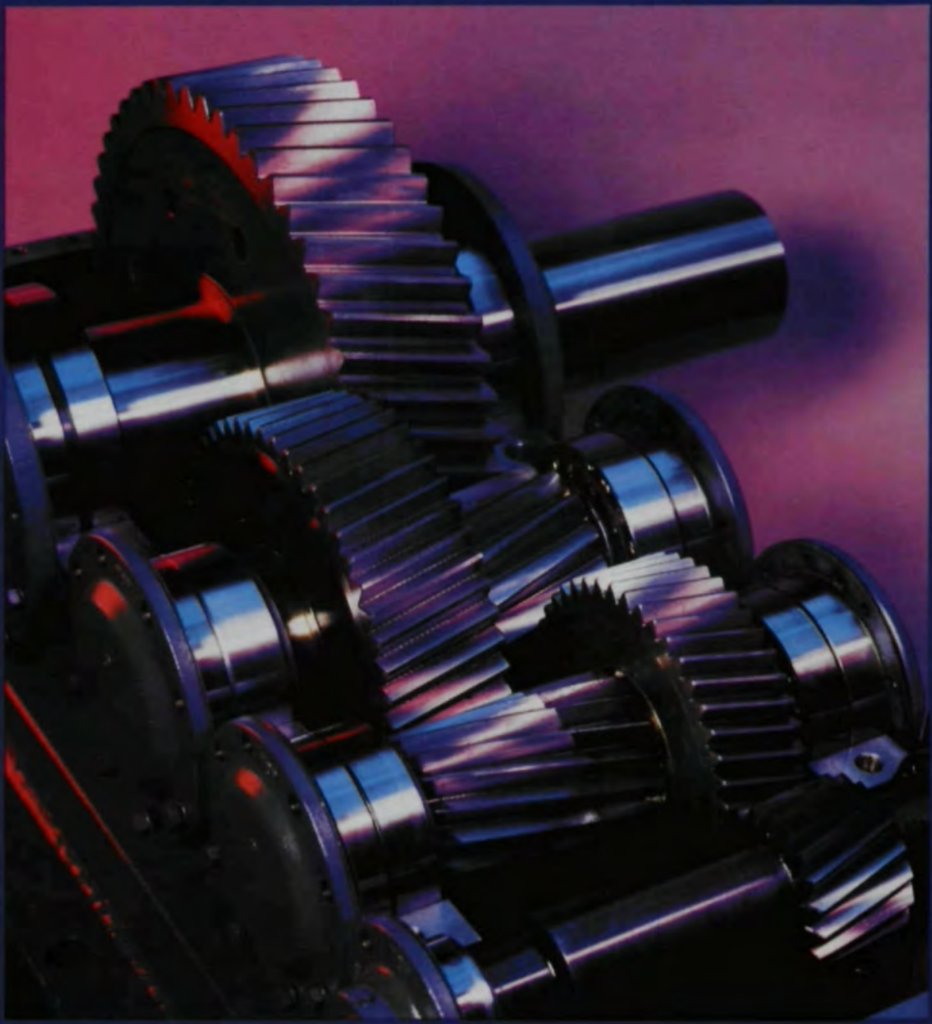
GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

NOVEMBER / DECEMBER 1992



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VOL. 9, NO. 6

GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published bimonthly by Randall Publishing, Inc., 1425 Lunt Avenue, P.O. Box 1426, Elk Grove Village, IL 60007. Subscription rates are: $40.00 in the U.S.; $50.00 in Canada; $55.00 in all other countries. Second-Class postage paid at Arlington Heights, IL, and at additional mailing office. Randall Publishing makes every effort to ensure that the processes described in GEAR TECHNOLOGY conform to sound engineering practice. Neither the authors nor the publisher can be held responsible for injuries sustained while following the procedures described.

Postmaster: Send address changes to GEAR TECHNOLOGY, The Journal of Gear Manufacturing, 1425 Lunt Avenue. P.O. Box 1426, Elk Grove Village, IL, 60007.

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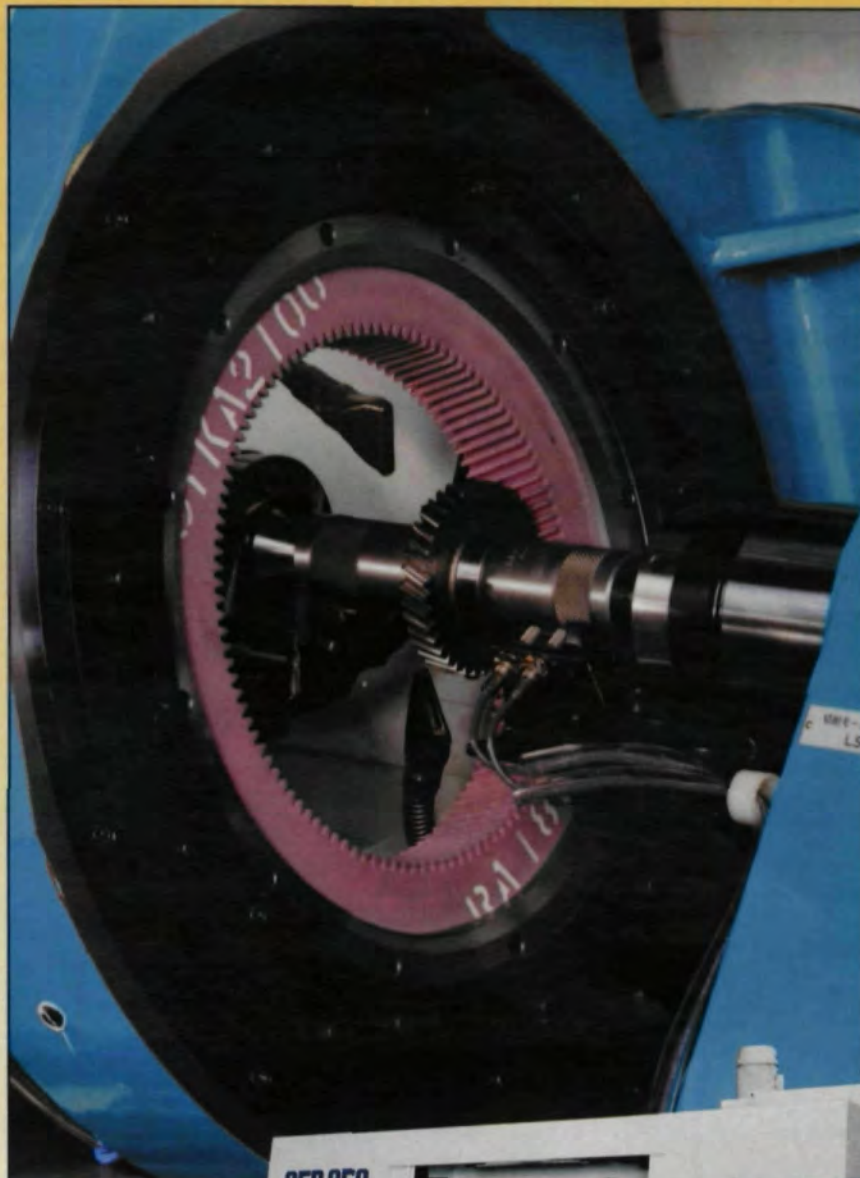

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A variety of tests are used by tooling experts to document hob accuracy. The most common test employed to meet AGMA, British and MCTI standards focuses on checking the lead in 3 revolutions (the maximum number of revolutions necessary to generate an involute profile). Although effective, this particular test does not incorporate hob profile errors in the result.

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This test is absolute assurance of the quality of the hob and its resultant impact on the generated gear profile. The Line of Action test is directly related to the total hob error—not including mounting errors, machine errors, etc. Once again, the involute profile on the generated piece part will be identical to the hob Line of Action. Shown above is the typical hob Line of Action test as well as the gear profile obtained with this hob.

While Line of Action testing is not universally acknowledged, it is specifically recognized by the DIN 3968 and ISO standards. It is an effective test because it's a documented measurement of the final result the end user experiences from the hob he purchased. In effect it is an absolute verification that the hob, when properly mounted in a rigid hobbing machine and running with a properly mounted piece part, will generate a tooth profile in accordance with the desired accuracy.

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"THE METROLOGY AND MOTION PEOPLE"

The Sum of Our Fears

"Prosperity is not without many fears and distastes; and adversity is not without comforts and hopes."

Sir Francis Bacon.

Last month I attended a meeting in Mexico City sponsored by CIATEQ, a quasi-governmental organization in Mexico, which has as one of its aims the encouragement of the growth of the gear industry in Mexico. The purpose of the meeting was to provide a catalyst among the attendees to form a Mexican equivalent of AGMA and to encourage an alliance with AGMA. Joe Franklin, the Executive Director of AGMA, Bill Boggess, the President, Vice-President Ray Haley, and I were among the few Americans at the meeting.

Not surprisingly, one of the major topics of discussion at the meeting was the NAFTA - the North American Free Trade Agreement, which will probably be passed by the U.S. Senate, if not this year, then in 1993. While the details of the treaty are still vague in many respects, the general thrust of the arrangement is quite clear. If NAFTA is approved, trade barriers between the United States, Canada, and Mexico will gradually be all but eliminated. And the question on the minds of all of us at the meeting was the same, what will this mean for our business?

The interesting thing to me was that regardless of which side of the border we resided on, our hopes and, particularly our fears, were almost identical. U.S. businesses are concerned, rightly enough, that free trade with Mexico and Canada will mean a loss of jobs in the U.S. We see Mexico as a giant to the south with vast quantities of cheap labor that will be able to manufacture goods at far lower prices than we can. How will we be able to compete with a country that pays its workers less than half of what we do?

On the other hand, our Mexican counterparts feel dwarfed by what they imagine as an industrial colossus on their northern border, filled with state-of-the-art factories capable of 24-hour shifts staffed by robots, CNC machines, and the latest in high-tech wonders they cannot hope to afford to purchase for years. They ask themselves, how can we hope to compete against that?

The reality on both sides of the border, of course, falls somewhere in between these two "worst case" visions. If and when the trade

PUBLISHER'S PAGE

Attending the CIATEQ meeting in Mexico City were left to right: GT publisher, Michael Goldstein, AGMA President, Bill Boggess, Dr. Juventino Balderas Moreno of the Mexican delegation to the NAFTA talks, and AGMA Executive Director, Joe Franklin, Jr.

barriers between our countries come down, there will be winners and losers on both sides of the Rio Grande. The customary, comfortable ways of doing things both here and in Mexico will of necessity have to change. We'll all have to learn some tough new lessons.

At times like these, when a multitude of conflicting interests all demand to be heard, and a variety of conflicting needs have to be met, it is always tempting to act out of our fears rather than out of our hopes. But is doing so the best way to meet the challenge of an industrially growing Mexico?

A look at the positive side is in order. Since 1988, U.S. direct investment in Mexico has grown from $5.5 billion to $9.4 billion. Two-way trade between the two countries was $64.5 billion last year. That's a lot of dollars and pesos changing hands, and people on both sides of the border are benefitting - even before the NAFTA is in place. And access to a share of those dollars and pesos is not limited to a fortunate few. It is open to any businessperson willing to do what it takes to become involved.

Successful trading with Mexico is governed by the same rules as successful exporting anywhere. Besides the necessary capital and willingness to take a risk, the successful exporter needs time, imagination, flexibility, and openness to new ideas: Time to learn the country, the language, the business culture, the needs of the local markets, and to let investments grow and

PUBLISHER'S PAGE

"No nation was ever ruined by trade."

Benjamin Franklin.

develop at a reasonable pace, which may be slower than the accustomed one; imagination to see how a particular business can fit into the local environment; flexibility enough to alter plans that don't work or don't fit the changing circumstances across the border; openness to ways of thinking and doing business that aren't "the way we've always done it back home."

It is also important to remember that exporting does not have to be an all or nothing proposition. By means of local representatives, joint ventures, and other kinds of cooperative arrangements, businesses on both sides of the border can get their feet wet in the export market without having to go in at the deep end of the pool. In cooperative ventures, each member can play to its own strengths, whether those be extremely competitive labor, state-of-the art equipment, valuable local knowledge, or a well-developed customer base. Each partner can continue to service his or her own customer base, do business in his native tongue, and follow local customs. At the same time, each partner has vastly expanded the kind of products and services he can offer.

Finding the perfect match for a trading partner is hard work. It has been suggested that finding the right partner for overseas joint ventures may be the management challenge of the 1990s. That may be overstating the case, but finding the right partner is key to success in joint ventures. To find the right partner with whom to do business across the border, you have to travel, meet with many of your counterparts in the other country, research the potential partners, learn as much as you can about the local economy; in short, you have to do a lot of homework and be open to new possibilities. Perhaps this is a place where AGMA and CIATEQ can help by arranging meetings where potential investment partners can meet and get to know one another and the kinds of opportunities available on both sides of the border.

Success in the new global economy demands that we operate out of our hopes rather than our fears and see the promise offered by trade with our neighbors rather than only the threat. Living next to a strong, economically healthy neighbor with industries that compete directly with ours is without a doubt more of a challenge than living next to one with no such industries. But such strong neighbors also provide better markets for our products and a wealth of opportunities for the shrewd and imaginative businessperson.

To respond to the new global economic realities out of the sum of our fears is to envision a nightmare world which doesn't really exist. Making business plans to cope only with this nightmare in the end does us more harm than good. To respond out of our hopes gives us the chance to evaluate our fears in the clear light of day and deal with them - and with the opportunities that exist side by side with them - in a way that can benefit us all.

Michael Goldstein
Publisher/Editor-in-Chief

Camshaft Gears

William L. Janninck

One of our readers in England has asked for our help in locating published technical data and information on the design, manufacture, and inspection of camshaft gears. Although millions of these gears have been made and are in constant use, we are not aware of any formal material having been published. We would be pleased to hear from anyone who has knowledge of such information.

The camshaft gear gets its name from the fact that it is located on an engine camshaft where it is usually nested tightly between the cam lobes or bearing journals. The gear is of a high helix angle and drives a much smaller diameter gear which turns a distributor or oil pump. Both gears have the same number of teeth, rotating on a 1-to-1 ratio. Since these gears operate at a 90°-axis angle, the distributor gear has a complementary helix angle to the camshaft gear helix. Fig. 1 illustrates the arrangement of these gears.

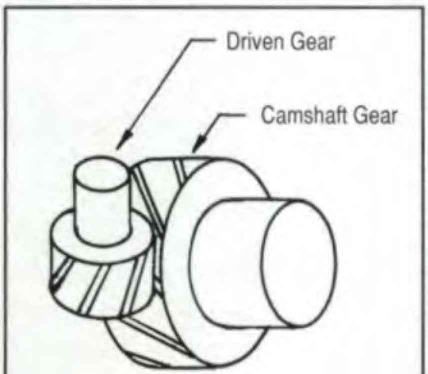

Fig. 1

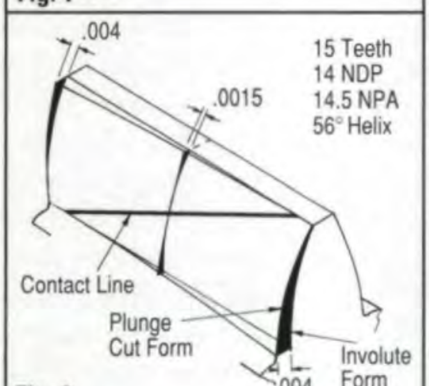

Fig. 2

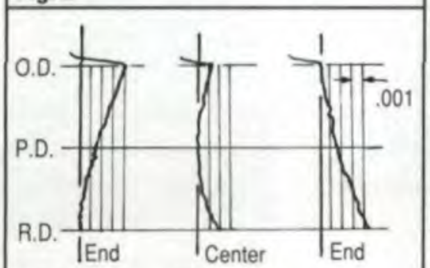

Fig. 3 Profile Check

Since the distributor gear is a conventional helical gear, it is usually manufactured by the finish hobbing process. The camshaft gear, however, being confined to available space on the shaft, poses a special problem. There is usually insufficient space for tool travel to completely generate a true helical gear, and the compromise is to plunge cut the gear with as large a hob as possible without damaging the adjacent cam or journal surfaces. This leaves a non-involute gear which is totally functional in use, but presents another problem, that of gear inspection.

