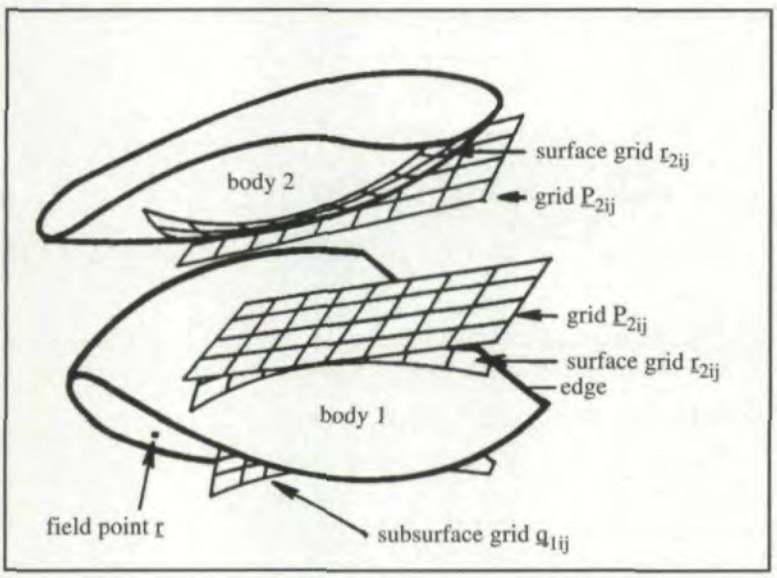


**Fig. 8 — The contacting bodies and the computational grids.**

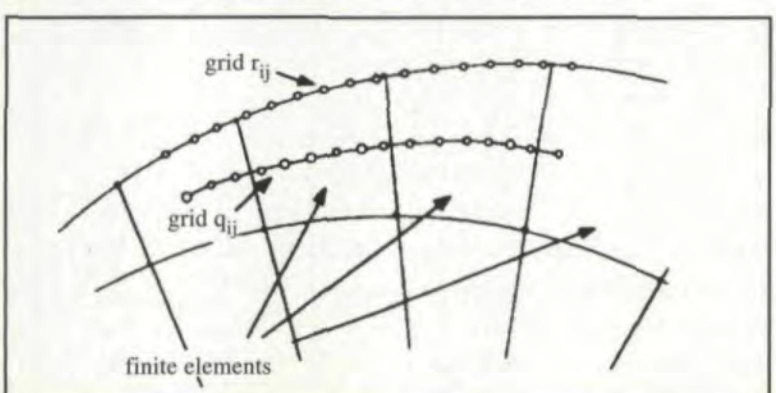


**Fig. 9 — The subsurface grid $q_{ij}$.**

Harper (Ref. 8) have used such an asymptotic matching method to determine the relative approach of two rigid cylinders pressed against an elastic cylinder in plane strain.

In order to combine the surface integral solution with the finite element solution, a reference or "matching" interface embedded in the contacting body is used. This matching surface is far enough removed from the principal point of contact so that the finite element prediction of displacements along this surface is accurate enough. At the same time, it is close enough to the principal point of contact so that the effect of the finite extent of the body does not significantly affect the relative displacement of points on this surface with respect to points in the region of contact.

Contact analysis is carried out in several steps. The first step is to lay out a grid at each contact zone. Then cross-compliance terms between the various grid points are calculated using a combination of a surface integral form of the Bousinesq and Cerruti solutions and the finite element model of the contacting bodies. Finally, load distributions and rigid body movements are calculated using an algorithm based on the Simplex method (Ref. 11).

In order to discretize the contact pressure distribution that is applied on the surfaces of two contacting gear teeth, a computational grid is set up. Fig. 6 shows such a computational grid that has been set up in the contact zone of the gears. The entire face width of one of the gears (gear 1), which is mapped onto $\{\zeta:\zeta\in[-1,+1]\}$, is divided into $2N+1$ slices. $N$ is a user- selectable quantity. *The thickness of each slice in the* $\zeta$ *parameter* space is $\Delta\zeta = 2/(2N+1)$. For each slice, $j = -N$ *to* $+N$, a cross section of gear 1 is taken at the middle of the slice, and a point is located on this slice that approaches the surface of the mating gear (gear 2) the closest. This selection is carried out using the undeformed geometry. If the separation between the two gears at this closest point is larger than a user-selectable *separation tolerance*, then the entire gear slice is eliminated from further consideration. Otherwise, a set of grid cells identified by the grid cell location indices $(i, j)$, $i = -M$ *to* $M$, and the position vector $\mathbf{r}_{ij}$ is set up centered around this closest point of slice $j$. The number $M$ is user-selectable. The dimension of the grid cells in the profile direction $\Delta s$ is also user-selectable.

Let $\mathbf{u}(\mathbf{p};\mathbf{q})$ denote the displacement vector at the location $\mathbf{q}$ on a gear due to a unit normal compressive force applied at the location $\mathbf{p}$, which is on the surface of the gear. The superscripts (si) and (fe) on a term will mean that the term has been calculated using surface integral formulae and a finite element model, respectively. Subscripts 1 and 2 will denote gears number 1 and 2, respectively. When the subscript is omitted in an equation, the equation will be understood to apply to both the gears.

Let $u(\mathbf{p};\mathbf{q}) = -u(\mathbf{p};\mathbf{q})\bullet\mathbf{n}$ be the inward normal component of the displacement vector $u(\mathbf{p};\mathbf{q})$, where $\mathbf{n}$ is the outward unit normal vector at the point $\mathbf{p}$.

The displacement $u(\mathbf{r_{ij}};\mathbf{r})$ of a field point $\mathbf{r}$ due to a load at the surface grid point $\mathbf{r_{ij}}$ can be expressed as:

$$u(\mathbf{r}_{ij};\mathbf{r}) = (u(\mathbf{r}_{ij};\mathbf{r}) - u(\mathbf{r}_{ij};\mathbf{q})\ ) + u(\mathbf{r}_{ij};\mathbf{q})$$

where $\mathbf{q}$ is some location in the interior of the body sufficiently removed from the surface (Fig. 7). If the first two terms are evaluated using the surface integral formulae, and the third term is computed from the finite element model, then we obtain the displacement estimate:

$$u(\mathbf{r}_{ij};\mathbf{r})(\mathbf{q}) = (u^{(si)}(\mathbf{r}_{ij};\mathbf{r}) - u^{(si)}(\mathbf{r}_{ij};\mathbf{q}) + u^{(fe)}(\mathbf{r}_{ij};\mathbf{q})$$

The term in parentheses is the deflection of $\mathbf{r}$ with respect to the "reference point" $\mathbf{q}$. This relative component is better estimated by a local deformation field based on the Bousinesq and Cerruti half-space solutions than by the finite element model. The gross deformation of the body due to the fact that it is not a half space will not significantly affect this term. On the contrary, the remaining term $u^{(fe)}(\mathbf{r_{ij}};\mathbf{q})$ is not significantly affected by local stresses at the surface. This is because $\mathbf{q}$ is chosen to be far enough beneath the surface. This term is therefore best computed using a finite element model of the body. The value $u(\mathbf{r_{ij}};\mathbf{r})(\mathbf{q})$ thus computed will, in general, depend on the location $\mathbf{q}$ because of the different values of the surface integral and finite element displacement fields there. The location is a so-called reference or "matching" point. We would like to match the surface integral and finite element solutions not only at one point, but also at a set of points belonging to a "matching interface" (Fig. 7). We will then be interested in that value of $u(\mathbf{r_{ij}};\mathbf{r})$, which will minimize the least squares deviation:

$$\int_{\Gamma} [u(\mathbf{r}_{ij};\mathbf{r}) - u^{(si)}(\mathbf{r}_{ij};\mathbf{r}) - u^{(si)}(\mathbf{r}_{ij};\mathbf{q}) + u^{(fe)}(\mathbf{r}_{ij};\mathbf{q}))]^2 d\mathbf{r}$$

where $\mathbf{q}$ varies over the reference surface $\Gamma$.

Another possibility, which lends itself better to spatial discretization is to choose a value for $u(\mathbf{r_{ij}};\mathbf{r})$ which minimizes:

$$\sum_{q\in\Gamma}[u(\mathbf{r}_{ij};\mathbf{r}) - (u^{(si)}(\mathbf{r}_{ij};\mathbf{r}) - u^{(si)}(\mathbf{r}_{ij};\mathbf{q}) + u^{(fe)}(\mathbf{r}_{ij};\mathbf{q}))]^2$$

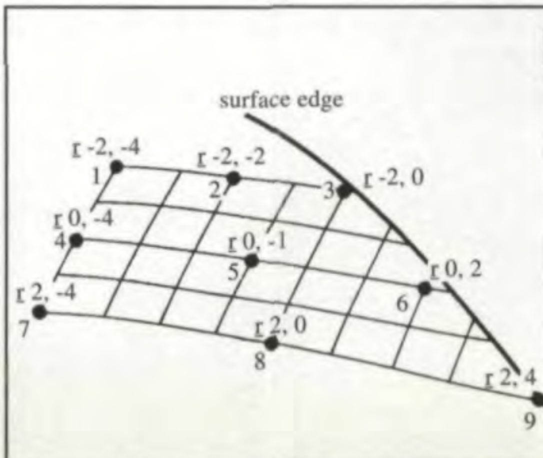
where $\mathbf{q}$ varies over a grid of points $\mathbf{q_{ij}}$ laid out over the matching interface $\Gamma$ (See Figs. 8-9). For convenience, points in this grid $\mathbf{q_{ij}}$ were chosen to lie half a finite element thickness below corresponding points in the surface grid $\mathbf{r_{ij}}$. . . . Let N be the total number of points in the grid $\mathbf{q_{ij}}$ . . . . Then the value that minimizes the least square deviation above is:

$$u(\mathbf{r}_{ij};\mathbf{r}) = (u^{(si)}(\mathbf{r}_{ij};\mathbf{r}) + \frac{1}{N}$$

$$\sum_{q\alpha\beta}[(\mathbf{r}_{ij};\mathbf{q}_{\alpha\beta}) - u^{(si)}(\mathbf{r}_{ij};\mathbf{q}_{\alpha\beta}))]^2$$

In order to obtain sufficient resolution of the contact stresses, the number of points in the grid $\mathbf{r_{ij}}$ will have to be very large, typically in the hundreds. Computation of all the terms of the type $u^{(fe)}(\mathbf{r_{ij}};\mathbf{q}_{\alpha\beta})$ would involve hundreds of back-substitutions through the decomposed finite element stiffness matrix. This would be prohibitively time-consuming because of the complexity of the three-dimensional finite element model of the body. Furthermore, the finite element model does not usually have an adequate degree of freedom at the surface to allow all the terms $u^{(fe)}(\mathbf{r_{ij}};\mathbf{q}_{\alpha\beta})$ to be independent of each other. Thus evaluating each such term by a separate back-substitution is probably also superfluous.

A better method is to obtain $u^{(fe)}(\mathbf{r}_{i(k)j(k)};\mathbf{q}_{i(l)j(l)})$ for a much smaller subset $\{\mathbf{r}_{i(k)j(k)};\ k = 1,2,\ldots,M\}$ of the grid $\{\mathbf{r_{ij}}\}$, as shown in Fig. 10, and the corresponding subset $\{\mathbf{q}_{i(k)j(k)};\ k = 1,2,\ldots,M\}$ of the grid $\{\mathbf{q_{ij}}\}$. If the




| k | (i(k), j(k)) |
|---|---|
| 1 | (-2, -4) |
| 2 | (-2, -2) |
| 3 | (-2, -0) |
| 4 | (0, -4) |
| 5 | (0, -1) |
| 6 | (0, 2) |
| 7 | (2, -4) |
| 8 | (2, 0) |
| 9 | (2, 4) |

**Fig. 10 — The grid subset $r_{i(k)j(k)}$.**

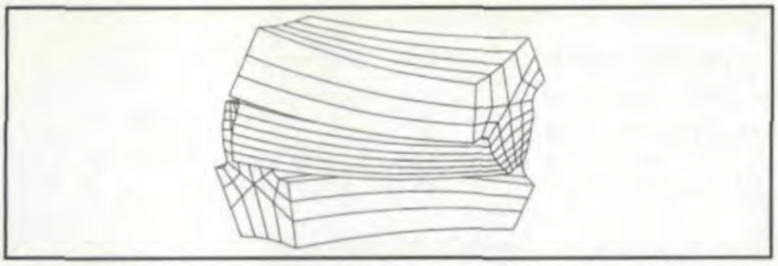


**Fig. 11 — The finite element model of a pair of contacting teeth from a 90° crossed helical gear set.**

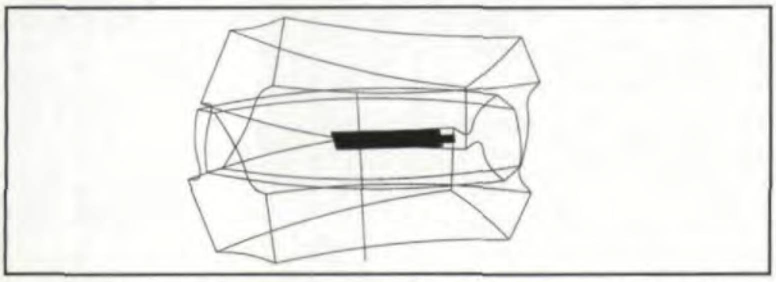


**Fig. 12 — The computational grid between a pair of contacting teeth from a 90° crossed helical gear set.**

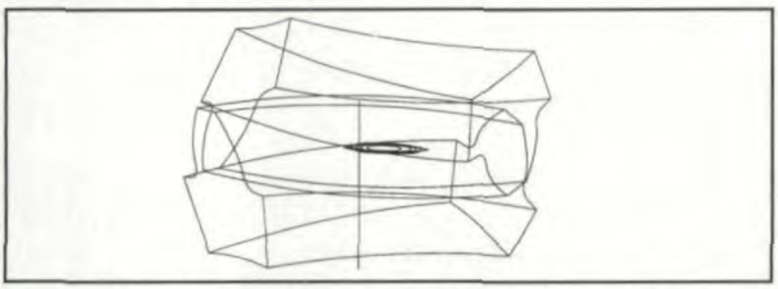


**Fig. 13 — The contact pressure distribution between the contacting pair of crossed helical gear teeth.**

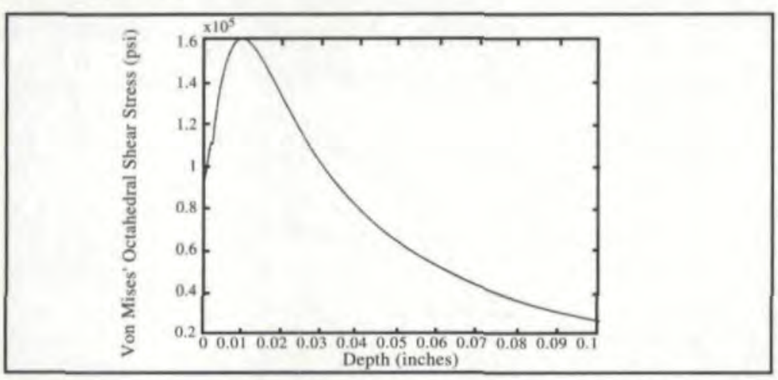



**Fig. 14 — Variation of subsurface shear stress with depth under the point of maximum contact pressure.**

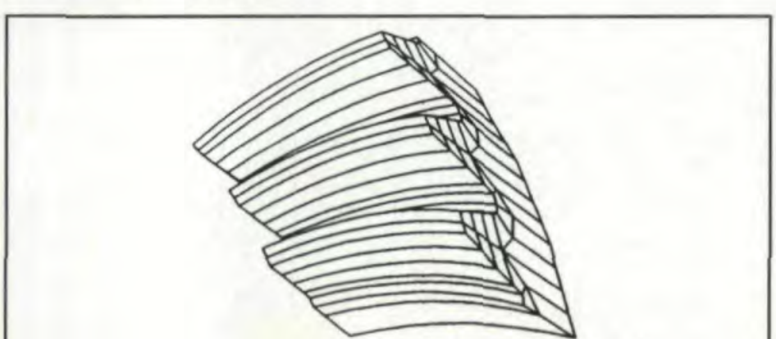


**Fig. 15 — A three-tooth finite element model of the gear showing the active surfaces.**

number of points M in this restricted set of grid points is small, then all the terms $u^{(fe)}(\mathbf{r}_{i(k)j(k)};\mathbf{q}_{i(l)j(l)})$ can be computed using only a small number M of back-substitutions.