Fig. 2 shows a comparison of the true involute helicoid surface with that actually cut by infeed hobbing. The dimensions shown on the sketch show the material that is left on the gear relative to the involute helicoid form. Along the diagonal line of contact swept out, the form is correct, and the set functions properly. Fig. 3 shows the involute trace on the gear flank at end, center, and end. The center plane will show a slight hollow form, not a true involute. Lead checks will also reflect the variations along the flank from top to root. The same geometry occurs on the opposite flank. ■

SHOP FLOOR

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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 Illinios Tools - ITW, Inc. He is the author of numerous articles on gear-related topics.

Non-Involute Hobbed Forms

Thomas D. Ware
Starcut Sales, Inc., Farmington Hills, MI

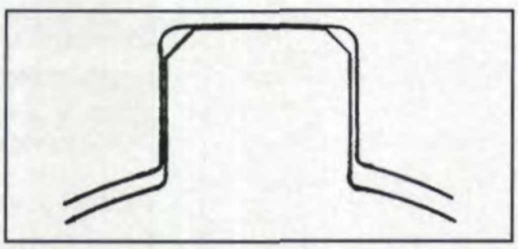


Fig. 1 - Major diameter fit.

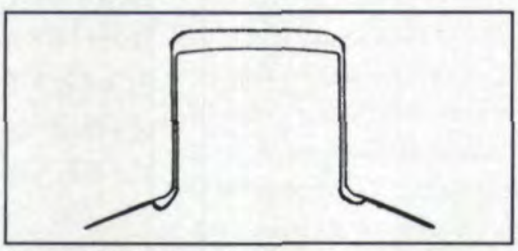


Fig. 2 - Minor diameter fit.

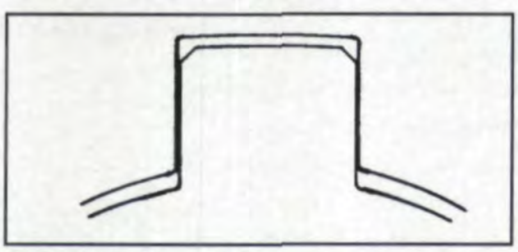
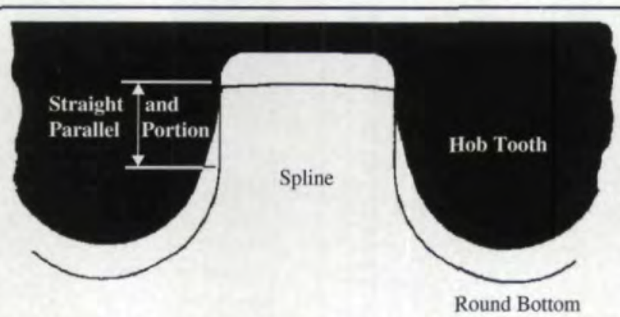


Fig. 3 - Side fit.



Straight-sided splines have maximum strength when a full-fillet design is utilized. Such full fillets (round bottoms) can be simply reproduced by the hobbing tool. The fillet on the tool differs from that on the produced fillet on the spline.

Fig. 4 - Straight side round bottom.

Although hobbing is usually associated with the manufacturing of involute gears and splines, many other tooth forms can be produced by a generating hob. A generating hob is defined as one whose tooth form is not reproduced on the part. These tooth forms must be equally spaced about an axis, but do not have to be symmetrical. The types of parts this article will cover are parallel key splines, serrations, sprockets, ratchets, and special tooth forms.

Parallel Key Splines

This is the most common non-involute tooth form. A parallel key spline can be made for major diameter fit, minor diameter fit, or side fit. (See Figs. 1-3.) These fits are divided into three classes: permanent, sliding without load, and sliding with load. The different fits and classes require different clearances between the spline tooth and the mating part.

The hobs can be designed to produce different features in the spline root or the major diameter. These features are full fillet (round-bottom), flat root, and clearances grooves. (See Figs. 4-6.) Clearance grooves are produced on the workpiece by lugs on the tip of the hob teeth. Non-topping, semi-topping, full topping, and shoulder clearance are hob features that affect the major diameter of the spline.

All parallel key spline hobs are single-purpose tools designed for a specific number of teeth, key width, and diameter. The tools are supplied in three quality levels: A, C, and D. The quality level is controlled by the piece part data. A sample is cut with each hob and inspected to assure tool quality. Normally this coupon is shipped with each hob.

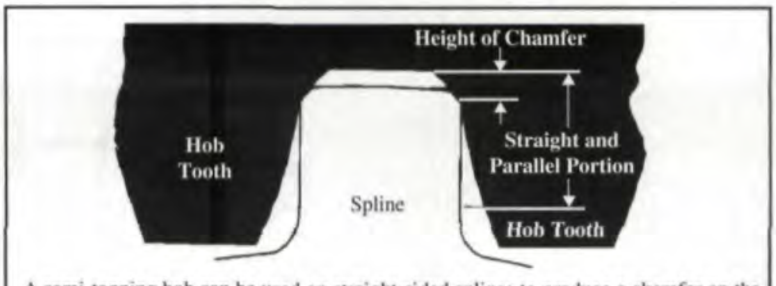
ANSI Standard B94.7-1980 has tolerances for hob manufacturers' test cuts. Part A is for spline tooth thickness held. Part B is for minor diameter held. The standard covers 6-, 10-, and 16-tooth splines. Part C controls the maximum undercut (per side). Hobs are also designed and built to specific customer part tolerances.

B94.7-1966 also established standard hob sizes for single-thread tools based on the number of splines, depth of cut, and part diameter. These sizes are recommended minimums based on an acceptable number of generating flats and regrindable length. This diameter can be restricted by sweep-out radius limitations. Multiple-thread hobs can also be used to cut parallel key splines, but as the number of generating teeth are reduced, the surface finish roughness and chip load will increase.

Two of the special features need some explanation. Lugs are used only as a last resort to obtain the required straight and parallel depth and not completely remove the minor diameter. Because of their shape and location, these lugs wear very rapidly and must be sharpened often. The space width must be large enough to leave a sufficient minor diameter bearing. (Fig. 6)

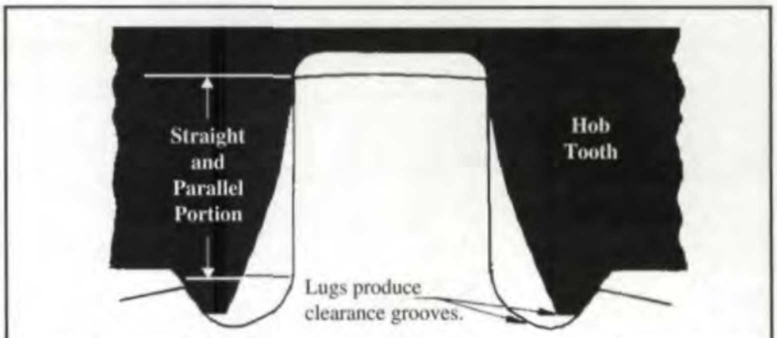
Shoulder clearance type hobs are used when a full-depth spline is required up to the backing shoulder. The hob tooth is of sufficient depth to clear the shoulder diameter. In the shoulder portion of the shaft, the key profile will consist of a straight side to the outside diameter of the spline, topped by an involute curve generated by the straight or slightly angled clearance groove. (Fig. 7)

The hob tooth form is a curve. Its shape is determined by the generating diameter selected by the hob manufacturer. Normally it is at the O.D. of the spline to allow as much cutting clearance in the hob as possible. (See Figs. 8-9.) This dimension must be balanced with the height of the start of straight-sided diameter. Approximately one quarter of the whole depth will be fillet radius. Once the hob design is established for a specific tooth thickness and root diameter, the tool will not accurately cut a different tooth thickness or root diameter. If the hob is sunk-in deeper, as in



A semi-topping hob can be used on straight-sided splines to produce a chamfer on the outside corners of the spline teeth.

Fig. 5 - Semi-topping flat root.



Proper fit of straight-sided spline members is achieved by hobbing a clearance groove in the area where the straight side meets the minor diameter of the spline. Small projections can be provided on the corners of the hob teeth to provide a blended clearance groove.

Fig. 6 - Straight side clearance groove.

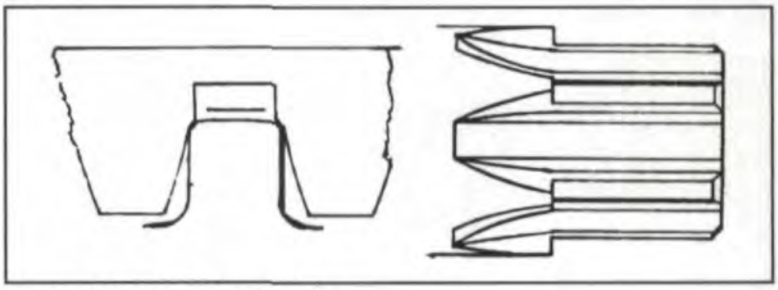


Fig. 7 - Shoulder clearance hob.

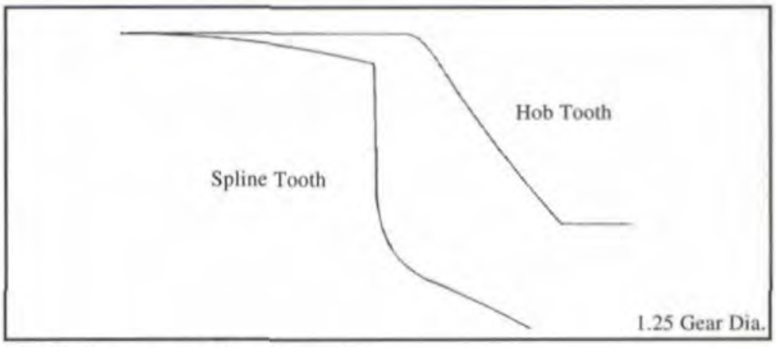


Fig. 8 - Large generating diameter.

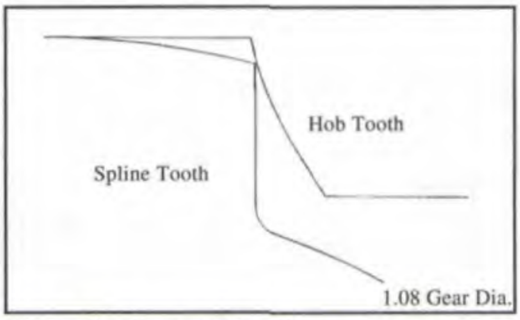


Fig. 9 - Small generating diameter.

Thomas D. Ware
is the Chief Engineer, Gear Tools Department, at Starcut Sales, Inc., Farmington Hills, MI. He is the author of several papers on gearing subjects.

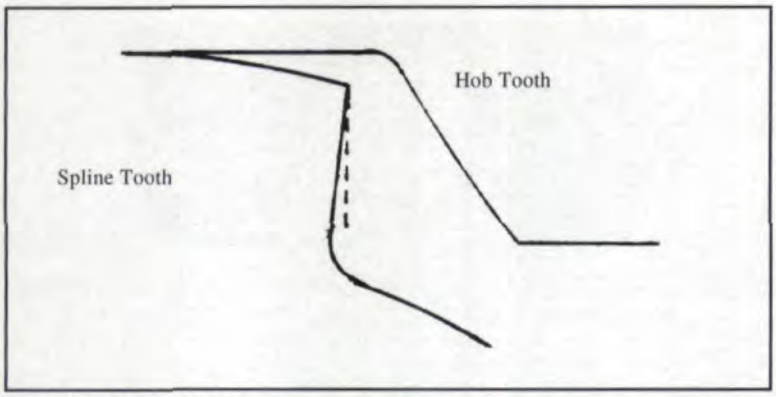


Fig. 10 - "Sunk in" hob.

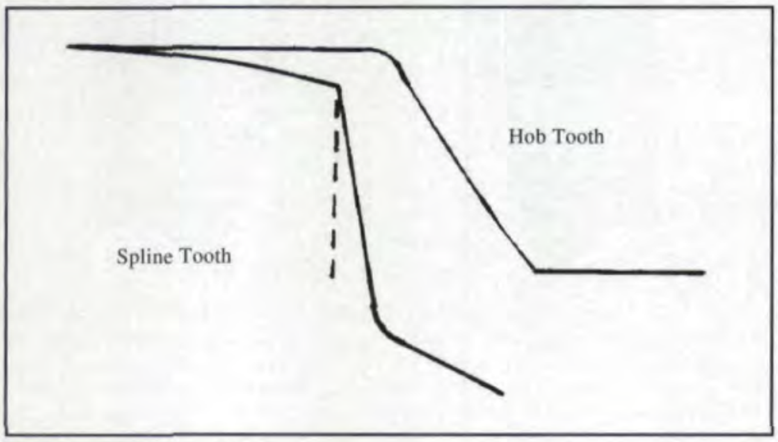


Fig. 11 - "Pulled out" hob.

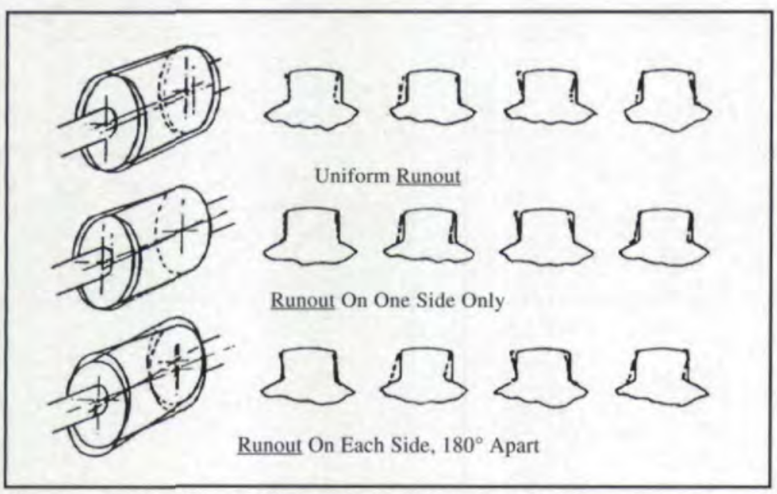


Fig. 12 - Sharpening problems: Illustrations showing three types of runout on a spline hob and their effect on splines.

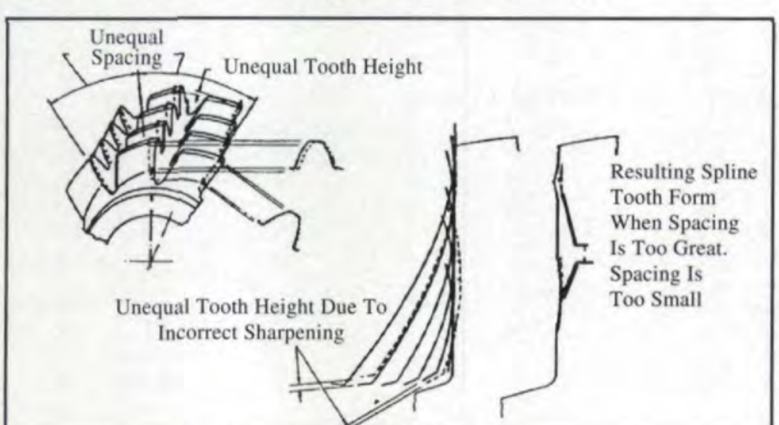


Fig. 13 - Sharpening error: cutting faces of spline hob unevenly spaced.

Fig. 10, it will produce a tapered tooth minus at the root. If it is pulled out, as in Fig. 11, the spline will be tapered with plus tooth thickness at the outside diameter.