In the numerical examples to follow, M was either 9 or 3. The values of $u^{(fe)}(\mathbf{r}_{ij};\mathbf{q}_{\alpha\beta}\}$ for the complete set of grid points can be obtained by using two-dimensional interpolants set up on the surface grid $\mathbf{r}_{ij}$ and the subsurface grid $\mathbf{q}_{ij}$, by the interpolation method:

$$u^{(fe)}(\mathbf{r}_{ij};\mathbf{q}_{\alpha\beta})$$

$$= \sum_{k,l=1,M} u^{(fe)}(\mathbf{r}_{i(k)j(k)};\mathbf{q}_{i(l)j(l)}) N_l(\alpha,\beta)\, N_k(i,j)$$

Where the functions $N_k(i,j)$ are biquadratic functions of i and j:

$$N_k(i,j) = \sum_{\alpha,\beta=0,1,2} a_{k\alpha\beta} i^{\alpha} j^{\beta}$$

The coefficients $a_{k\alpha\beta}$ are chosen such that

$$N_k(i(l)\ j(l)) = \delta_{kl}$$

where $\delta_{kl}$ is the Kronecker delta.

The finite element formulation that was used to evaluate the terms $u^{(fe)}(\mathbf{r}_{i(k)j(k)};\mathbf{q}_{i(l)j(l)})$ has been discussed in considerable detail in earlier papers (Refs. 10, 12).

The method described above is used to calculate all the terms $u_1(\mathbf{r}_{1ij};\mathbf{r}_{1kl})$ and $u_2(\mathbf{r}_{2ij};\mathbf{r}_{2kl})$ to build a compliance matrix. The contact force distribution over the grid and rigid body motions are determined by setting up the contact equations using this compliance matrix, and solving these contact equations by any of the numerous methods available in the literature. In the numerical examples described below, a method based on the Simplex algorithm of linear programming was used. Readers are referred to Reference 11 for more details.

## Numerical Examples

The following examples have been chosen to illustrate a few of the features of CAPP.

*Crossed Axis Helical Gear Set.* The first example shown in this article is that of a pair of identical helical gears whose axes are at right angles and whose operating helix angle is 45°. This makes an interesting example because the location and orientation of the contact zone can be easily predicted by simple calculations and by using the symmetry of the situation. Figs. 11-12 show a pair of contacting teeth of the 90° crossed helical gear set. Fig. 11 shows the finite element meshes of the two teeth, and Fig. 12 shows a contact grid that has been set up in the contact zone. The diametral pitch of this gear set is 10. Fig. 13 shows the contact pressure distribution between the teeth as calculated by CAPP. Fig. 14 shows the variation of subsurface Von Mises' shear stress as a function of depth below the point of maximum contact pressure.

*Hypoid Gear Set.* The next case chosen here is that of contacting hypoid gears. The cutting machines used to manufacture these gears have many kinematic settings. The settings are chosen such that the contact zone remains in the center of the tooth surfaces as the gears roll against each other. A heuristic procedure is available to select the settings, but in practice these setting have to be selected after a tedious iterative process involving cutting and testing actual gears. Even so, it is very difficult to predict the actual contact stresses, fatigue life, kinematic errors, and other design criteria, especially when not installed in ideal conditions. The contact stresses are so sensitive to the actual surface profile that conventional 3-D contact analysis is not feasible.

A sample 90° hypoid gear set from the rear axle of a commercial vehicle was selected. The gear ratio of this set was 41:11, and the axial offset was 1.5 inches. The gear surfaces had been experimentally shown to be ideal for this particular gear ratio and axial offset. In other words, the contact zone was found to remain in the central portion of the gear teeth in the operational torque range. The object of this numerical study is to verify this by looking at the manner in which the contact pattern shifts when the gears are moved around from their ideal locations.

The model was constructed by first generating values of coordinate normal vectors for points on the surface by simulating the gear cutting machines. The finite element description of the surface was then created by fitting tenth-order truncated Chebyshev series approximations to this data. The interior portions of the finite element were created semi-automatically. Only a sector containing three teeth of each gear was modeled, with each tooth being identical. The gear (gear 1) and the pinion (gear 2, the smaller gear) were then oriented in space as per the assembly drawings, and the analysis was carried out for each individual time step. Fig. 2 shows the six-tooth gear and pinion model. Sectoral

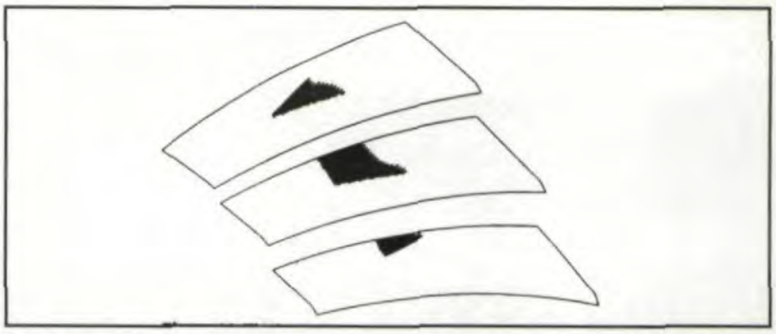


**Fig. 16 — The locus of the contact zone at a gear torque of 240 in-lbs.**

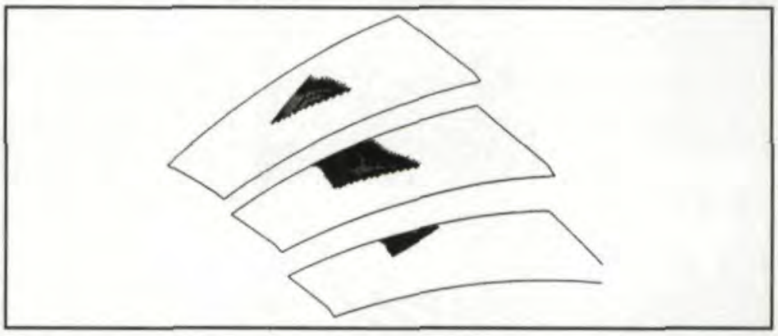


**Fig. 17 — The locus of the contact zone at a gear torque of 480 in-lbs.**

**Fig. 18 — The locus of the contact zone at a gear torque of 960 in-lbs.**

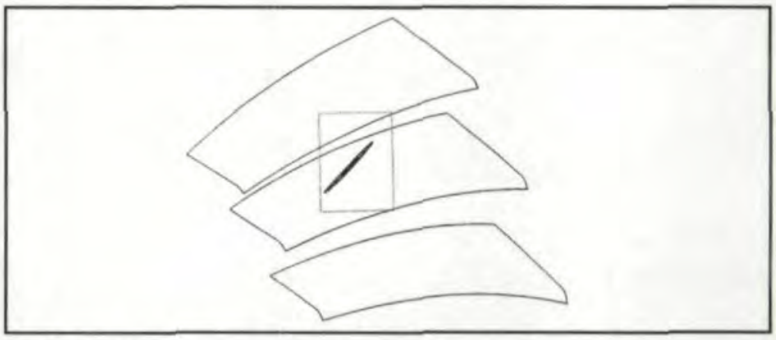


**Fig. 19 — Contact pressure contours for Position 1.**

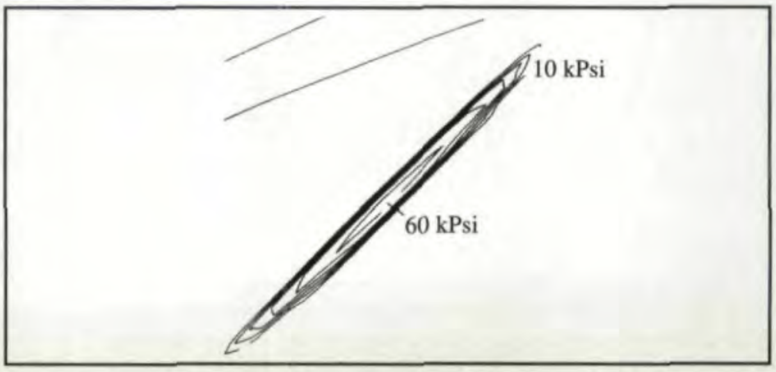



**Fig. 20 — Contact pressure contours for Position 1 magnified.**

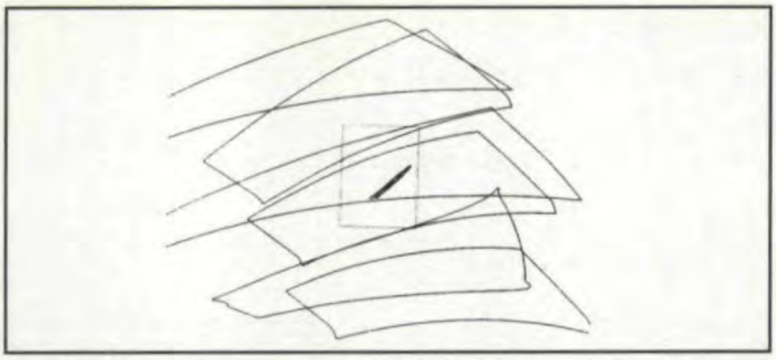


**Fig. 21 — Contact pressure contours for Position 2.**

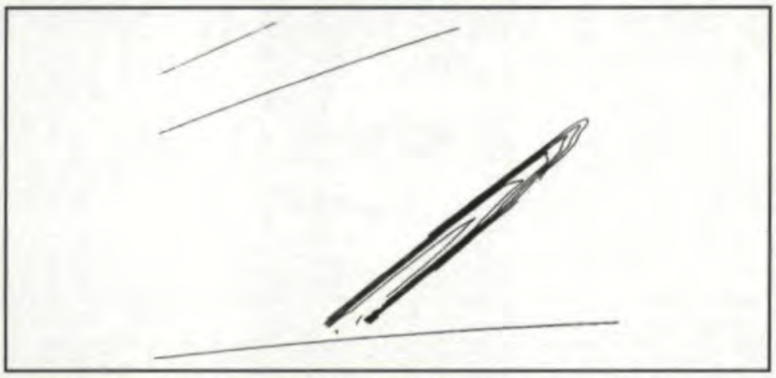


**Fig. 22 — Contact pressure contours for Position 2 magnified.**

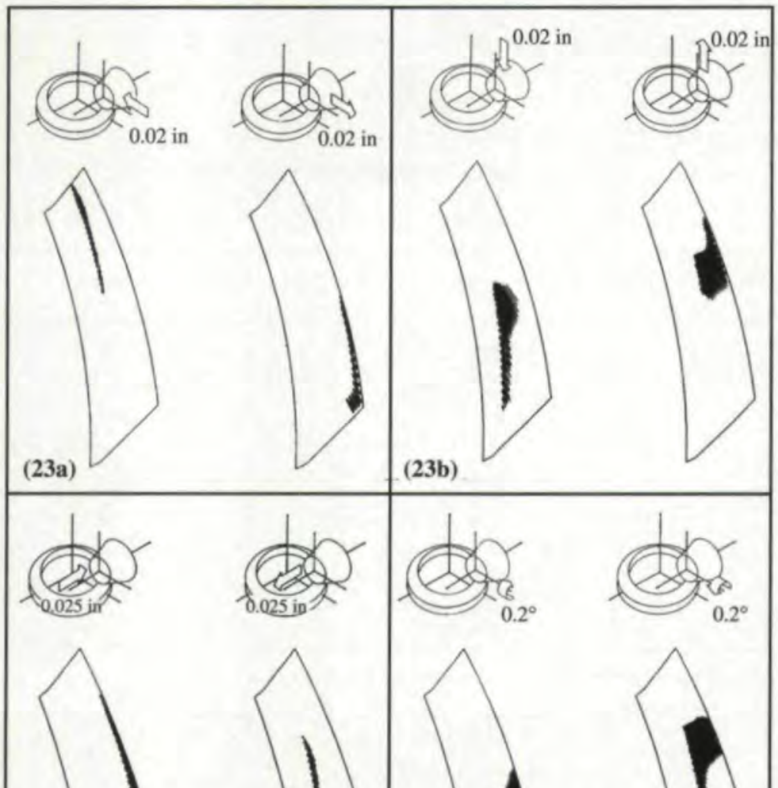



**Fig. 23a — The effect of an X translation on the contact pattern.**
**Fig. 23b — The effect of a Y translation on the contact pattern.**
**Fig. 23c — The effect of a Z translation on the contact pattern.**
**Fig. 23d — The effect of an X rotation on the contact pattern.**

symmetry is used to generate stiffness matrices from the stiffness matrix of one tooth. For this particular gear set, a three-tooth model suffices because at the most two teeth contact at a time. Fig. 15 shows the surfaces of the three-tooth gear. Figs. 16-18 show the contact pattern (which is the locus of the contact zone as the gears roll against each other), for a gear torque of 240, 480, and 960 in-lbs, respectively. Figs. 19 and 21 show views of the contact zone with contact pressure contours on the gear for two particular angular positions. Figs. 20 and 22 show magnified views of the contact zone for these two positions. They show contours of normal contact pressures on the surfaces. Computational grids of 11 x 25 cells were used on these surfaces to obtain the pressure distributions. Finally, the position of the pinion was perturbed slightly from the design location, and Figs. 23a-d show the contact patterns that were obtained. When compared to the contact pattern for the unperturbed position in Fig. 16, it shows that the best contact pattern does indeed occur at the designed position, lending credence to the notion that an analysis of the kind described in this article has the potential to be used in the design process itself.

*Examples of Other Post-Processing Features.* A variety of post-processing options are available for the display of the state of stress in contacting gears. Fig. 24 shows contour curves of maximum principal normal stress calculated at various sections in a pair of contacting helical gear teeth. Fig. 25 shows contour curves of maximum principal normal stress drawn along the surface of the gear tooth, and Fig. 26 shows a contour surface of maximum principal normal stress within a gear tooth. It is also possible to draw contour curves and surfaces for the minimum principal normal stress, and the Von Mises' octahedral shear stress. Fig. 27 is an example of an arrow diagram that can be used to show both the magnitude as well as direction of the principal normal stresses. Stresses are depicted by arrows pointing in the principal directions. Tensile stresses are depicted by outward pointing arrows, and compressive stresses are depicted by inward pointing arrows. The length of an arrow is proportional to the magnitude of the principal stress.

## Conclusions

Using a combination of finite element and surface integral methods seems to be, in the authors' opinion, the most practical method of modelling stiffness behavior of contacting bodies. When this method is used along with an efficient algorithm for solving contact equations, one can predict contact stress distributions and deformations in more realistic detail than otherwise possible. Results obtained from a contact analysis program (CAPP) based on this methodology have been found to compare well with calculations based on other methods. ■

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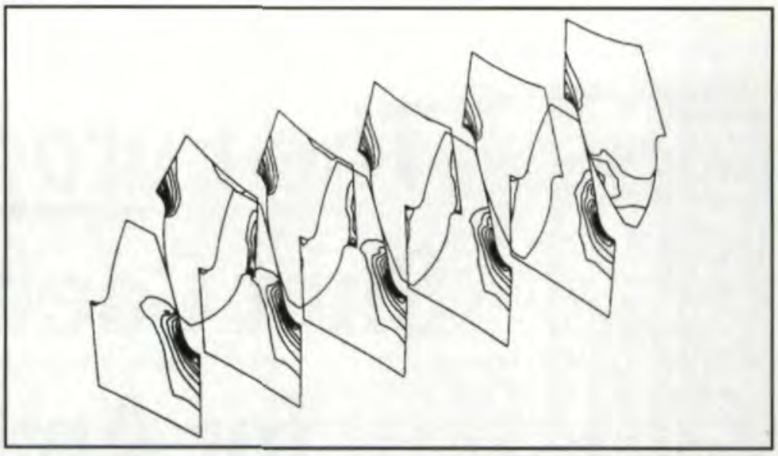

**Fig. 24 — Contour curves of maximum principal normal stress drawn within sections of a contacting pair of helical gear teeth.**

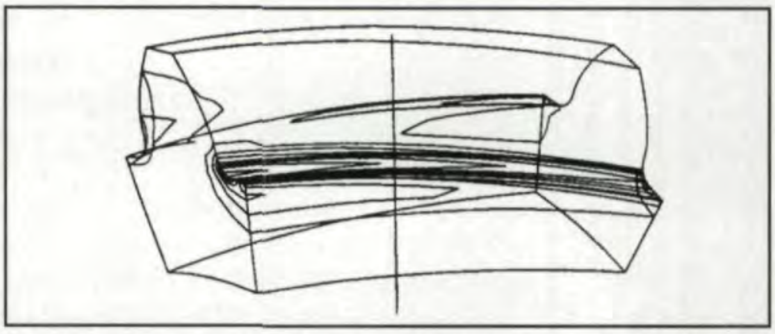

**Fig. 25 — Contour curves of maximum principal stress drawn along the surface of a tooth of a crossed helical gear set.**