Sharpening errors and mounting runout can also adversely affect the quality of the spline. Parallel key splines are very sensitive to errors due to the nature of the generating action of the hob. Uniform or single direction runout will cause the keys to become undercut on both sides or tapered on both sides. With one-end runout or wobble, the key will be undercut on one side and tapered on the other side. (Fig. 12)

Excessive flute spacing error will generate either a bump or a hollow on the sides of the teeth. (See Fig. 13.) The root diameter will also have high and low flats and, if the spacing error is progressive, it may show up as an eccentric root diameter. Negative rake error produces a tapered tooth and a plus root diameter. (See Fig. 14.) Positive rake or hook cuts a key with undercut and an undersized root diameter. (See Fig. 15.) Adjusting the hob to cut the proper root diameter only magnifies the taper or undercut problems.

It is also important to remember that as the hob tip wears, the fillet radius becomes larger and can cause interference.

Serrations

Straight-sided serrations or straight-sided splines are similar to parallel key splines in that they have a flat tooth surface that is generated with a curve-sided, single-purpose hob. (See Fig. 16.) The obvious difference is the sides of the teeth are at an angle to one another rather than parallel.

The SAE has a standard for 90° included angle straight-sided serrations. (See Fig. 17.) The standard is based on nominal outside diameters for 36 and 48 teeth only. Side fit is the only type of fit specified. Allowable errors and machining tolerances are the same as an equivalent pitch involute serration. To insure that the hobs will produce the part within tolerances, a sample is cut and inspected. For finer pitch hobs, an involute serration hob can be used in place of the curved form hob.

As with the parallel key spline, when the straight-sided serration hob is "sunk-in" or pulled out, the angle and form will change, as shown in Fig. 18. It is possible by using an intermittent hob design to produce a sharp point at the O.D., but a practical limit of .003R is obtainable in the root.

Sprockets

There are many different types of sprockets. They are roller chain, silent (inverted tooth), synchronous belt drive, hinge type, block chain, and film sprocket. All have a non-involute tooth form. Most are covered by an ANSI standard or a special application design.

Roller Chain Sprockets. Most roller chain sprockets are manufactured to the ANSI B29.1 "Power Transmission Roller Chains" standard. (Figs. 19-20) These sprockets are specified in two quality levels, commercial and precision. Precision sprockets are used for high speed and high loads, critical timing, or accurate register. The sprocket can be a Type I (with pitch line clearance) or Type II (without pitch line clearance). Another common standard is the British Standard BS 228. The main difference from the ANSI standard is the roller size. A number of special roller chain sprocket tooth forms are in use.

Standard sprockets are classified by chain number. This number specifies both the pitch and roller diameter. Minimum hob sizes are also based on the chain number for both single and multiple thread hobs and are found in ANSI B94.7-1966.

Sprockets can be cut with either a full topping or non-topping hob. Only one non-topping hob is needed to cut all numbers of teeth of a specific chain number. A topping hob can cut only a small range of teeth without exceeding the standard tolerances. As the number of teeth increase, the whole depth must be increased. These ranges are 7-8 teeth, 9-11 teeth, 12-17 teeth, 18-34 teeth, and 35 and over.

The most critical features of roller chain sprockets are tooth spacing, seating curve radius, and root diameter. When single-thread hobs are used, tooth spacing is primarily dependent upon the hobbing machine, and the root diameter and seating curve radius will be uniform, since all the teeth are finished by the same group of hob teeth. Multi-thread hobs with a prime thread-to-tooth ratio will produce accurate tooth spacing and a uniform

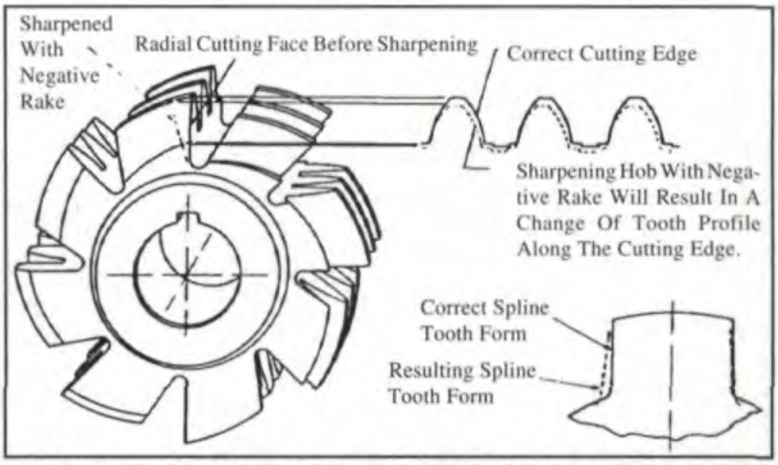


Fig. 14 - Sharpening error: cutting face of spline hob ground with negative rake when it should be radial.

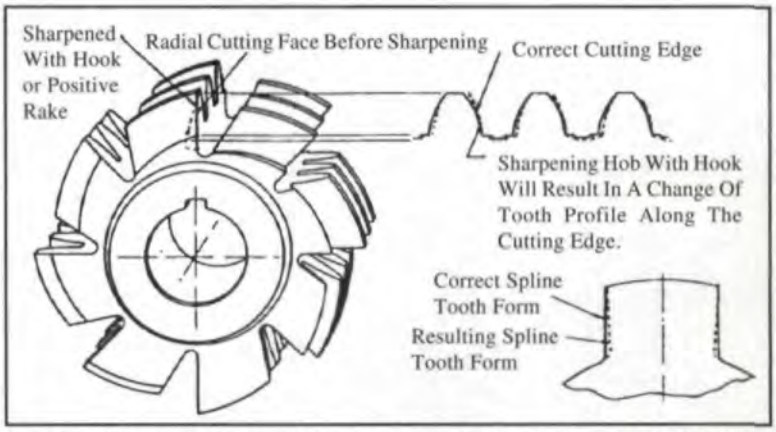


Fig. 15 - Sharpening error: Cutting face of spline hob ground with hook when it should be radial.

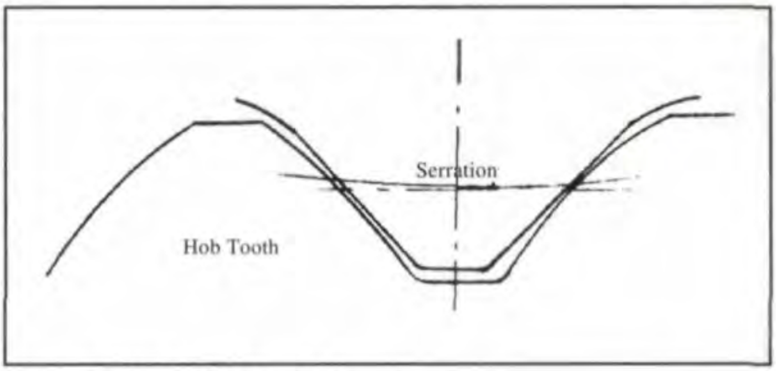


Fig. 16 - Serration tooth and hob.

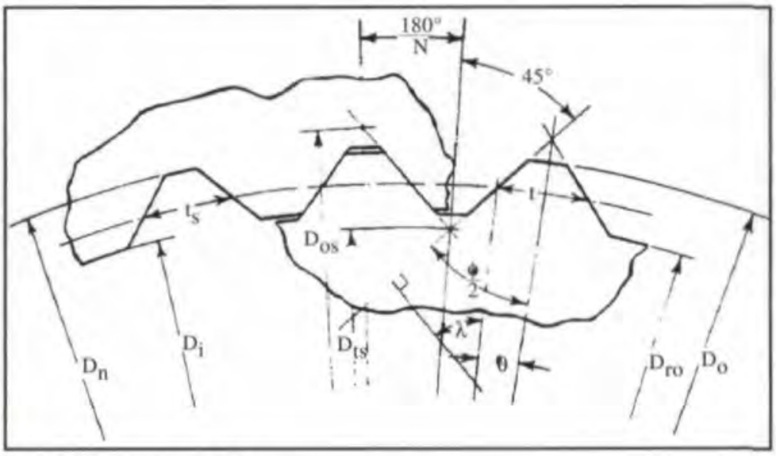


Fig. 17 - Serration tooth.

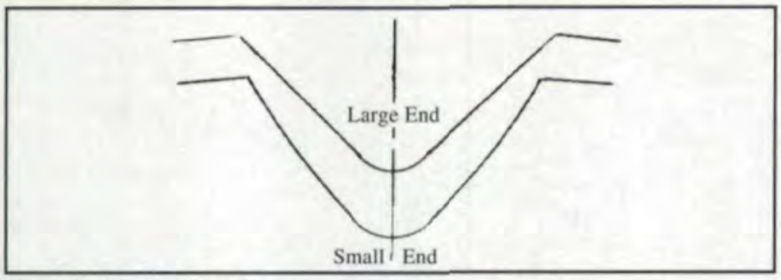


Fig. 18 - Tapered depth.

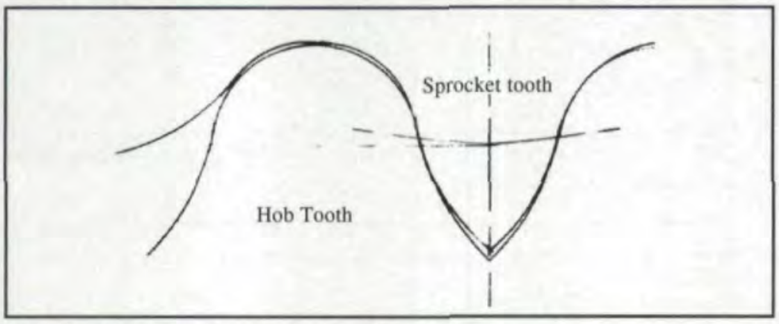


Fig. 19 - Roller chain sprocket tooth and hob.

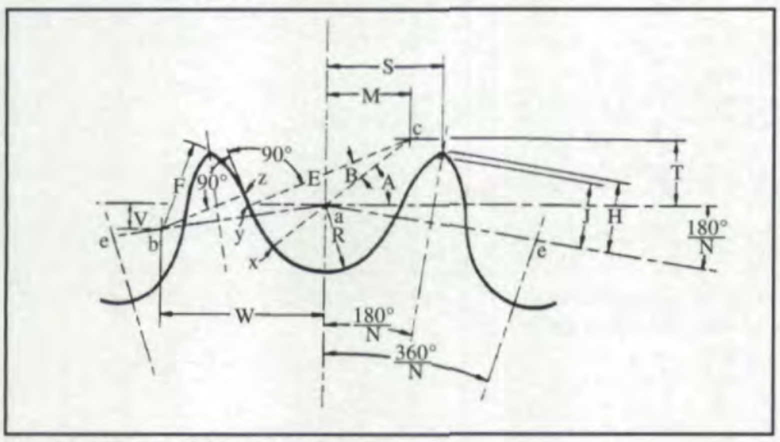


Fig. 20 - Roller chain sprocket tooth.

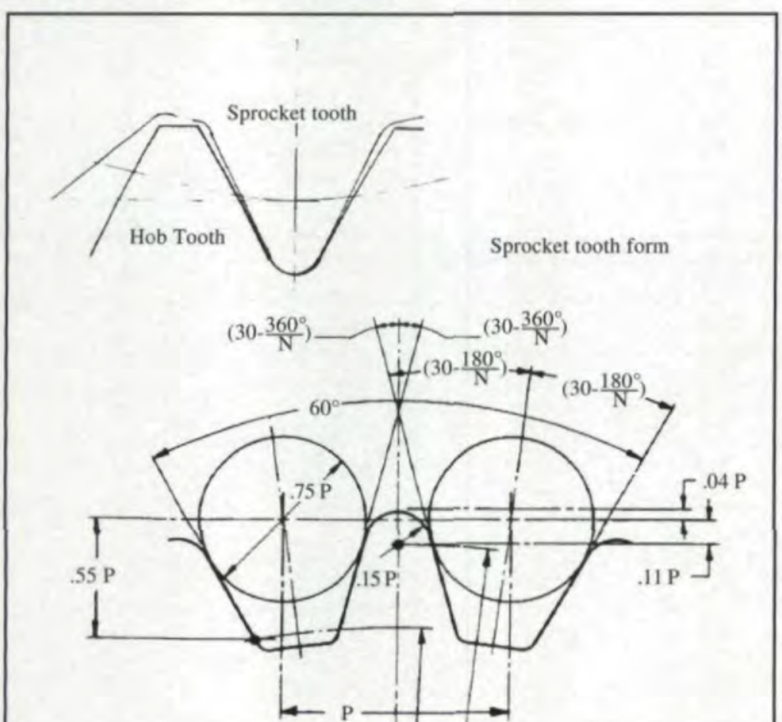


Fig. 21 - Silent chain sprocket teeth.

root diameter, but the seating curve radius may vary. An even thread-to-tooth ratio will produce an accurate seating curve radius, but the tooth spacing may not be accurate, and the root diameter may not be uniform. Sharpening errors will affect the tooth form, but they are not as sensitive as gear or spline hobs.

Silent Chain Sprockets. Silent chain or inverted tooth sprockets are specified by ASA B29.2 "Inverted Tooth (Silent) Chains and Sprocket Teeth" or special tooth forms. (Fig. 21) The chain links have no sliding action on or off the teeth. This results in a smooth, practically noiseless action. Standard sprocket teeth have straight sides, requiring hobs with curved teeth. Theoretically, each number of teeth requires a different hob, but tolerances are broad enough to allow ranging. The range of teeth for the hobs are 17-23, 24-32, 33-43, 44-58, 59-79, 80-110, and 111-150.

Each standard pitch is specified by a chain number, designated SC*n*. The number is the pitch in eighths of an inch; i.e., SC5 = 5/8" pitch.

Tooth spacing and tooth profile are the most critical features of a silent chain sprocket. Accurate tooth spacing is needed because of the number of teeth engaged at one time. Tooth form must be held (with no concavity) to eliminate chain whip. Of the sprocket type hobs, the inverted tooth hob is most sensitive to sharpening errors.

Synchronous Belt Sprockets. Synchronous belt or timing belt sprockets come in many different tooth forms. (See Fig. 22.) A sampling of the various types are Trapezoidal (ANSI/RMA IPZ4) shown in Fig. 23 and Synchroflex (DIN 7721) shown in Fig. 24. These sprockets are specified in public standards. The balance were developed by different manufacturers and are patented. Some of the tooth forms are Gilmer, Goodyear, Gates Rubber (HTD and Poly Chain GT), Dodge-Reliance Electric (Dynasink), Dayco, and Pirelli.

The tooth forms vary from a straight-side to smooth radius to complex mathematical curves. Each comes in various pitches, most of which are metric. One common factor is that the pitch diameter is above the outside diameter of the sprocket. This matches the solid section of the belt.

Hobs used to generate timing belt sprockets are topping and can cut only a small range of teeth. A sample cut is included as part of the hob inspection process. The hobs used to manufacture the belt molds have a different tooth form than the sprocket hob.

Ratchets

The ratchet-type tooth form can be generated using a conventional multi-position hob. These ratchets must have a large fillet radius. (See Figs. 25-26.) The straight portion of the tooth face is approximately two-thirds of the whole depth. Also the tooth face may be radial, negative, or slightly positive. A sharp outside diameter can be produced with an intermittent or staggered-tooth hob. Ratchets with hooked faces, small fillet radii, or nearly parallel sides require a special single position hob. (Figs. 27-28)

Special Tooth Forms

Non-circular shapes and unlike tooth forms can also be generated using a conventional multi-position hob. Examples of this type of part are squares and other poly-sided parts, ovals, semi-circular teeth, and slots and grooves with different tooth forms. As with all these types of hobs, they are single-purpose for the exact part and have no flexibility.