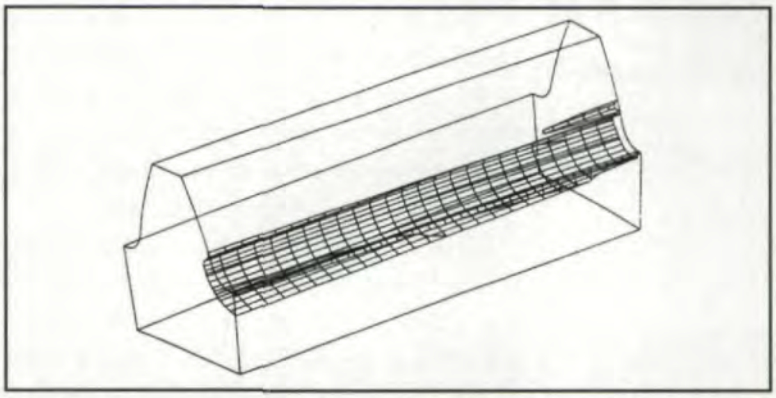

**Fig. 26 — A contour surface of maximum principal normal stress within a tooth of a helical gear set.**

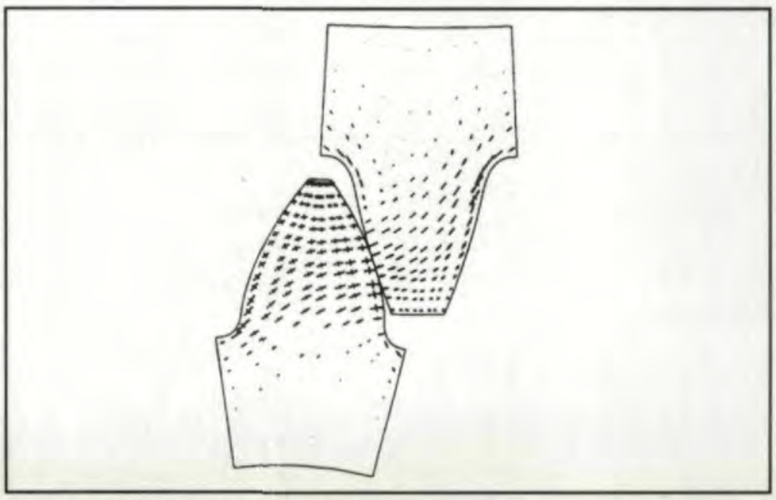

**Fig. 27 — A stress arrow diagram.**

# The European Rack Shift Coefficient "X" for Americans

**Don McVittie**
**Gear Engineers, Inc., Seattle, WA**

### Introduction

The use of dimensionless factors to describe gear tooth geometry seems to have a strong appeal to gear engineers. The stress factors I and J, for instance, are well established in AGMA literature. The use of the rack shift coefficient "x" to describe nonstandard gear proportions is common in Europe, but is not as commonly used in the United States. When it is encountered in the European literature or in the operating manuals for imported machine tools, it can be a source of confusion to the American engineer.

What follows is intended to provide a source for the background and derivations of the "x" factor as used in European standards and papers. The addendum modification, rack shift, or profile shift factor has several mathematical definitions in the U.S. Most European documents use a specific definition, based on a theoretical "zero backlash" gear pair in tight mesh at the nominal center distance. (See McVittie, 86 FTM 1 for discussion.)

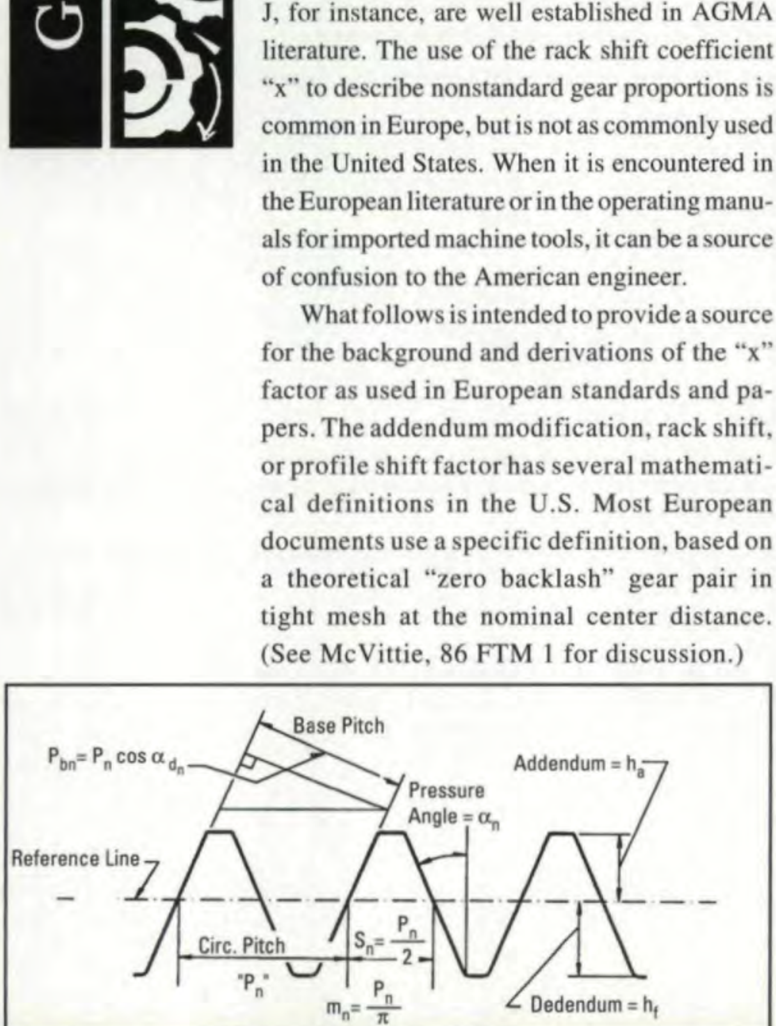


**Fig. 1 — Basic rack definition — normal plane.**

### Basic Rack

The definitions and equations in this article are based on a "basic rack" in which addendum and dedendum are measured from a reference line located where the tooth thickness and the space width on the reference line are equal. The basic rack represents the tooth form in the normal plane of a gear with an infinite number of teeth. The normal module of the basic rack is equal to the normal circular pitch divided by $\pi$. The normal diametral pitch of the basic rack is equal to $\pi$ divided by the normal circular pitch.

The basic rack represents the theoretical gear tooth form, not the form of the cutting tool. No allowance is made for backlash, finishing stock, or manufacturing method.

The standard 20° normal pressure angle basic rack of ISO 53 is commonly used. This document is valid for that basic rack and for any other basic rack which meets the criteria of Fig. 1.

### Addendum Modification Factor

The addendum modification factor "x" (*Profilverschiebungsfaktor,* "profile shift factor" in German) represents the distance, in tight mesh, from the reference line of the basic rack to the reference circle of the gear (rack shift or profile shift) for normal module = 1.0 or normal diametral pitch = 1.0.

### Sum of X Factors

The European practice is to define the sum of $x_1 + x_2$, ($\Sigma x$) for a theoretical gear pair which operates in tight mesh (has no backlash) on the nominal operating center distance.

The basic equation can be derived from the basic tooth thickness involute geometry equations and the requirement that the sum of the transverse tooth thicknesses at the operating pitch diameter is equal to the transverse circular pitch at that diameter.

$$\Sigma x = \frac{z_1 + z_2}{2} \cdot \frac{\text{inv}\,\alpha_{wt} + \text{inv}\,\alpha_t}{\tan\alpha_n}$$

(1) DIN 3992 Eq 9

$$\tan\alpha_t = \frac{\tan\alpha_n}{\cos\beta}$$

(2) DIN 3992 Eq 6

$$a = m_t \cdot \frac{z_1 + z_2}{2} \cdot \frac{\cos\alpha_t}{\cos\alpha_{wt}}$$

(3) DIN 3992 Eq 5

$$m_t = \frac{m_n}{\cos\beta} \qquad (4)$$

**X Factor for Each Gear**

The values of x for gear and pinion are chosen somewhat arbitrarily (See Maag, DIN 3992, and ISO/TR 4467 for further information on choice of x factors) according to the operating conditions and gear ratio, so that their total is equal to Σx. The theoretical addendum (tip) diameters and tooth thicknesses of the two gears in the gear pair are defined by their x factors.

$$d_a = d + 2 \cdot m_n\,(1+x) \qquad (5)$$

$$s_n = \left(\frac{\pi}{2} + 2 \cdot x \cdot \tan\alpha_n\right) \cdot m_n$$

(6) ISO DTR 10064/2 Eq 6.4

The actual addendum diameters and tooth thicknesses are then adjusted (usually reduced) to control backlash and tip to root clearance.

**Backlash Allowance**

A common convention among gear manufacturers is to reduce the normal tooth thickness of each member by the same amount, which may be a value in μm or a function of module, such as $.024 \cdot m_n$. This maintains the same cutting depth for both members and maximizes contact ratio. The direction (normal, transverse, reference circle, or base tangent plane) in which the tooth thickness reduction is to be measured must be specified, since there is no recognized convention.

| Table of Symbols | | |
|---|---|---|
| ISO | AGMA | Definition |
| a | C | Center Distance |
| c | | Clearance ($h_a - h_f$) |
| d | d | Diameter |
| E | | Tooth Thinning for Backlash |
| $h_a$ | a | Addendum |
| $h_f$ | | Dedendum |
| k | | Tip Shortening Factor |
| j | B | Backlash |
| m | m | Module |
| p | p | Circular Pitch |
| s | t | Tooth Thickness |
| x | | Addendum Modification Factor |
| α | φ | Pressure Angle |
| β | ψ | Helix Angle |

| Table of Subscripts | |
|---|---|
| Subscript | Meaning |
| (none) | At Reference Diameter |
| a | At Addendum (Tip) Diameter |
| b | At Base Cylinder Diameter |
| f | At Root Diameter |
| n | Normal Plane |
| o | Tool Dimensions |
| t | Transverse Plane |
| w | At Working Diameter |
| y | At Any (Undefined) Diameter |
| 1 | Pinion |
| 2 | Gear or Rack |

Working Group (WG)2 of ISO/TC60 is considering a draft technical report, DTR10064/2, containing tables which recommend that the tooth thinning for backlash, called "upper allowance of size", $E_{ssn}$, be a function of the pitch diameter of each part. The values are measured normal to the helix angle in the reference cylinder. The values can be converted as follows:

The transverse circular allowance, $E_{sst}$, is:

$$E_{sst} = \frac{E_{ssn}}{\cos\beta} \qquad (7)$$

The normal allowance in the base tangent plane, $E_{bsn}$, (normal to the tooth surface) is

$$E_{bsn} = E_{sst} \cdot \cos\alpha_t \cdot \cos\beta_b \qquad (8)$$

which can also be expressed as

$$E_{bsn} = E_{ssn} \cdot \cos\alpha_n \qquad (9)$$

The resulting transverse circular backlash at

**Don McVittie**

*is President of Gear Engineers, Inc., Seattle, WA. He is a past president of AGMA and Chairman of the U. S. Technical Advisory Group for International Gear Standards. He is a licensed professional engineer in the State of Washington.*

the working diameter is a function of allowance, center distance, and tooth accuracy. (See AGMA 2002 for more information.)

### Tip Shortening for Clearance

When $\Sigma x > 0$, the tips of external gear pairs should be shortened to maintain standard tip to root clearance. The reduction in clearance is often ignored for small values of $\Sigma x$, but for larger values the addendum should be shortened by $k \cdot m_n$.

$$k =$$

$$\frac{z_1 + z_2}{2} \cdot \left[ \frac{\text{inv}\,\alpha_{wt} - \text{inv}\,\alpha_t}{\tan \alpha_n} - \frac{1}{\cos \beta} \cdot \left( \frac{\cos \alpha_t}{\cos \alpha_{wt}} - 1 \right) \right]$$

(10) Maag Eq 68

Tip diameters of internal gear pairs should be checked for clearance and interference with cutters and mates by calculation of actual cutting and mating conditions.

### Actual Root Diameter and Clearance

The European method doesn't calculate the actual root diameter of gears which are thinned for backlash by feeding the cutter to greater depth. When the actual root diameters are calculated, the addendum diameter required for standard clearance can be calculated more accurately from Eq. 11.

$$d_{a1} = 2 \cdot (a - c) - d_{f2} \qquad (11)$$

The root diameter at maximum tooth thickness can be calculated as follows:

$$d_f = d - 2 \cdot \left( h_{ao} - x \cdot m_n + \frac{E_{ssn}}{2 \cdot \tan \alpha_t \cdot \cos \beta} \right) \quad (12)$$

Equation 12 is based on the assumption that the cutter addendum, $h_{ao}$, is measured as shown in Fig. 1 for the basic rack. If the gear is to be finished in a second operation, as by shaving, skiving, or grinding, a more detailed study is required to estimate the finished root diameter. (See Appendix E, Sec. E6 of AGMA 218.01 for more information.)

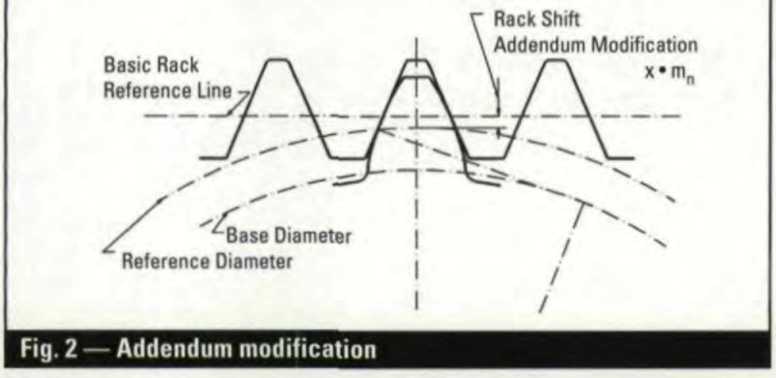


Fig. 2 — Addendum modification

### Convention for Signs

For external gears, the value of x is positive when the tooth thickness is increased and the value of $\Sigma x$ is positive when the center distance is greater than standard.

The same convention can be used for internal gears if the sign of the center distance is considered negative. "Long addendum" internal gears have a negative x. This convention is common, but is not universal.

### Internal Gears

The equations in this article are arranged for external gears. With a few exceptions, they can be used for internal gears if the internal diameters, center distance, and number of teeth are made negative. The convention for signs must be checked carefully. One trap is division by a negative value to calculate an involute function, which must be positive. It is good programming practice to take the absolute value of the quotient before calculating the angle from the involute function. ■

### Appendix — Derivations of Equation 9

$$E_{bsn} = E_{sst} \cdot \cos \alpha_t \cdot \cos \beta_b$$

$$= \frac{E_{ssn}}{\cos \beta} \cdot \cos \alpha_t \cdot \cos \beta_b \qquad (8)$$

$$\cos \beta = \frac{p_n}{p_t} \quad \cos \beta_b = \frac{p_{bn}}{p_{bt}}$$

$$\cos \alpha_n = \frac{p_{bn}}{p_n} \quad \cos \alpha_t = \frac{p_{bt}}{p_t}$$

$$E_{bsn} = E_{ssn} \quad \cos \alpha_n \qquad (9)$$

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# Money Business

## Exporting – Part III

***Letters of credit, documentary collection, forward options, and other "must know" banking terms for exporters.***

**Nancy Bartels**

The object of any business transaction, be it foreign or domestic, is making a profit. That's why you go through all the effort of making and selling your product in the first place. Getting paid in a timely and convenient manner is crucial to making a profit, but when your customer is in another country, this "timely and convenient" payment can become complicated; hence, your need for a banker with expertise in international markets.

To help explore the basics of the banking part of your export equation, *Gear Technology* spoke with Terese S. Gravenhorst, second vice president, International Banking Division, American National Bank and Trust Company of Chicago.

Like other experts we have consulted for this series, Gravenhorst emphasizes that you should contact your banker early and ask lots of questions. "Call your banker at the same time you're contacting your freight forwarder, your lawyer, and your accountant. Even if you're still in the 'just thinking about it' stage, that's a good time to get your bank involved."

The questions you should be asking relate to your bank's capability to assist you with foreign transactions. Can it help you with wire transfers, foreign exchange, various international payment methods, documentary collections, letters of credit, etc.? How deep is the bank's expertise in these areas?

If your bank is a large one in a major metropolitan area, the chances are that it can handle your export banking needs. But even if you are accustomed to dealing with a smaller bank without this background, you can probably get the services you need without changing bankers.

Many smaller banks are "domestic correspondents" of larger banks; that is, they count on the larger bank to provide services like international banking, which the smaller bank cannot. Gravenhorst explains: "[The smaller bank] doesn't have the international expertise, so it will piggyback off us. On the export side we will deal directly with the customers, but it's understood that we are assisting the smaller bank as a service to their customer, not competing with them in any way."

Involving your banker in your export plans early has other advantages as well. He or she will be essential if you need to increase your line of credit to finance some of your export ventures, but, in addition, your banker may be

## MANAGEMENT MATTERS

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**Terese S. Gravenhorst**
*is the second vice president in the International Banking Division of American National Bank and Trust Company of Chicago, IL.*

able to provide you with other ideas or connect you with people or organizations who can advance your export plans. "They can sometimes point you in other directions or ask, 'Have you thought of this or that?'" says Gravenhorst.

**The Documents in the Case**

The financial documents required in export transactions have an exaggerated reputation for being complicated and difficult to understand. They can be highly detailed, but their basic function in every case is simple — to see that you get your money once your product has been delivered to the buyer.