In conclusion, if the user is willing to sacrifice some flexibility, a generating hob can be used to cut much more than involute tooth forms. With proper care and the correct machine adjustments, such hobs can also cut parallel key splines, serrations, roller chain, silent chain, and synchronous belt sprockets, ratchets, and special tooth forms. ■

Acknowledgement: *From the Society of Mechanical Engineers Fundamentals of Gear Manufacturing Clinic, 1992. Reprinted with permission.*

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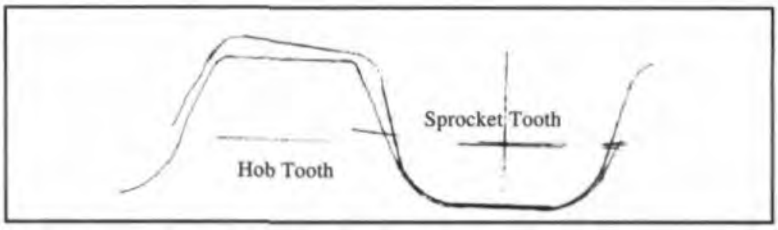

Fig. 22 - Timing belt tooth and hob.

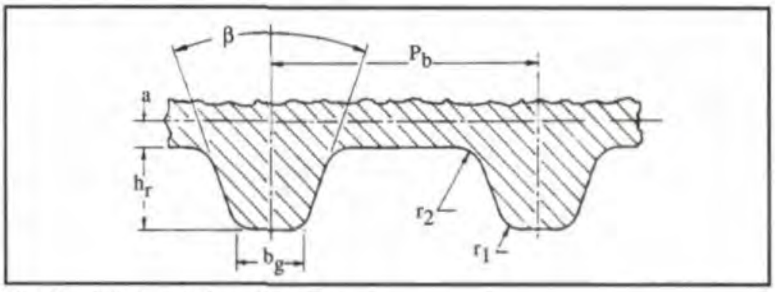

Fig. 23 - Timing belt rack teeth

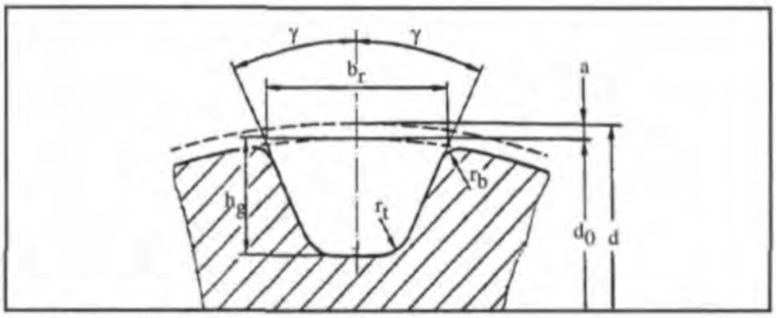

Fig. 24 - Timing belt sprocket tooth.

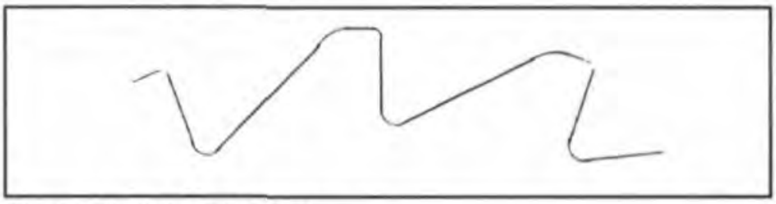

Fig. 25 - Typical ratchet tooth.

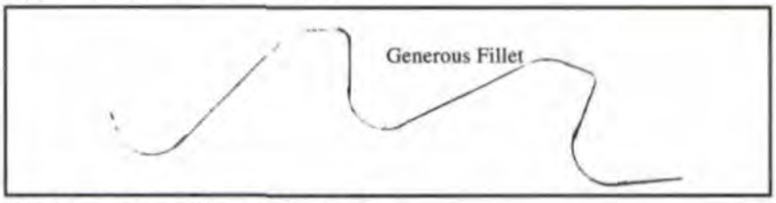

Fig. 26 - Large radius ratchet tooth.

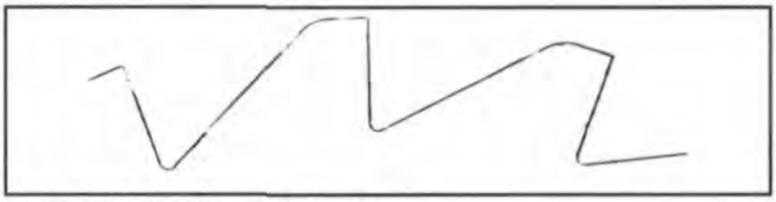

Fig. 27 - Small radius ratchet tooth.

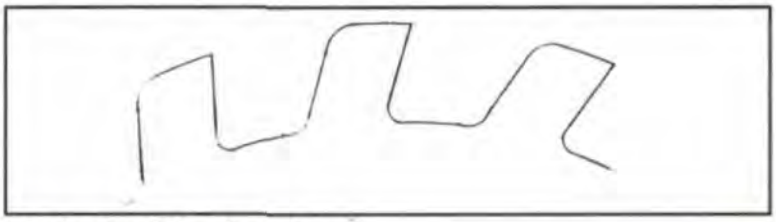

Fig. 28 - Parallel tooth ratchet.

1992 Marks Important Gear Design Milestone: Lewis' Bending Strength Equations Now 100 Years Old

Columbus' first voyage to the Americas is not the only anniversary worthy of celebration this year. In 1892, on October 15, Wilfred Lewis gave an address to the Engineer's Club of Philadelphia, whose significance, while not as great as that of Columbus' voyage, had important results for the gearing community. In this address, Lewis first publicly outlined his formula for computing bending stress in gear teeth, a formula still in use today.

While the Lewis equations are taken for granted today, a hundred years ago, there were at least 48 known "rules" for calculating bending strength, yielding values which varied as much as 500%. Lewis brought some order to this chaotic situation when he found that the load stress on a gear tooth could be compared to applying that same load onto a parabola of the same general size as the gear tooth. While it is virtually impossible to compute the stress on an involute curved member, such as a gear tooth, it is possible to compute the stress of a load imposed onto a parabola. It was on the basis of this observation that he developed the Lewis bending strength formula.

At the time he delivered his paper, "The Investigation of the Strength of Gear Teeth," to the Engineers' Club of Philadelphia, Lewis was employed by William Sellers & Co., of Philadelphia. He began his work there as a mechanic after his graduation from the Massachusetts Institute of Technology and subsequently served as a draftsman, designer, assistant engineer, and finally, director of the plant. In 1900 he became president of Tabor Manufacturing Company, where he remained until his retirement.

Lewis was the author of numerous books, papers, and over 50 inventions. In 1927 he received the Medal of the American Society of Mechanical Engineers for his contributions in the field of gearing. He won the Franklin Institute's Longstreth Medal twice; in 1899 for his invention of an "inertia indicator" and in 1927 for a "shockless jarring machine." He died in 1929 on the return trip from Tokyo, where he had attended a convention of the World Engineering Conference, and was buried at sea.

The editors wish to thank ***William R. Rollins*** *for bringing the Wilfred Lewis paper to our attention and for his help with the research and preparation of the article. Mr. Rollins is the principal in Aerospace Planetary Gear Consultants, a gear research, design, and development firm in South Windsor, CT.*

Investigation of the Strength of Gear Teeth

Wilfred Lewis, Member, Engineers' Club of Philadelphia
Read October 15, 1892

To mechanical engineers, the strength of gear teeth is a question of constant recurrence, and although the problem to be solved is quite elementary in character, probably no other question could be raised upon which such a diversity of opinion exists, and in support of which such an array of rules and authorities might be quoted. In 1879, Mr. John H. Cooper, the author of a well-known work on "Belting," made an examination of the subject and found there were then in existence about forty-eight well-established rules for horsepower and working strength, sanctioned by some twenty-four authorities, and differing from each other in extreme cases of 500%. Since then, a number of new rules have been added, but as no rules have been given which take account of the actual tooth forms in common use, and as no attempt has been made to include in any formula the working stress on the material so that the engineer may see at once upon what assumption a given result is based, I trust I may be pardoned for suggesting that a further investigation is necessary or desirable.

In summing up his examination, Mr. Cooper selected the following formula from an English rule published at Newcastle-under-Lyme in 1868, and, as an expression of good general averages, it may be considered passably correct.

$$X = 2{,}0000\,p\,f$$

in which

X = breaking load of tooth in pounds
p = pitch of teeth in inches
f = face of teeth in inches

In conclusion he makes this pertinent observation: "It must be admitted that the shape of the tooth has something to do with its strength, and yet no allowance appears to have been made by the rules tabulated above for such distribution of metal, the breaking strength being based upon the pitch or thickness of the teeth at the pitch line or circle, as if the thickness at the root of the tooth were the same in all cases as it is at pitch line." Notwithstanding the fact that the necessity for considering the form of the tooth, as well as its pitch and face, was thus clearly set forth over thirteen years ago, I am not aware that anyone else has taken the trouble to do it, and, as the results to be presented have been well-tested by experience in the company with which I am connected, I believe they will be of interest and value to others.

In estimating the strength of teeth, the first question to be considered is the point or line at which the load may be applied to produce the greatest bending stress. In the rules referred to, the load is sometimes assumed to be applied at the pitch line, sometimes at the end of the tooth, and sometimes at one corner; but in good modern machinery, the agricultural type excepted, there is seldom any occasion to assume that the load is not fairly distributed across the teeth. Of course, it may be concentrated at one corner as the result of

While it is impossible to compute the load stress on an involute curved gear tooth, it is possible to compute the stress of a load imposed onto a parabola. This observation is the basis for the Lewis bending strength formula.

careless alignment or defective design, in which the shafts are too light or improperly supported, and, for a rough class of work, allowance should certainly be made for this contingency, but, in all cases where a reasonable amount of care is exercised in fitting, a full bearing across the teeth will soon be attained in service. It must be admitted, however, that on account of the inevitable yielding of shafts and bearings, even of the stiffest construction, the distribution of pressure may not be uniform under variable loads, and that the assumption of uniform pressure across the teeth is not always realized in the best practice. To what extent it is realized I shall not attempt to estimate, but in general practice, where the width of the teeth is not more than two or three times the pitch, the departure can not be regarded as serious. The conclusion is therefore reached that in first-class machinery, for which the present investigation is intended, the load can be more properly taken as well-distributed across the tooth than as concentrated at one corner. Having thus disposed of the first question, the second is, at what part of the face should the load be assumed to be carried in estimating the strength of a tooth?

Evidently the load may be carried at any point within the arc of action, and it might be argued that when a tooth is loaded at its end, there are always two teeth in gear, and that the load should be divided between them. This is theoretically true of all teeth properly formed and spaced, but it must be admitted that mechanical perfection in forming and spacing has not yet been reached, and that the slightest deviation in either respect is sufficient to concentrate the whole load at the end of a single tooth. In time this concentration may be relieved by wear, but it is not so easily corrected as unequal distribution across the teeth, and, as the present practice of cutting gears with a limited number of equidistant cutters makes it almost impossible to obtain teeth of proper shape, it is evident that the load cannot safely be assumed as concentrated at a maximum distance less than the extreme end of the tooth. In some cases, of course, the teeth will not be so severely tested, and the error in this assumption compensates in a measure for the error in the first assumption of equal distribution across the teeth. Having thus concluded that gear teeth may fairly be considered as cantilevers loaded at the end, the influence of their form upon their strength remains to be disposed of.

In interchangeable gearing, the cycloidal is probably the most common form in general use, but a strong reaction in favor of the involute system is now in progress, and I believe an involute tooth of 22-1/2° obliquity will finally supplant all other forms. There are many good reasons why such a system should be generally adopted, but it is not my purpose at present to discuss the merits and defects of different systems of interchangeable gearing, and I now propose to explain how the factors given in the table herewith were determined by graphical construction.

A number of figures were carefully drawn on a large scale, to represent the teeth cut by a complete set of equidistant cutters, making the fillets at the root as large as possible to clear an engaging rack. See Plate 1.

The addendum was $.3p$ and root $.35p$, as shown in the illustrations, and the clearance was $.02p$.

When the load is applied at the end of a tooth and normal to its face in the direction ab, it may be resolved into two forces, one tending to crush the tooth, and the other to break it across. The radial component, which tends to crush the tooth directly, has but a slight effect upon its strength. In material which is stronger in compression than in tension, the transverse stress due to the other component W is partially counteracted on the tension side, and the teeth are stronger by reason of their obliquity of action; but in material which is weaker in compression the reverse is the case, and, in general, it may be said that the strength of teeth will not be affected more than 10% either way by the consideration of this radial component.

It should therefore be understood that for the sake of simplicity, the factors given have been determined only with reference to the transverse stress induced by the force W, which may be regarded as the working load transmitted by the teeth. This load is applied at the point b, but it does not at once appear where the tooth is weakest, and, to determine that point, advantage is taken of the fact that any parabola in the axis *be* and tangent to *bW* at the point *b* encloses a beam of uniform strength. Of all the parabolas that may thus be drawn, one will be tangent to the tooth form, and it is evident that the point of tangency will indicate the weakest section of the tooth. In the rack tooth of 20° obliquity, this is found at once by prolonging *ca* to its intersection *g* with the center line *fb*, and laying off $bf = bg$; and in other cases, the weakest section *cd* may be found tentatively to a nice degree of accuracy in two or three trials. Having found the weakest section, the strength at that point is also determined graphically by drawing *bc* and erecting the perpendicular *ce* to intersect the center line in *e*. Then *ef* or *x* is taken to measure the strength of the various forms of teeth.

To understand the reason for this construction and the actual relation which the distance *x* bears to the strength of the tooth, it will be observed that the bending moment W*l* on the section *cd* is resisted by the fiber stress *s* into one-sixth of the face *f* times the square of the thickness *t*, or, by the well-known formula for beams, we have

$$W\,l = \frac{s\,f\,t^2}{6}, \text{ or}$$

$$W = \frac{s\,f\,t^2}{6\,l} \qquad (1)$$

But by similar triangles

$$x = \frac{t^2}{4\,l} \qquad (2)$$

and substituting this value in Equation 1 we have

$$W = s\,f\,\frac{2\,x}{3}, \qquad (3)$$

or we may write

$$W = s\,p\,f\,\frac{2\,x}{3\,p} \qquad (4)$$

The factor $\frac{2x}{3p}$ or *y* is determined by graphical construction and is given in Table 1 for convenient reference. This is multiplied by the pitch face and fiber stress allowable in any case when the working load *W* is to be determined. What fiber stress is allowable under different circumstances and conditions cannot be definitely settled at present, nor is it probable that any conclusions will be acceptable to engineers unless based upon carefully made experiments. In the article referred to, certain factors are given as applicable to certain speeds, and in the absence of any later or better light upon the subject, Table 2 has been constructed to embody in convenient form the values recommended. It cannot be doubted that slow speeds admit of higher working stresses than high speeds, but it may be questioned whether teeth running at 100 feet a minute are twice as strong as at 600 feet a minute, or four times as strong as