The documents required will depend on the method of payment upon which you and your customer have agreed. These terms can be an important negotiating point and will be a factor in determining how competitive your product will be and what your costs will be. They will affect your collection risk, how long you will have to wait for payment, and your ability to obtain financing.

Your decision about terms should be based on a number of factors, such as the value of the shipment, whether you have had to make special modifications to the product, the relative risk of shipping to the particular country, the creditworthiness of the buyer, and the terms your competitors are offering.

The riskiest method of payment from the seller's point of view is the open account. This is the same kind of arrangement you might enter into with a domestic customer: You deliver the goods, and the customer sends you a check or arranges for a wire transfer to your account within the agreed-upon amount of time.

This is a simple approach, but it has some obvious risks. Recourse is difficult should the customer default. Gravenhorst explains, "It's more complicated if the customer is halfway around the world. You have to go to his country, deal with his country's laws, etc."

The open account arrangement is probably wise only if you know your customer very well and are absolutely sure of his or her creditworthiness. And even then, if an overseas customer is willing and ready to pay, political realities may make it impossible. "Remember the situation in Kuwait," says Gravenhorst. "Those people had the money, but our government blocked payment. You have to consider the political stability of your customer's country."

Documentary collection offers some reduced risk to the seller. Under this method of payment, you ship the goods, and the shipping and commercial documents which enable the buyer to get the goods are transmitted through banking channels. They are released to the buyer only when he or she releases funds for payment.

Gravenhorst explains: "You ship your goods and then present the documents like the invoice, packing list, and ocean bill of lading to your buyer's bank, who notifies him and says, 'We've received the documents you need to get your goods released. We will release them to you only when you make payment.' The buyer instructs the bank to debit his account, and the foreign bank sends the funds to your bank. The problem with this arrangement is that the buyer can also say, 'I've changed my mind. I don't want this shipment.' In that case you either have to ship the goods back home or find another buyer for it there."

The least risky and, therefore, most popular method of payment with sellers is the letter of credit. This document works much like a certified check. The bank guarantees the payment of the money as long as the requirements specified in the letter of credit are met. "The issuing bank is saying, 'Whether or not our customer, who is your buyer, goes bankrupt or falls off the edge of the earth, we, the bank, promise payment to you.'" says Gravenhorst.

**Letters of Credit**

Letters of credit come in several forms, the most com-

## MANAGEMENT MATTERS

### The 10 Most Common Discrepancies in a Letter of Credit

1. The letter of credit has expired.
2. Documents are presented late.
3. Shipment is late.
4. Documents are inconsistent with one another.
5. Applicant's/beneficiary's name and/or address differs on the documents and the letter of credit.
6. Goods are shipped via air instead of ocean or vice versa.
7. Partial shipments are made when the letter of credit prohibits them.
8. Documents are not signed when required.
9. Draft not presented.
10. Insurance policy or bill of lading is not endorsed.

mon being irrevocable, confirmed, and advised. If you and your customer agree to use a letter of credit, be sure that you get an irrevocable one. This means that none of the conditions in the letter can be changed without your consent. Confirmed and advised letters of credit provide even more guarantees of payment under particular circumstances.

The letter of credit is great insurance, but it places obligations on you as well as on the buyer. A letter of credit will contain stipulations with which you must comply, or the issuing bank will not pay out on it. These stipulations may include method of shipment, dates and times of delivery, documents that will be required, etc.

It's a good idea to get copy of the terms and conditions that are to be included before an actual irrevocable letter of credit is issued. Your banker, lawyer, and freight forwarder should review these terms to make sure you can comply with them before the letter is issued, because once it is, additional fees will be incurred and time will be lost for every change you make.

"You as exporter have to review that letter," says Gravenhorst. "Sometimes customers will get the letter of credit and just stick it in a drawer and go ahead and make the shipment. Then comes time to present the documents, and they say, 'Well, we missed this expiration date,' or 'We needed to have these documents notarized by the Saudi Arabian consulate.' Now the bills of lading and the invoices are gone, and getting all that done may be impossible. Letters of credit are not that complicated, but you do have to review them line by line."

Most letters of credit are also "advised". This means that your banker in the U.S. verifies the authenticity of the letter of credit issued by your customer's bank. Gravenhorst explains: "If letters of credit are issued in telex form, code numbers are included by the issuing bank. Our telex department can decode these numbers to make sure that the letter of credit has really been issued by, say, Deutsche Bank in Germany, and not by the customer pretending to be Deutsche Bank. We also have books of signatures from thousands of banks around the world, and we can verify all the signatures. We don't see it often, but sometimes fraudulent letters of credit do come through."

Even with an advising bank to verify the authenticity of the letter, the customer would still look to the issuing bank, not the advising one, for payment.

**Additional Guarantees**

Under some circumstances, a simple irrevocable letter of credit is not enough to insure payment. If additional guarantees are required, you will want to look into confirmed letters of credit.

The confirmed letter of credit would be appropriate in a case where you do not feel comfortable with the issuing bank's ability to meet

# Methods of Payment for Overseas Transactions

## Open Account

Payment made as agreed upon between buyer and seller.

*Risks:*
- The buyer may default.
- Government regulations may cause delay in obtaining foreign exchange or funds transfer.
- Payment may be blocked because of political events.

*Appropriate When:*
- Seller trusts the buyer and knows payment will be made.
- No government regulations inhibiting foreign exchange or funds transfer exist.
- Seller has sufficient liquidity or access to outside financing to extend deferred payment.

## Documentary Collections

**Documents Against Payment** — Payment will be made when sight draft is presented to buyer's bank, and buyer agrees to pay draft. Goods are available to buyer after payment.

*Risks*
- Buyer may refuse shipment.
- Government regulations may cause delay in obtaining foreign exchange or funds transfer.
- Payment may be blocked because of political events.

*Appropriate When:*
- Seller knows buyer will accept shipment.
- No government regulations inhibiting foreign exchange or funds transfer exist.

**Documents Against Acceptance** — Shipping documents are presented to buyer's bank with instructions to release documents after buyer has "accepted" draft (a promise to pay at a future date). Payment will be made on maturity date of accepted draft.

*Risks*
- Buyer will default.
- Government regulations may cause delay in obtaining foreign exchange or funds transfer.
- Payment may be blocked because of political events.

*Appropriate When:*
- Seller confident buyer will pay accepted draft.
- No government regulations inhibiting foreign exchange or funds transfer exist.
- Seller has sufficient liquidity or access to outside financing to extend deferred payment.

## Commercial Letters of Credit

**Sight** — Payment is made when the documents are presented to issuing bank, provided that the documents are in compliance with the letter of credit terms. Goods are available to buyer after the draft has been paid.

**Time** — Payment is made on the maturity date of the accepted draft. Goods are available to the buyer after the draft has been accepted by the issuing bank (a promise to pay has been established).

*Risks*:
- Issuing bank defaults on its payment obligation.
- Payment blocked due to political events.
- Discrepancies in documents prevent payment.

*Appropriate When:*
- Seller isn't sure of buyer's creditworthiness.
- Seller is confident that buyer's country won't take any action to block payment.
- Seller is confident issuing bank will fulfill obligation.

## Confirmed Letter of Credit

Time of payment and goods are available on the same basis as with a confirmed letter of credit.

*Risks*:
- Confirming bank defaults on payment obligation.
- Discrepancies in documents prevent payment.

*Appropriate When*:
- Seller is unsure or unable to evaluate issuing bank's creditworthiness.
- Seller is unwilling to accept political risks in issuing bank's country.

## Cash in Advance

Payment is made before shipment. Goods are available after payment.

*Risks*:
- None.

*Appropriate When:*
- Seller has negotiating strength to demand cash in advance.
- Buyer's country doesn't prevent advance payment.

its obligations. In that case, you would ask your bank to add its confirmation (for which it will charge a fee), to that of the issuing bank. Then if the issuing bank cannot make payment for some reason, as long as all the documents are in compliance with the letter of credit, the U.S. bank will pay.

According to Gravenhorst, you probably would not need to ask for a confirmed letter of credit from a major bank in a major exporting country, such as Japan or Germany. But if you are doing business in a country where political and economic stability are questionable, or where the issuing bank itself is not as well-known, the confirmed letter of credit is an option you may wish to consider.

**The Price of Security**

What does this additional payment guarantee cost the seller? This will depend on a number of factors. And who (you or the buyer) pays what share will be one of the subjects of negotiation. But basic costs for an irrevocable letter of credit are about one-half percent of the invoice amount. Charges for amending and advising on a letter of credit run between \$40.00 and \$60.00. Fees for a confirmed letter of credit are on a sliding scale; the greater the risk in a particular case, the higher the fees.

**The Other Side of the Coin**

While an irrevocable letter of credit guarantees the seller payment, it is a riskier proposition for the buyer. If he or she agrees to such a payment method, what happens if the customer is not satisfied with the product when it's delivered? What recourse does he have?

From the issuing bank's point of view, none. Says Gravenhorst, "It's important to remember that banks deal only in documents. Sometimes customers will ask us not to pay out on a letter of credit, and we have to tell them that if the documents are in order, we have to pay. Customer satisfaction has nothing to do with it."

This is why buyers of big-ticket or custom-made iterms (like specialty gears) will commonly negotiate an inspection of the product prior to shipment. The buyer will insist on a signed inspection certificate as part of the terms of the letter of credit. Another approach to the same problem is to agree to partial payment upon delivery and the rest after a specified period of time during which the customer can inspect and use the product.

**Foreign Exchange**

Another complication of the overseas market is the fact that currencies in different countries have different values. If you have sold a product for \$25,000 U.S., how do you guarantee that by the time the goods are delivered and payment is made, you will have \$25,000 in payment, in spite of fluctuating currency markets?

The simplest way, of course, is to insist on payment in U.S. funds. But that may not be a realistic approach. If your competitors are dealing in pounds, yen, or deutschemarks, you may have to as well.

"This is a place where you need to get your bank's foreign exchange division involved," says Gravenhorst. "You have to be aware that, yes, there are currency risks, and if you're selling in currencies other than your own, you can't be dumb. You have to be aware that they do fluctuate, and if you sell today at one rate, and you actually receive your money at a later date, the amount could be quite different.

"There are instruments called forward options which allow you to lock into a rate of exchange, so no matter what happens to the value of the currency in six months, you get your $25,000. But you do need to work with people who are well versed in foreign exchange to explore these options.

"This is another place to ask questions of your bank. Sit down and talk to the people in your bank's foreign exchange department. They can give you their feelings about what these other currencies are going to do in the future. They won't be right 100% of the time, but they do provide a another resource for you, another piece of input."

**EXIMBANK & FCIA**

Letters of credit are not the only way to strengthen your financial hand when exporting. Two organizations which can provide you with export credit insurance are EXIMBANK and FCIA.

EXIMBANK is the Export/Import Bank of the United States, which is part of the Treasury Department. Its main function is to promote exports. Its programs are split up into two sections — insurance and guarantee.

On the insurance side, EXIMBANK will insure your export receivables. Gravenhorst explains: "Say most of your foreign competitors are offering open account terms, but you don't want to have to take that risk because that means you have foreign receivables on your books. Maybe your bank is doing receivable financing, and they're not going to take foreign receivables. So in order to mitigate that foreign risk and to keep your bank happy, you get this export credit insurance."

## MANAGEMENT MATTERS

**Banks deal only in documents. If the documents are in order, a bank must pay out on a letter of credit, regardless of any dispute between the buyer and seller.**

This credit insurance comes in two kinds — commercial and political. Under commercial coverage, if for any reason the buyer fails to pay, you can collect from the insurer. With political coverage, if the political situation in your buyer's country makes it impossible for him or her to get money out to you, the insurer pays.

FCIA is the Foreign Credit Insurance Association. At one time it was affiliated with EXIMBANK, but is now an independent organization. It provides the same kinds of insurance and guarantee programs as EXIMBANK, but because it is a private company, it can offer arrangements under conditions where it might not be politically expedient for a government organization to do so, say, in the area of defense-related exports.

This kind of insurance can cost anything from 10 cents to $1.50 per $100 of invoice, depending on the risks involved.

EXIMBANK also provides loan guarantees for companies involved in exporting. "For example," says Gravenhorst, "one of our customers wanted to use some of their work in process as collateral for a line of credit for the purpose of export transactions. Because the work in process was for a foreign job, we wouldn't allow that as collateral, but EXIMBANK was willing to. So with the EXIMBANK guarantee, we could offer a line of credit to our customer."

Exporters should note that while EXIMBANK is interested in encouraging exports, its loan guarantees are by no means "free money." The bank is always looking for reasonable assurance of repayment and requires at least two or three years of financial operating experience on the part of the borrower.

**Summing Up**

Getting your money from a buyer in a foreign country is never as simple as collecting from one down the street, but the complications should not be a deterrent to the seller sincerely interested in overseas markets. Exporting is as business strategy as old as commerce itself, and the rules for success are just as venerable.

As Gravenhorst says, "Be prepared. Do your homework. Ask questions. Any bank or freight forwarder would rather have you ask questions early or while the work is in process than to wait until you've shipped the goods. Then it's too late. Ask as many people as you can and get a good team behind you. That's the way to do well." ■

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Sales Manager

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NORDIC INTERNATIONAL INC.
Mr. Thad N. Schott
1340 Depot Street, Cleveland, Ohio 44116 USA
Tel: (216) 331-2231, Fax : (216) 331-2232

### REPRESENTATIVE WANTED:

**Largest manufacturer of gear hobs, shaper cutters, shaving cutters, and broaches in India is interested in selling in the U. S. market. Direct users or selling representatives can contact M/S.S.R.P. Tools, Ltd., 10, 6th Street, Gopalapuram, Madras, 600 086. Phone: 91-44-8277123. Fax 91-44-8276965.**

## HELP WANTED

DIRECTOR OF SALES & MARKETING: $85,000/Bonus. Custom Gear Design and Repair. National Rep and Direct Sales.
DESIGN ENGINEER: $55,000. Custom Gear Boxes, AutoCAD.
PLANT SUPERINTENDENT: $50's/Bonus. Supervise 100-150. Gear Cutting.
Contact: Ann Hunsucker, Excel Associates, P.O.Box 520, Cordova, TN 38018 or call (901) 757-9600 or FAX (901) 754-2896.

**To advertise in *Gear Technology* Call (708) 437-6604.**

**Rates:** Line classified - $37.50 per line. 8 lines per inch. $300 minimum. Classified Display - per inch (3" min.) **1X** - $170, **3X**-$160, **6X** - $150. Type will be set to advertiser's layout or *Gear Technology* will set type at no extra charge.
**Payment:** Full payment must accompany classified ads. Send check or Visa/Mastercard/American Express number and expiration date to: *Gear Technology*, P. O. Box 1426, Elk Grove Village, IL, 60009. **Agency Commission:** No agency commission on classifieds. **Materials Deadline:** Ads must be received by the 25th of the month, two months prior to publication. **Acceptance:** Publisher reserves the right to accept or reject classified advertisements at his discretion.

## HEAT TREATING



## COMPUTER AIDS


You need a Hofler.
You deserve a Hofler.
You were born to own a Hofler.
Now you can afford one.
Gear Expo
"Come see us at AGMA Gear Expo, Booth #331."
Introducing the new baby Hofler. If your life is gears but your wallet is thin, this is the measuring instrument for you.
The Zeiss Hofler ZP-250 will save you money because it costs far less than the competition. It'll save you time because it measures the average gear in less than four minutes. It'll save you space because its footprint is only 63" by 102," making it the ideal shop floor instrument.
Yet for all the savings, it's a Hofler, which means it's more accurate, more sophisticated, and just plain better than the competition.
Today is a good day to fulfill your destiny. Call 1-800-888-1967, ext. 42 for a free brochure on the Zeiss Hofler ZP-250.
ZEISS
More Than Accurate. Prezeiss.
CIRCLE A-8 on READER REPLY CARD

# THE INDUCTION EDGE

TOCCO PROFILE HARDENING (TPH) eliminates three of a gear user's worst problems:

- tooth breakage
- pitting
- spalling

Gears processed by the TPH process exhibit increased hardness and strength at the pitch line with an optimum strength gradient at the root fillet. These metallurgical properties are achieved without excessive tip temperature and without tooth form brittleness.

Don't confuse the TOCCO TPH process with conventional dual frequency induction heating. The TPH process merges three distinct induction heating techniques: sequentially-programmed, audio low frequency preheating, incremental induction hardening and high intensity radio frequency final hardening. The result is good austenitic/martensitic transformation and beneficial residual compressive stress at the root with elevated hardness and strength/depth at the pitch line.

The TPH process accommodates a broad range of metals, including medium carbon steel, cast iron and powder metals. You're not locked into expensive alloy steels. The process is economical for batches of one or 1,000, and is used today by top automobile and gear manufacturers.

Gear Profile Hardening is another example of the "edge" TOCCO gives you in induction heating. Our metallurgists will send you a fresh report on TPH technology. All you need do is ask.

**TOCCO®**

30100 Stephenson Hwy.
Madison Heights, MI 48071
TEL: (313) 399-8601
FAX: (313) 399-8603

**CIRCLE A-3 on READER REPLY CARD**