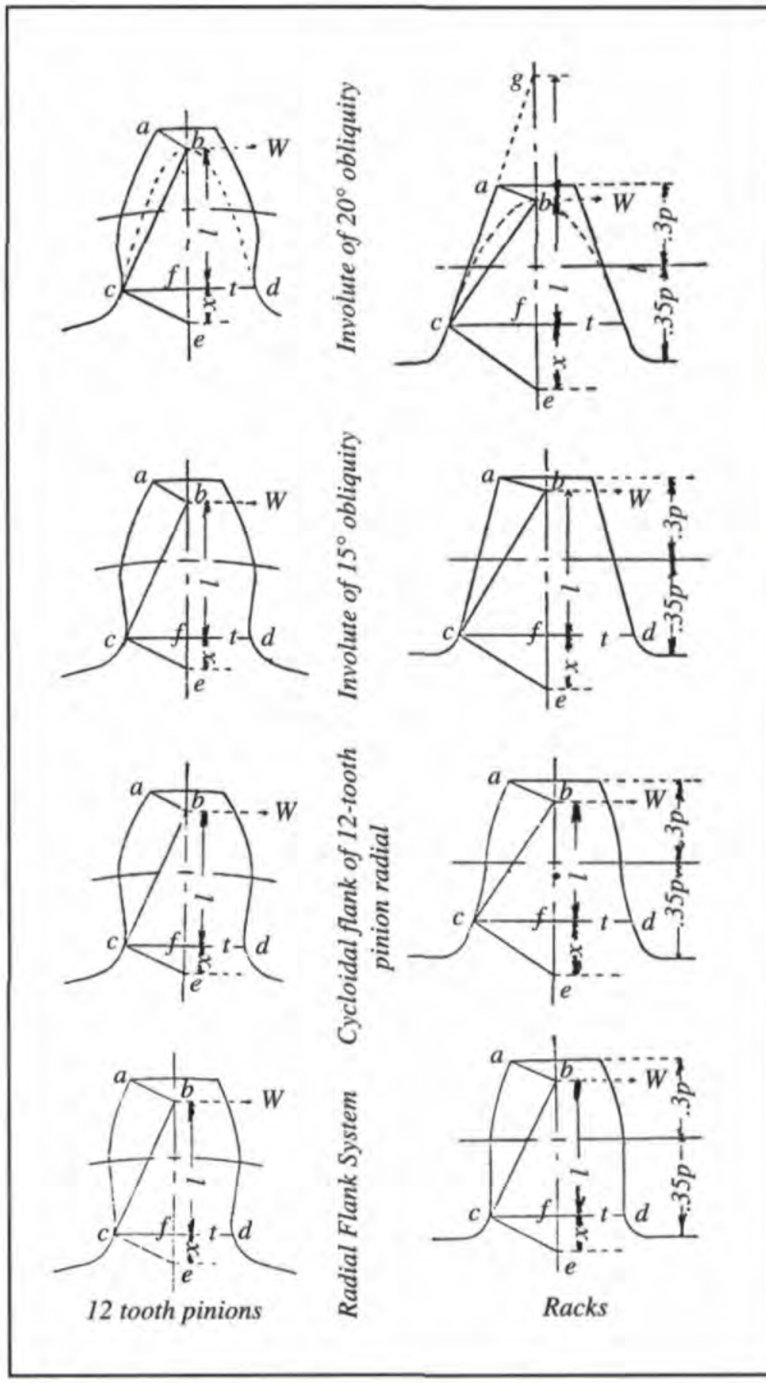

Plate I

TABLE 1			
Factor for Strength, *y*			
Number of Teeth	Involute 20° Obliquity	Involute 15° and Cycloidal	Radial Flanks
12	.078	.067	.052
13	.083	.070	.053
14	.088	.072	.054
15	.092	.075	.055
16	.094	.077	.056
17	.096	.080	.057
18	.098	.083	.058
19	.100	.087	.059
20	.102	.090	.060
21	.104	.092	.061
23	.106	.094	.062
25	.108	.097	.063
27	.111	.100	.064
30	.114	.102	.065
34	.118	.104	.066
38	.122	.107	.067
43	.126	.110	.068
50	.130	.112	.069
60	.134	.114	.070
75	.138	.116	.071
100	.142	.118	.072
150	.146	.120	.073
300	.150	.122	.074
Rack	.154	.124	.075

TABLE 2
Safe Working Stress, *s*, for Different Speeds

Speed of Teeth in Ft. per Minute	100 or less	200	300	600	900	1200	1800	2100
Cast Iron	8,000	6,000	4,800	4,000	3,000	2,100	2,000	1,700
Steel	20,000	15,000	12,000	10,000	7,500	6,000	5,000	1,300

the same teeth at 1,800 feet a minute. For teeth which are perfectly formed and spaced, it is difficult to see how there can be a greater difference in strength than the well-known difference occasioned by a live load or a dead load, or two to one in extreme cases. But for teeth as they actually exist, a greater difference than two to one may easily be imagined from the noise sometimes produced in running, and it should be said that this table is submitted for criticism rather than for general adoption. It is one which has given good results for a number of years in machine design, and its faults, such as they may be, are believed to be in the right direction from another point of view; for when the durability of a train of gearing is considered, it would seem that all gears in the train should have the same pitch and face, because all transmit the same power and are therefore subject to the same wear. But this argument is modified by the further consideration that equal wear does not mean equal life in a train of gearing, and a compromise between the considerations of life and strength must result in the adoption of different values for different speeds, somewhat similar to those given in Table 2.

To illustrate the use of Tables 1 and 2, let it be required to find the working strength of a 12-toothed pinion of 1" pitch, 2 1/2" face, driving a wheel of 60 teeth at 100 feet or less per minute, and let the teeth be of the 20° involute form. In the formula $W = spfy$, we have for a cast-iron pinion,

$$s = 8{,}000,\ pf = 2.5, \text{ and } y = .078,$$

and multiplying these values together we have $W = 1{,}560$ pounds. For the wheel we have $y = .134$, and $W = 2{,}680$ pounds.

The cast-iron pinion is, therefore, the measure of strength, but if a steel pinion be substituted, we have $s = 20{,}000$ and $W = 3{,}900$ pounds, in which combination the wheel is the weaker, and it therefore becomes the measure of strength. In teeth of the involute and cycloidal forms, there is a marked difference between racks and pinions in working strength, while in radial flanked teeth, which are used more especially on bevel gears, the difference is not so pronounced. These teeth are to be found in the great majority of all cut bevels, because they can be more cheaply produced on milling machines and gear cutters, but the 15° involute bevel tooth, as made by the Bilgram process, is superior in accuracy of form and finish, and is often preferred for patterns and fine machinery. There are, therefore, two well-defined forms of bevel gears to be considered: and to bring the strength of bevel gearing within the scope of the present investigation, it will be necessary to understand how the variation in their pitch and radius of action is allowed for, and without going into a demonstration of the formula, its simple statement will probably be sufficient.

Referring to Plate II,

D = large diameter of bevel.

d = small diameter of bevel.

p = pitch at large diameter.

n = actual number of teeth.

N = formative number of teeth = n secant a, or the number corresponding to radius R.

f = face of bevel

y = factor depending upon shape of teeth and formative number N.

W = working load on teeth referred to in Diagram D.

Then it can be shown that

$$W = s\,p\,f\,y\,\frac{D^3 - d^3}{3\,D^2\,(D - d)} \qquad (5)$$

To illustrate the use of this formula, let it be required to find the working strength of a pair of cast-iron miter gears of 50 teeth, 2" inch pitch, 5" face, at 120 revolutions per minute.

In this case, $a = 45°$, $D = 31.8$, $d = 24.8$, and secant $a = 1.4$, $N = 50 \times 1.4 = 70$, for which $y = .071$. The speed of the teeth is 1,000 feet per minute, for which, by interpolation in the table, $s = 2{,}800$, and the formula becomes, by substituting these values.

$$W = 2{,}800 \times 2 \times 5 \times .071 \times \frac{31.8^3 - 24.8^3}{3 \times 31.8^2\,(31.8 - 24.8)}$$
$$= 1{,}988 \times .795 = 1{,}580 \text{ lbs.}$$

This result is attained by some labor which is practically unnecessary, because d should never be made less than *2/3 D*, and when this rule is observed, the approximate formula,

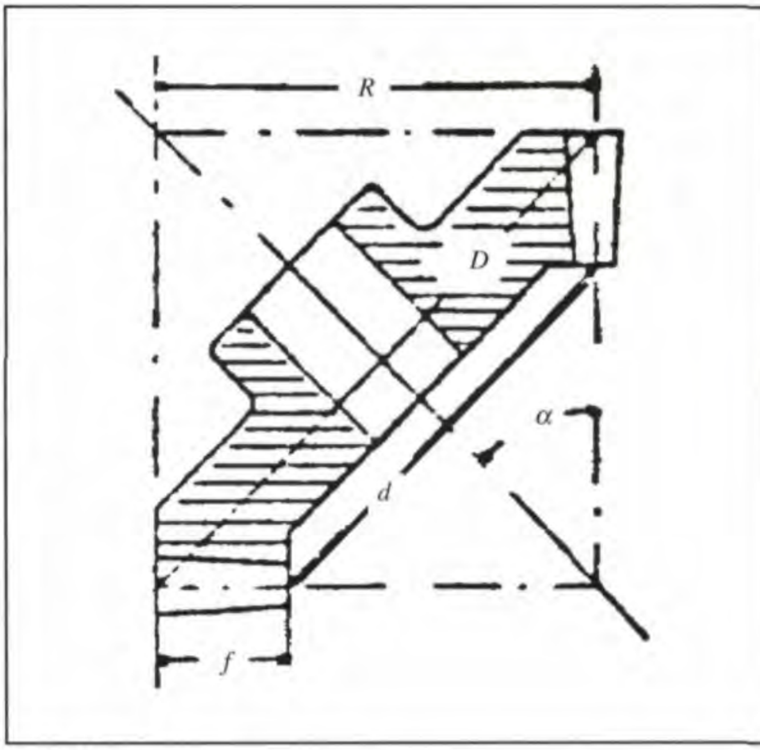

Plate II

$$W = s\,p\,f\,y\,\frac{d}{D} \qquad (6)$$

gives almost identical results. The reason for fixing a limit to the ratio $\frac{d}{D}$ is found in the fact that a further increase in face adds very slowly to the strength and increases very rapidly the difficulty of properly distributing the pressure transmitted. Where long-faced bevels are used, the teeth near the shaft are generally broken by improper fitting, or, when properly fitted, by the spring of the shaft or the yielding of its bearings, and, as the limit imposed gives about 70% of the strength attainable, by extending the face to the center, it is thought to be as liberal as experience can justify.

In presenting certain forms and proportions for teeth, on which Table 1 is founded, I am aware that other forms and proportions are in common use, which have some claims to recognition, but my chief object at present is to show that the strength of gearing can be reduced to a rational basis of comparison on which all authorities may ultimately unite. ■

Tooth Contact Shift in Loaded Spiral Bevel Gears

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Nomenclature

a Tooth mid-cone plane addendum, m
A Distance from gear to first bearing, m
A_o Outer-cone distance, m
B Distance from gear to second bearing, m
D Tooth contact shift, m
D_o Mid-cone distance, m
EI Elastic support stiffness, N-m^2
EI_o Elastic support stiffness of base design, N-m^2
f Tooth face width, m
N Number of teeth
O Center of curvature
p_b Base pitch equivalent spur gear, m
r Mid-plane pitch radius, m
R Mid-cone pitch radius, m
R_e Effective pitch radius, m
R'_e Loaded effective pitch radius
T Torque, N-m
U Pitch point tangential motion, m
W Force component, N
X Bearing radial deflection, m
Y Pitch point deflection, m
Z Gear center deflection, m
β Orthogonal coordinate frame
Γ Cone angle, rad
θ Pitch point slope, rad
ρ Radius of curvature, m
Σ Shaft angle, rad
ϕ_n Normal pressure angle, rad
ϕ'_n Loaded normal pressure angle, rad
ψ Spiral angle, rad

Subscripts

a Axial first bearing
b Second bearing
bi, i = 1, 2, 3 motion from bearing in first coordinate frame direction
ci, i = 1, 2, 3 motion from shaft in first coordinate frame direction
di, i = 1, 2, 3 first coordinate frame direction
ei, i = 1, 2, 3 second coordinate frame direction
fi, i = 1, 2, 3 third coordinate frame direction
gi, i = 1, 2, 3 fourth coordinate frame direction
g Gear (as last subscript)
p Pinion (as last subscript)
r Radial
t Tangential

Abstract: An analytical method is presented to predict the shifts of the contact ellipses on spiral bevel gear teeth under load. The contact ellipse shift is the motion of the tooth contact position from the ideal pitch point to its location under load. The shifts are due to the elastic motions of the gear and pinion supporting shafts and bearings. The calculations include the elastic deflections of the gear shafts and the deflections of the four shaft bearings. The method assumes that the surface curvature of each tooth is constant near the unloaded pitch point. Results from these calculations will help designers reduce transmission weight without seriously reducing transmission performance.

Introduction

Spiral bevel gears are important elements for transmitting power. In designing spiral bevel gear transmissions, the designer meets a tradeoff between a transmission's weight and its life and reliability. By removing weight from transmission components, one increases the overall flexibility of the transmission. This flexibility affects the deflections of the loaded components in the transmission.

The design of an efficient spiral bevel gear box includes the design of gear and support geometry, bearing and shaft sizes, and material properties. The gear tooth interaction is more complex than that of spur or helical gears. The loaded region of the gear mesh shifts due to the deflections of the gear and the pinion. A primary cause of these motions is the flexibility of the gear support shaft and bearings. Although the teeth also deflect, tooth beam deflections are small in comparison to tooth gear support deflections. If the shift of the loaded region is large, the life of the gear mesh may reduce significantly.

The classic work of Wildhaber[1-2] describes the kinematic operation and the generation of spiral bevel gears. More recently, Baxter[3] and Coleman[4] expanded on this fundamental theory. They described a tooth contact analysis program which analyzes the kinematic action of two spiral bevel gears in mesh. The program considers effects of tooth manufacturing parameters on the gear mesh kinematics.

Litvin and Coy[5] and Litvin, Rahamn and Goldrich[6] also presented the theory of spiral bevel gear generation and design. These works describe kinematic errors induced in the transmission by errors in gear manufacturing and assembly. The articles also suggest tooth profile modifications to improve kinematic transmission. They give direct relationships for the generated curvatures and directions on the terms of the principal curvatures and directions on the bevel gear teeth surfaces. These relationships are in terms of the principal curvatures and directions of the tool generating surfaces.

Coleman[7] described the experimental measurement of existing bevel and hypoid gear deflections under load. The article cites the importance of these deflections for the capacity and noise level of the gear set. The test results also suggest gear mounting and tooth manufacturing changes which can improve the capacity and life of the gear set.

Taha, Ettles, and MacPherson[8] presented the interaction of the structural rigidity and performance from a helicopter tail rotor gear box. They used a finite element model for the housing. The article presents effects of deflections on bearing roller load distribution, bearing fatigue life, and the gear motions at the unloaded pitch point. Their work demonstrates the importance of rigidity to minimize deflections and maximize bearing life in a transmission.

Winter and Paul[9] presented work on the influence of spiral bevel gear deflection on tooth root stresses.

This article presents an analytical method to predict the shift of the loaded region on the tooth. The method treats the shift as a result of the elastic deflections of the gear support shafting and bearings. The analysis is sequential.

The first stage defines the gear geometry and loading. The second stage determines the elastic

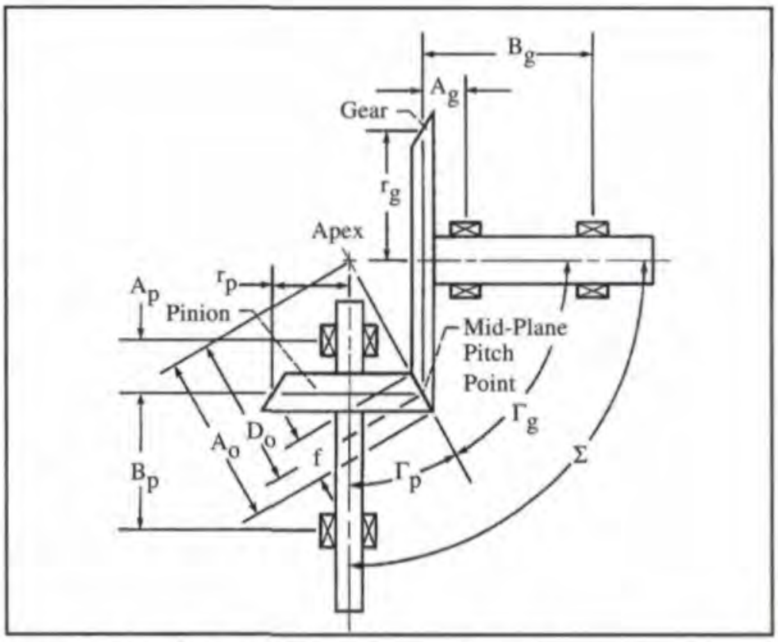

Fig. 1 - Spiral bevel gear mesh and support bearing geometry.

deflections of the bearings and shafts and the slopes of the shafts at the gears. The third stage finds the motions of the gear teeth caused by each elastic deflection. The total deflections of the gear teeth are the algebraic sum of these motions. The fourth stage determines the geometry and curvatures of the gear and pinion teeth. These directions and curvatures combine with the separate motions of the gear teeth to produce the contact shift. The contact shift is the motion of the contact ellipses on the two tooth surfaces under load from the unloaded pitch point. The analysis assumes that the curvatures are constant over the affected surfaces of the teeth.

A gear box model similar to the main rotor bevel gear reduction of the U.S. Army OH-58 light duty helicopter serves as an example for the method. The gear box includes a single spiral bevel gear drive with a pinion input and the supporting shafts and bearings. The analysis includes a parametric study. The article presents radial and tangential shifts of the contact ellipses on the pinion and gear teeth as functions of shaft stiffness.

Geometry and Loading

Fig. 1 shows the geometry of a spiral bevel gear mesh. The mid-cone distance, D_o, is the distance from the apex of the bevel cones to the mid-place pitch point. This distance is along the pitch line of the two pitch cones. It is equal to the outer-cone distance, A_o, minus one-half the tooth face width, f. The mid-cone distance and the tooth face width describe the basic gear blank sizes. The cone pitch angles, Γ_g and Γ_p, for the

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gear and pinion also contribute to the gear sizes. The shaft angle, Σ, is the sum of the cone pitch angles. The shaft angle defines the relative orientation of the gear and pinion shafts. The pitch radius of the gear in its mid-line, r_g is:

$$r_g = D_o * \sin \Gamma_g \qquad (1)$$

The cone pitch angle, Γ_g, in terms of the numbers of teeth on the gear and pinion, N_g and N_p, and the shaft angle, Σ, is:

$$\tan \Gamma_g = \frac{\sin \Sigma}{(N_p/N_g) + \cos \Sigma} \qquad (2)$$

Similar equations define the pitch radius, r_p, and cone pitch angle, Γ_p, of the pinion with the subscripts p and g interchanged.

Fig. 2 shows the spiral angle, ψ, and normal pressure angle, ϕ_n. The spiral angle, ψ, is the angle of inclination of the teeth to the pitch ray. It is in the place tangent to the two pitch cones. The gear of Fig. 2 has a right hand spiral. The normal pressure angle, ϕ_n, is the angle between a normal to the tooth and the tangent to the pitch cone surface in the tooth normal plane.

Fig. 2 includes four orthogonal right handed coordinate frames. All four coordinate frames are at the mid-plane pitch point of the spiral bevel tooth. The first of these, β_{di}, has coordinates in the tangential, axial, and radial directions of the gear body. The second coordinate frame, β_{ei}, is the first coordinate frame rotated through the cone angle, Γ_g, in a negative direction about the β_{d1} vector. The unit vectors of this frame are in the tangential, pitch ray, and mid-cone radial directions, respectively. The third coordinate frame, β_{fi}, is the second coordinate frame rotated through the spiral angle, ψ, in a positive direction about the β_{e3} vector fora right handed spiral angle. The unit vectors of this frame are tangent to the pitch cone in the tooth normal place, tangent to the tooth in the cone tangent plane, and in the mid-cone radial direction respectively. The fourth coordinate frame, β_{gi}, is the third coordinate frame rotated through the normal pressure angle, ϕ_n, in a negative direction about the β_{f2} vector. The unit vectors of this frame are in the tooth normal direction, tangent to the tooth in the cone tangent plane, and tangent to the tooth in the tooth normal plane respectively.

Fig. 2 shows the mid-plane pitch radius of the gear, r_g, in the axial section plane. The mid-cone radius of the gear, R_g, is in the same view.

This equivalent spur pitch radius is:

$$R_g = \frac{r_g}{\cos \Gamma_g} \qquad (3)$$

The gear assembly includes the support bearings and their locations. The position of the bearings relative to the gear or pinion describes the support system geometry. The two most common bearing configurations are straddle and overhung mountings. Fig. 1 shows both. The pinion has a straddle mounting, while the gear has an overhung mounting. In both cases, A is the distance from the gear mid-plane to the bearing closest to the pitch cone apex. B is the distance from the gear mid-plane to the bearing furthest from the apex. Distance A is positive for an overhung mounting and negative for a straddle mounting. B is always positive.

The normal force acts on the tooth at the mid-plane pitch point. The method assumes that the force is a concentrated force. The force has three orthogonal components in the directions of the β_{di} coordinate frame relative to the tooth. The tangential component, W_t, produces the torque on the gear. It acts in the β_{d1} direction. The axial component, W_a, acts in the β_{d2} direction. And the radial component, W_r, acts in the β_{d3} direction.

Fig. 3 shows these forces for both the gear and the pinion with their respective coordinate frames. For the gear, the components are:

$$W_t = \frac{T_g}{D_o * \sin \Gamma_g} \qquad (4)$$

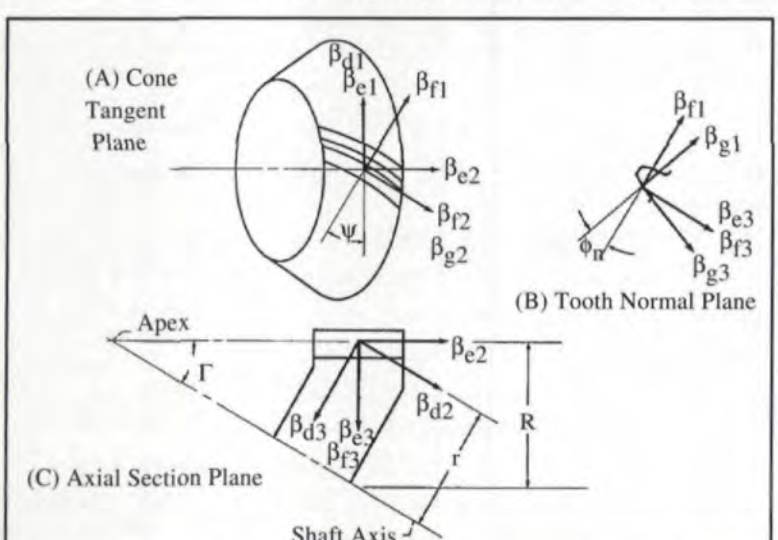

Fig. 2 - Spiral bevel gear tooth coordinate frames.

$$W_a = \frac{|W_t|}{\cos\psi}\,[\,\tan\phi_n * \sin\Gamma_g + \sin\psi * \cos\Gamma_g\,] \quad (5)$$

$$W_r = \frac{|W_t|}{\cos\psi}\,[\,\tan\phi_n * \cos\Gamma_g - \sin\psi * \sin\Gamma_g\,] \quad (6)$$

where T_g is the torque on the gear. Replacing the subscript g with the subscript p in Equations 4 through 6 gives the equations for the pinion force components. In Equations 5 and 6, the sign of the last term depends on the direction of rotation, the hand of the spiral, and whether the gear is driving or being driven. These equations are valid for a right-handed spiral driving gear rotating counterclockwise about the β_{d2} direction. Each change in the spiral hand, power direction, or rotation direction reverses the signs.

The sign of the tangential load, W_t, also depends on the directions of the rotation and power flow. The tangential component is positive for a driving gear or pinion which is rotting counterclockwise. It is also positive for a driven gear of pinion which is rotating clockwise. The tangential load is negative otherwise. However, Equations 5 and 6 use the absolute value of W_t.

Component Deflections

Under load, the motion of the ideal pitch point of a spiral bevel gear is mainly the superposition of three elastic deflections. These deflections result from shaft deflections at the gear centers, shaft slopes at the gear centers, and the deflections of the support bearings.

Table I lists the results of a strength of materials shaft analysis for the deflections, Y_{ci}, and slopes (rotations), ϕ_{ci}, of the gear at the pitch point in and about the β_{di} directions. In the analysis, both the axial and radial forces contribute to both the radial deflection and the tangential rotation. The axial deflection is due to the tangential rotation and the radial location of the pitch point. In this instance, the straddle and overhung cases require different formulae due to the different elastic configurations of the two cases. As before, interchanging gear and pinion subscripts yields valid equations for the pinion.

The bearings which support the shaft also deflect. Each bearing has a nonlinear stiffness and its own contact angle. Harris[10] and Houghton[11] present methods for determining the rolling element load sharing and resulting deflection. The method of this article combines the radial and axial loads on each bear-

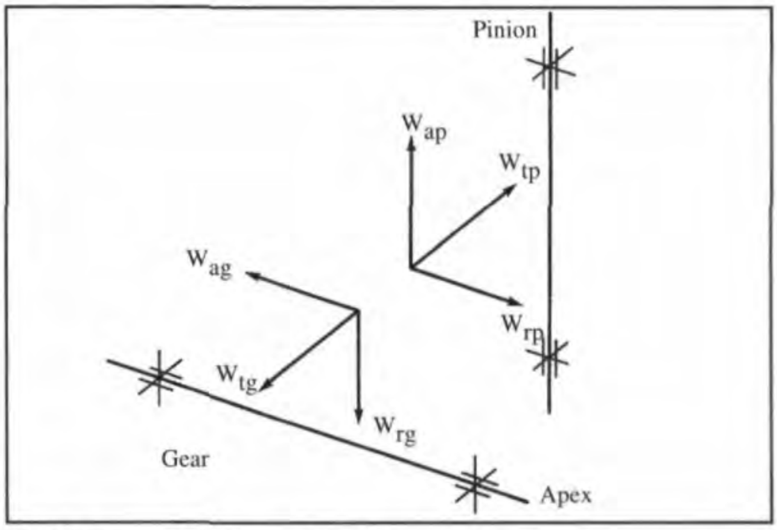

Fig. 3 - Gear and pinion tooth force components.

DEFLECTIONS	
Straddle	Overhung
$Y_{c1} = \frac{W_t A^2 B^2}{3\,EI\,(B-A)}$	$\frac{W_t A^2 B}{3\,EI}$
$Y_{c2} = r * \theta_{c1}$	$r * \theta_{c1}$
$Y_{c3} = \frac{-W_a\, rAB(B+A)}{3\,EI\,(B-A)} + \frac{W_r A^2 B^2}{3\,EI\,(B-A)}$	$\frac{-W_a\, r\,A\,(2B+A)}{6\,EI} + \frac{W_r A^2 B}{3\,EI}$
SLOPES	
$\theta_{c1} = \frac{W_a\, r\,(A^2+AB+B^2)}{3\,EI\,(B-A)} - \frac{W_r\, AB\,(B+A)}{3\,EI\,(B-A)}$	$\frac{W_a\, r\,(2A+B)}{3\,EI} - \frac{W_r\, A(2B+A)}{6\,EI}$
$\theta_{c2} = 0$	0
$\theta_{c3} = \frac{W_t\, AB(B+A)}{3\,EI\,(B-A)}$	$\frac{W_t\, A(2B+A)}{6\,EI}$

Table 1 - Elastic Support Induced Pitch Point Motion

DEFLECTIONS
$y_{b1} = \frac{B * X_{a1} - A * X_{b1}}{B - A}$
$y_{b2} = X_2 + r * \theta_{b1}$
$y_{b3} = \frac{B * X_{a3} - A * X_{b3}}{B - A}$
SLOPES
$\theta_{b1} = \frac{X_{b3} - X_{a3}}{B - A}$
$\theta_{b2} = 0$
$\theta_{b3} = \frac{X_{a1} - X_{b1}}{B - A}$

Table 2 - Bearing Induced Pitch Point Motion

ing to find the bearing deflections. The program then resolves each bearing deflection into its radial and axial components in the β_{di} coordinate directions.

The deflections at the bearings are X_{ai} and X_{bi}. Here the subscript a identifies the bearing located the distance *A* from the gear. And the subscript *b* identifies the bearing located the distance *B* away. The last subscript, *i*, denotes the deflection direction in the β_{di}coordinate frame.

For $i = 2$, both bearings have the same axial deflection, X_2. Table II lists the deflections, Y_{bi}, and the rotations, β_{di} of the gear pitch point caused by these bearing deflections. The motions are due to the rigid body motion of the gear and support shaft in the bearings.

The total motions of the gear pitch point in and about the θ_{di} coordinate directions, neglecting any elastic motion of the transmission housing are:

$$Y_{di} = Y_{bi} + Y_{ci} \ for \ i = 1, 2, 3 \qquad (7)$$

$$\theta_{di} = \theta_{bi} + \theta_{ci} \ for \ i = 1, 2, 3 \qquad (8)$$

Similar equations describe the motion of the pinion pitch point. The motions of the gear and pinion pitch points combine in the analysis for the shift of the contact ellipse under load.

Contact Analysis

The principal curvatures of the teeth are needed in addition to the teeth deflections in the contact ellipse shift model. Unfortunately, no direct equations exist for the spiral bevel tooth surface due to the complexity of the surface geometry[5-6]. However, an indirect approach can determine the principal curvatures and directions of curvature. Litvin[5-6] has developed equations to determine these principal curvatures and their directions at the pitch point.

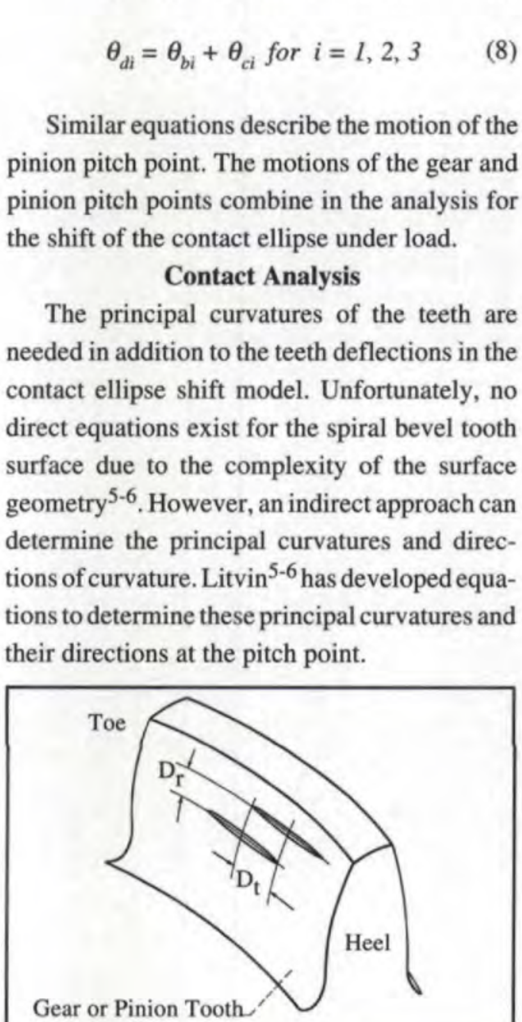

Fig. 4 - Tooth contact point motion.

The analysis assumes that direct conjugate relationships exist between the gear cutting tool surface curvatures and motion and the sought quantities. These are the principal curvatures and their directions on the gear tooth surface. The analysis uses the gear and pinion geometry along with the cutter machine settings for both the gear and pinion. The cutter machine's manufacturer provides the gear cutter settings. With this method, one can determine the curvatures of the gear tooth surface without equations for the surface.

The separate pitch point motions of the two gear teeth effect the shift of the contact ellipse under load. In addition, the motions of the centers of transverse curvature of the teeth surfaces effect the tangential contact ellipse shift. The motions of the mid-plane base circles of the teeth also effect the radial contact ellipse shift. In addition, the standard Hertzian contact stress formulae[12] yield the size and orientation of the contact ellipse on the gears. All these calculations assume that the principal curvatures remain constant over the contacted portions of the spiral bevel gear teeth.

Relative Motions

The contact shift motion occurs in the plane tangent to the two teeth. Relative motions in the direction of the common normal to the teeth produce kinematic errors in the motion transmission. Kinematic error[5] is a forward or backward rotation of the output gear from its ideal position relative to the input gear. This motion does not produce a shift of the contact ellipse. Only the radial shift on the teeth in the tooth normal plane and the tangential relative motion in the cone tangent plane are of interest. The radial shift of the teeth produces a radial shift in the contact ellipse locations on both teeth. The tangential motion produces a tangential shift in the contact ellipse locations as shown in Fig. 4.

In the motion analysis, the small pitch point deflections and rotations are vectors. One can obtain the vector components in any of the four coordinate frames by direct matrix rotations.

Radial Shift - The radial shift of the two teeth appears in the mid-cone plane and in the tooth normal plane. Fig. 5 shows two equivalent spur gears in the tooth normal plane before and after application of the separating load. Equivalent spur gears have the tooth geometry of the spiral bevel tooth in its mid-cone plane. The involute action of these spur gears is the same as that of spiral bevels in the mid-cone plane.[1-3] The radial shift of the two gears is due to an increase of the center distance in this rotated mid-cone plane. The separating motion of each gear center is the pitch point motion in the β_{e3} direction minus a small relative motion. The relative radial motion of the gear center is toward the pitch point. It is due to the gear rotation about the original tangential direction β_{e1}.

The displacement of the pitch point in the β_{e3} direction is:

$$Y_{e3,g} = Y_{d2,g} * \sin \Gamma_g + Y_{d3,g} * \cos \Gamma_g \quad (9)$$

The effective radius of the equivalent spur gear in the tooth normal plane, R_{eg}, is a function of the mid-cone radius, R_g, and the spiral angle, ψ:

$$R_{eg} = \frac{R_g}{\cos^2 \Psi} \quad (10)$$

The motion of the center of the equivalent gear is thus:

$$Z_{e3,g} = Y_{e3,g} - R_{eg}(1.0 - \cos \theta_{d1,g}) \quad (11)$$

Due to the involute action on separation, the radial motion produces a new loaded pitch point. The normal pressure angle increases slightly. The gears rotate slightly, and a new loaded effective pitch radius results. The new pressure angle, θ'_n, is:

$$\phi'_n = \cos^{-1}\left[\frac{(R_{eg} + R_{ep}) * \cos \phi_n}{R_{eg} + Z_{e3,g} + R_{ep} + Z_{e3,p}}\right] \quad (12)$$

The loaded effective gear pitch radius, R'_{eg}, is:

$$R'_{eg} = \frac{R_{eg} * \cos \phi_n}{\cos \phi'_n} \quad (13)$$

The radial shift of the pitch point from the unloaded to the loaded position on the gear tooth, D'_{rg}, is now:

$$D_{rg} = \frac{R'_{eg} - R_{eg}}{\cos \phi_n} \quad (14)$$

Fig. 6 shows this shift distance and the two pitch radii. Fig. 6 is a normal view of the spiral bevel gear tooth. The analysis for the radial shift of the contact point on the pinion tooth is similar.

Tangential Shift - The relative tangential motion of the two teeth appears clearly in a plane normal to the tooth. This plane contains the tooth tangent vector β_{g2}. The plane is perpendicular to vector β_{g3}. It contains the tangential motion of the pitch points, and it contains the centers of transverse curvature of the gear and pinion teeth, O_g and O_p respectively. The radii of transverse curvature of the teeth, ρ_g and ρ_p, are primary curvatures which the tooth contact analysis determines.

Fig. 7 shows the line of centers in this plane for both the unloaded and the loaded condition. The centers of curvature, O_g and O_p, have a prime in their loaded positions. For clarity, the two centers of curvature are on opposite sides of the tooth surface tangent in Fig. 7. In reality, both centers are on the same side of the tooth surface tangent. However, the tangent shift relation is the same for both cases. The deflections of these centers of curvature in this plane are related to their respective pitch point deflections in the β_{g3} direction. The relation adds the motion of the curvature center relative to the original pitch

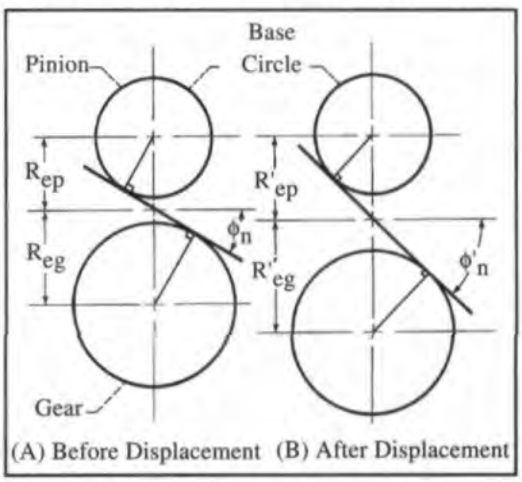

Fig. 5 - Equivalent spur gear separation in tooth normal plane.

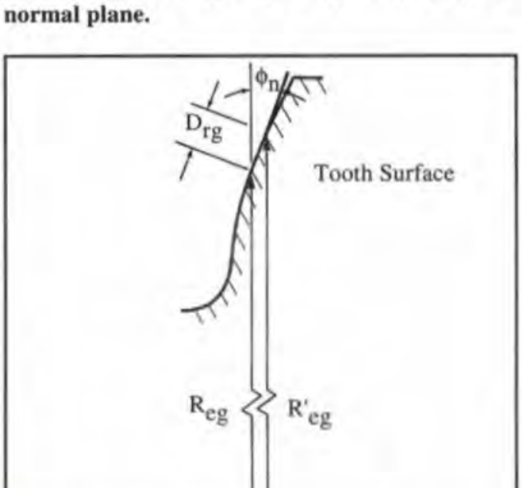

Fig. 6 - Radial motion of gear pitch point.

point. The relative motion is due to the gear body rotation perpendicular to this plane about the β_{g3} direction. For the gear, this deflection is:

$$Z_{g2,g} = Y_{g2,g} + \rho_g * \sin\theta_{g3,g} \qquad (15)$$

where

$$Y_{g2,g} = -Y_{d1,g} * \sin\psi + Y_{d2,g} * \cos\Gamma_g \cos\psi \qquad (16)$$
$$-Y_{d3,g} * \sin\Gamma_g \cos\psi$$

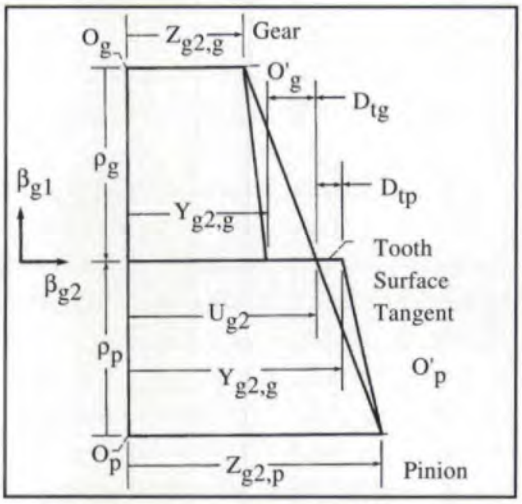

Fig. 7 - Tangential motion of spiral bevel pitch point.

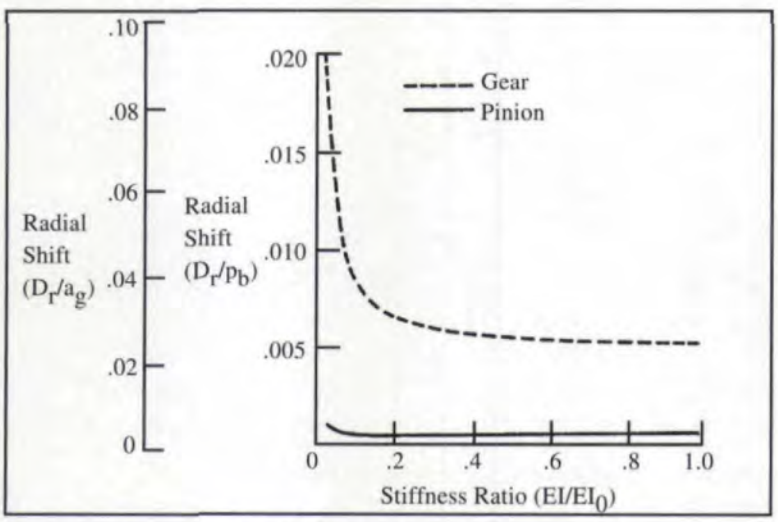

Fig. 8 - Tooth contact position shift in radial direction.

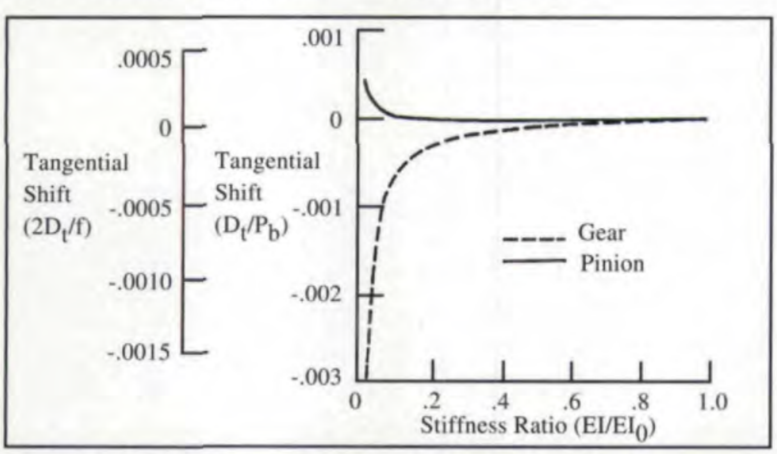

Fig. 9 - Tooth contact position shift in tangential direction.

and:

$$\theta_{g3,g} = \theta_{d1,g} * \cos\psi \sin\phi_n \qquad (17)$$
$$+ \theta_{d2,g} * (\cos\Gamma_g \sin\psi \sin\phi_n - \sin\Gamma_g \cos\phi_n)$$
$$- \theta_{d3,g} * (\sin\Gamma_g \sin\psi \sin\phi_n + \cos\Gamma_g \cos\phi_n)$$

The motion on the tooth surface from the unloaded pitch point to the loaded pitch point in this plane is U_{g2}. This is the deflection necessary to keep the centers of curvature on a common tooth normal.

$$U_{g2} = Z_{g2,g} + \frac{\rho_g}{\rho_g + \rho_p}(Z_{g2,p} - Z_{g2,g}) \qquad (18)$$

This locates the new loaded pitch point relative to the unloaded pitch point. The relative shift of the gear pitch point in the tangential direction on the tooth surface is D_{tg}. This is the difference between the original pitch point's motion and the pitch point shift, U_{g2}:

$$D_{tg} = U_{g2} - Y_{g2,g} \qquad (19)$$

A similar analysis determines the tangential pitch point shift on the pinion, D_{tp}.

Tooth Contact Shift

The radial and tangential components of tooth contact shift are valuable measures of the rigidity of a bevel gear mesh. Knowledge of the shift components can help a designer evaluate important properties of the gear mesh and support bearings. An interactive input Fortran 77 computer program[13] calculates these tooth contact shifts for both the gear and pinion. The program, SLIDE.FOR, can run on a personal computer.

Fig. 4 shows the shift of the nominal pitch point contact ellipse on a gear tooth. The principal radii of curvature of the teeth and the normal tooth load determine the size and orientation of the contact ellipse. One can compare the tangential shift to the difference between the half tooth width and the tangential radius of the ellipse. This comparison demonstrates whether edge loading can occur in the design. By subtracting the radial shifts from the teeth addenda, one also can evaluate the reduction in the ideal contact ratio.

This model can evaluate performance tradeoffs where transmission weight and transmission life are competing objectives. Figs. 8 and 9 are dimensionless plots of contact shift as function of support shaft stiffness for a single stage transmission. The plots vary both the gear and pinion shaft stiffnesses by the same percentage. The parametric study reduces the stiffnesses from the design values to show the effect of shaft stiffness on performance.

Fig. 8 is a plot of the radial contact shift on both the gear and the pinion. Fig. 9 is a plot of the tangential contact shift for the two gears. Both plots show the contact shift as a ratio to the mid-cone base pitch of the teeth. This gives a dimensionless comparison of the shift magnitudes. A second vertical axis on the radial plot shows the shift as a percent of the tooth addendum. This compares the shift of the contact point to the actual tooth size. In the tangential shift plot, the second vertical axis shows the tangential shift as a percent of the half tooth width. The horizontal axis of both plots is the ratio of the support shaft stiffness, EI, to the full support shaft stiffness, EI_o.

The figures demonstrate a definite hyperbolic relationship between the stiffnesses and the resulting pitch point contact shift. A valuable conclusion is that one may reduce the stiffnesses in the example to about 20% of the nominal design values. At this point, there is a significant increase in the contact position shift. With this information available at the design stage of a transmission's development, important weight savings may be possible. The weight savings will not sacrifice the transmission's life or power overload performance seriously.

Summary and Conclusion

This article presents a method to predict the shift of the loaded region on the gear teeth in a spiral bevel reduction. This shift is a result of the elastic deflections of the gear support shafting and bearings under load. The reduction is a single spiral bevel gear drive with a pinion input and supporting shafts and bearings. The assumed deflections were caused by the shaft deflections, the shaft slopes, and the bearing deflections. The analysis assumed that the curvatures of the teeth near the pitch point were constant. It determined the curvatures using the envelope of cutting tool positions.

The analysis was sequential. The first stage defined the gear geometry and loading. The second stage determined the elastic deflections of the bearings and shafts and the slopes of the shafts at the gears. The third stage determined the motions of the gear teeth caused by each elastic deflection. Superposition of these motions produced the total deflections of the gear teeth. The fourth stage analyzed the interaction between the tooth geometry and motion to predict the tooth contact ellipse motions under load. A Fortran 77 computer program determined the radial and tangential tooth contact shifts of the gear and pinion.

An example representative of the main rotor bevel gear reduction found in the U.S. Army OH-58 helicopter was modelled. A parametric study illustrated the determination of the radial and tangential tooth contact ellipse shifts. Graphs showed the variation in contact shift with reduction in support shaft stiffness. An important conclusion was that, for the example, one could reduce the shaft stiffnesses to about 20% of the present stiffnesses. At this level, significant changes in the shifts of the tooth contact positions occurred. Significant weight reductions may be possible without seriously affecting the gear action or the gear life. ■

Acknowledgements: *Originally published as NASA Technical Memorandum 101438 and in the Proceedings of the ASME Fifth International Power Transmission and Gearing Conference 1989.*

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GEAR FUNDAMENTALS

Classification of Types of Gear Tooth Wear - Part I.

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Introduction

The phenomena of deterioration of surfaces are generally very complex and depend on numerous conditions which include the operating conditions, the type of load applied, the relative speeds of surfaces in contact, the temperature, lubrication, surface hardness and roughness, and the compatibility and nature of materials.

Wear is a general term covering the local phenomena describing the removal of some material and occurring when two surfaces slide onto one another. This term also applies to the removal of material resulting from the presence of impurities in the lubricant. Other types of gear failures, such as surface fatigue, corrosion, plastic flow, and breakage are not covered by this article.

Wear may be divided into two distinct classifications: Qualitative, which is based on the action modes of different wear phenomena on gear teeth and characterized by surface appearance; and quantitative, which takes into account the intensity of the wear phenomena.

The qualitative wear phenomena we will discuss include abrasive wear with two bodies, streaks and scoring, polishing, hot and cold scuffing, abrasive wear with three bodies, scratches or grooves, and interference wear. In the quantitative analysis we will define normal wear, moderate wear, and excessive wear. A synthesis of these two classifications has been made under a chart found in Part II.

Current Wear (Called Two Body Abrasion)

Current wear is revealed very early in the life of the gear or gear train and is evidenced by the removal of micro-particles of metal on the gear teeth surfaces. This phenomenon is caused by contact and metal-to-metal sliding, which occurs through the oil film. The distinctive machining marks of the cutting procedure or finish (for example, the facets resulting from hobbing, the streaks left by a gear cutter or by a rack cutter, the surface pattern from certain grinding operations) are diminished or erased by wear.

This wear brings about a progressive reduction of gear tooth thickness, along with a more or less marked distortion of the profile in the heavy sliding zones, but without noticeable degradation of the surface roughness.

As indicated in Fig. 1, the wear is almost nil

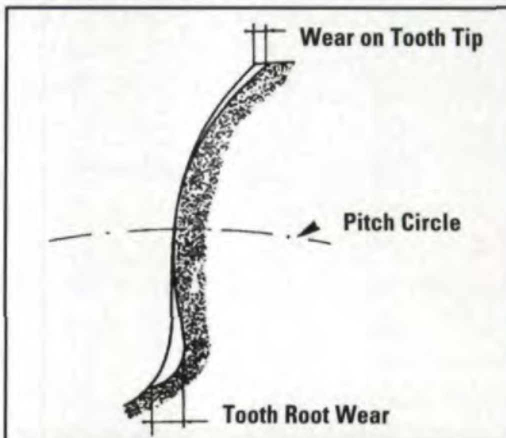

Fig. 1 - Distinctive aspect of distortion of gear tooth profile under wear action


Fig. 2 - Traces of wear recorded on a gear wheel in operation.

in the pitch zone where the sliding speeds are low or nil, and becomes more and more pronounced as we move away from this zone. This zone is maximum at the tip circle and at the active dedendum circle, where the sliding speeds are the highest.

The wear zone is generally gray colored and slightly dull, with sometimes lustrous areas and the presence of scoring. (See Fig. 2). Note the presence of a dull zone below the pitch, where the wear is greater, and of a lustrous area on each side of the pitch and slightly higher. Also note the transverse scoring slightly marked near the tooth tip and the machine streaks still visible near the pitch.

The appearance of this wear, as well as its developing speed, varies greatly according to the pressure level between the contact surfaces, the hardness of the materials, the roughness of surface, and the thickness of lubricant film. For example, in the case of slightly loaded gears operating with an oil of relatively high viscosity at medium speeds, we will have an oil film sufficiently thick to avoid metal-to-metal contact. This will not generate wear except during starting or stopping. The original machining traces will still be intact on the gear teeth after long periods of operation.

In practice it is not always possible to have a continuous lubricant film between the gear teeth according to the load transmitted. Then contact occurs between the top of the asperities made during machining, and there is a tendency to polish or score the surface in contact.

In the case of surface-hardened gears, because of the high hardness of flanks in contact, we encounter only slight wear which is often difficult to see.

Scoring - Streaks

This type of wear appears in the form of fine grooves or lines in the sliding direction. These streaks or scores are formed progressively in the zone characterized by a high sliding speed, at the tooth tip and near the root of the gear teeth. (Fig. 3)

They are often developed on pinion teeth and wheel teeth that are in contact at the very beginning of meshing of mating profiles. The bottom of the striae are smooth compared to those found in traces of scuffing.

Causes. The presence of these scores or streaks reveals the existence of relatively high pressures locally affecting the gear teeth flanks. Under high load action, the asperities caused by the roughness of the mating flanks, as well as foreign particles of small dimensions that imprint into the gear teeth flanks, along with the slippage effect, cause cavities which appear as streaks.

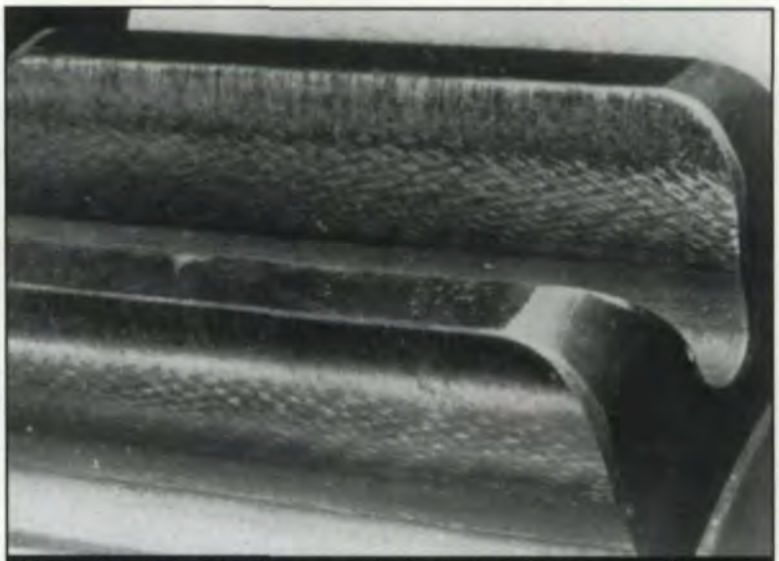

Fig. 3 - Scoring or streaks seen at the beginning of a gear operation.

The formation of scoring can be considered as a preliminary stage preceding scuffing. These streaks generally lead to more severe local wear of gear teeth in the zones where there is a higher pressure. Frequently we note in time a stabilization of these streaks when the wear level of the scored surfaces has been sufficient to generate a better distribution of the load, which as a consequence diminishes the maximum pressure on the gear teeth. In this case only the black color base of streaks or scoring will remain. This stabilization phenomenon can be accelerated by increasing the oil viscosity and by improving filtration. This type of wear is often encountered in worm gears. (Fig. 4)

Adhesive Wear

Adhesive wear appears on two surfaces sliding across one another when the pressure between the asperities in contact is sufficient to cause localized plastic deformations, micro-weldings, or local adhesions. When there is generation or plastic deformation, there is energy absorption which leads to overheating due to friction.

The mildest form of adhesive wear is the formation of "polishing" on the active flanks of the gear teeth. When the load condition and friction become more intense, and when the

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temperature at contact level increases, we may witness the appearance of localized metal adhesions on the teeth, hot scuffing, or cold scuffing (for low speeds and heavy loads).

Polishing. Polishing is a type of very slow progressive wear in which the asperities of the mating flanks are distorted and laminated and then appear on the gear teeth as very smooth and polished surfaces, which take on a mirror aspect. Such wear conditions are caused by metal-to-metal contact during operation. Generally the polishing occurs in applications at low speeds (<20 m/sec) when the lubricant oil film in elastohydrodynamic yield rate is not sufficiently thick and is near the limit of the lubricant performance. Generally, this type of wear does not cause large variations in the shape and in the dimensions of gear teeth. However, when examined under a microscope, the structure just below the contact surface reveals the presence of plastic flows in the material under slippage. Sometimes very localized overheating traces near the surface are also encountered. The polishing encourages establishing gear teeth contact patterns in service and allows obtaining a good conformity of surfaces in contact.

Fig. 4 - Scoring by circular streaks seen on a worm gear.

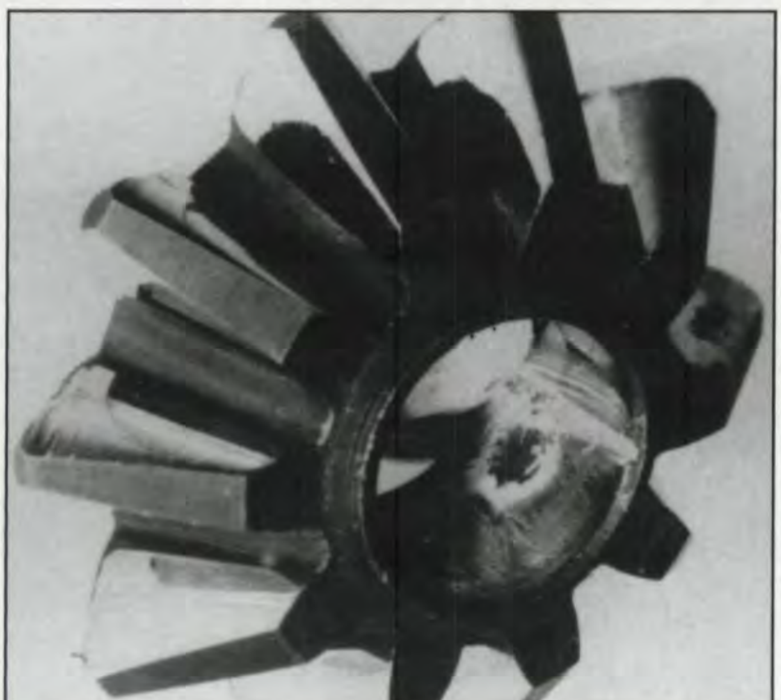

Fig. 5 - Polishing on a gearbox satellite.

This type of wear is not "damage" and can be tolerated in service unless the initial material specifications forbid it. If the load increases, or if the lubrication conditions become insufficient, this type of gear tooth wear can develop because the temperature between surfaces in contact will increase and render them superficially less hard and more sensitive to localized micro-welding formations. The appearance of adhesion or scuffing traces on the gear teeth flanks can then be seen. To prevent such traces, we can, after the appearance of polishing, increase slightly the oil viscosity to obtain in normal working conditions a thicker hydrodynamic film. We can also, when possible, reduce slightly the load to be transmitted, but in all cases we must assure that in service there is no risk of producing severe overloads (no matter how short they may be), which would damage greatly the gear teeth surfaces.

Generally the polishing appears quickly on loaded gears made with good precision and most often on surface-hardened gears. The most noticeable examples are found in automotive gears, such as, satellite and planet gears, gearbox pinions, and spiral bevel gears. (Figs. 5-6)

Adhesions. This phenomenon, which appears on some gear mating flanks, is always very localized and generally is only present on a small area of a few gear teeth. For each metal pull-off or adherence, we can almost always distinguish a zone where a brutal welding has occurred between the profiles in contact and which was immediately sheared. On one of the flanks, a metallic particle has pulled off and is often found fixed by adhesion onto the flank of the other gear. The irregularities on the two surfaces after separation have generated streaks or scratches oriented in the separation direction of the profiles, starting in the zone where the adherence or metal pull-off occurred.

In the welding zone, the profile is generally altered in depth, whereas it is more superficial near the scratches, which generally are less and less severe as we go further from the initially damaged zone. This type of damage is encountered on gear teeth flanks in the high slippage zones, where the contact surface is preponderant or localized because of the presence of a slight excess of material caused by a machining defect or a profile with excessive crowning; by a slight alignment defect of gear teeth caused by the machining operation of the gear of by distortion under load; or by a local distortion along the flanks caused by a temperature gradient generated by an irregular flow of heat during meshing. This contact surface localization causes such an increase in load that it no longer can be supported by the lubricant film, which leads to metal-to-metal contact and the formation of micro-welding by friction.

The development of this type of damage is encouraged by a state of "polishing" on the surface of flanks in contact and by the presence of severe undulations or poor teeth surface roughness. The first case is common in surface-hardened teeth, and the second in rough machine-cut teeth of normalized steel.

The formation of localized metal pull-off can also be caused by the passage of a foreign metallic particle between the profiles. In fact, after its lamination between the gear teeth, such a particle will weld itself onto the surface of one of the profiles under slippage, creating a localized excrescence.

This risks generating an overload during meshing which may cause a sudden rupture of the oil film, thus causing the formation of a metal pull-off on the mating profile. When the numbers of gear teeth on the pinion and on the wheel are in an integer ratio, and such an adhesion appears on one tooth, it may cause similar damage on a limited number of other teeth, and the phenomenon has all the chances of stabilizing by itself afterwards. If the numbers of teeth are prime between themselves, we will have a general sweep of all the teeth by those from the pinion and wheel which have been affected by the initial metal pull-off. The deterioration will thus progress to all the teeth flanks, and there is a risk of evolution towards hot scuffing (at high speeds) or cold scuffing (at low speeds). Quite often this evolution can occur within minutes.

We are often tempted to say that the presence of adhesion on a tooth is not too serious because its surface is small. In reality, this type of deterioration, in the case of surface-hardened teeth, can often go along with cracks that develop from the surface down and which will quickly lead to tooth breakage. Such cracks are formed by residual stress of thermal origin and appear at the moment of instant welding by slippage of the two flanks in contact.

Before planning to re-use surface-hardened gears with or without an eventual grinding surface operation, it is essential to perform a dye penetrant or magnetic particle control of the teeth flanks where the adhesion occurred.

In the case of through-hardened gears, this cracking risk is practically non-existent.

If we ascertain, upon opening a reducer or a gear wheel, the presence of adhesions on the teeth, we can come to the following conclusions:

• There has been periodic lubricant film breakage on the teeth where contact pattern area was preponderant.

• The initial precision of the gear or reducer is generally not affected, for we should have noticed this phenomenon under nominal load during start-up of reducer.

The causes of this anomaly may be:

1) Distortion of the gear support or reducer, bringing about an evolution of the contact sur-


Fig. 6 - Polishing on a spiral bevel gear.

face towards the tips of teeth (in cases of adherences near the tooth tip).

2) Progressive loss of lubricant oil viscosity characteristics or temporarily insufficient lubrication (Cf. case of open gears).

3) Unforeseen and brutal overloads.

4) Passage of a foreign body (adherent metallic particle).

When we are faced with such degradation, the remedies are generally simple. After checking that the gear teeth precision and their eventual profile and helix modifications fulfill their function, one must:

1) Assure that the lubricant and lubrication devices are in working order and capable of performing their function.

2) Eliminate the risk of untimely occasional overloads applied to the gear (Torsional vibrations, high torque variation, pumping effect, etc.).

For added safety the lubricant viscosity can be increased (if necessary). Fig. 7 gives an example of metal adhesions encountered on several teeth of a spiral bevel pinion.

Hot Scuffing. This wear phenomenon results from oil film breakage under excessive overheating during meshing, which causes a metal-to-metal contact of teeth flanks. Local welds and shearing are alternately produced between the contact surfaces and contribute to the quick pull-off of metallic particles from the teeth flanks, thus progressively modifying the state of their profile.

The scuffing traces appear in the form of streaks or scratches with rough bottoms and sides, often appearing as bands of variable depth widths oriented in the direction of the height of the tooth, and affect isolated zones or their whole width. The scuffing traces are generally more clearly marked at the tooth tip and root of the teeth in the high sliding zones.

Fig. 7 - Metal adhesion on two teeth of a spiral bevel pinion. Note the metal pull-off located on the contact surface and in the maximum slipping zone with transverse scratches in the direction of the sliding motion.

In the case of hot scuffing, it is the combined action of high pressure between surfaces, high sliding speeds, and excessive contact temperature, resulting from pressure and sliding speed values, which causes oil film rupture between the teeth flanks. During start-up or running-in of certain gears, some local scuffing of lesser importance, which is characterized by shallow traces and very fine roughness, may appear in certain points of the teeth in the zone where the contact pressure is maximum. In general, after a certain time of operation at reduced load, these localized traces of scuffing diminish by wear. Once this happens, the gear may be operated under its nominal load. In this case, a slight increase in the lubricant viscosity will allow a better safety in service. On ground gears, we can see the presence of localized scuffing at the tip and root of the teeth, which is the result of insufficient tip relief or too great a deviation in the profile. We can also encounter identical phenomena near tooth ends due to insufficient longitudinal corrections or too great helix deviation. As long as scuffing traces caused by these phenomena remain slight or shallow, they may be tolerated in service, for they will end up wearing off and reducing in time. (Fig. 8)

If they become coarse, the scuffing can evolve either toward periodic adhesions or toward more severe scuffing, which may become destructive. In general, the appearance of destructive scuffing is revealed in service by brutal oil temperature rise at the meshing exit. If the load and the operating conditions are maintained, the scratches will generalize on the whole surface of the flanks and will become deeper. There is also the risk of local metal pull-off, transfer of metallic particles, and progressive deformations of the profile surfaces.

When this deformation becomes serious enough, the noise level of the gear will increase. The same is true for the temperature of the wheel and pinion bodies.

If the operation is maintained under such conditions, signs of overheating will quickly appear on the teeth surface (brownish, bluish, or violet tempering colors).

In the case of surface-hardened gears, cracks will often occur under thermal stress of teeth, which lead quickly to their breaking.

In the case through-hardened gears, we can ascertain some extreme cases of very serious rise in temperature of gears able to cause a hot flow of the teeth, bringing their destruction.

General scuffing is a brutal phenomenon which can develop very quickly on the teeth (in the matter of a few minutes to a few hours) and bring about *irremediable gear failure*. It occurs because of thermal stress, which leads quickly to tooth breakage. (Figs. 9-10)

When the failure occurs a long time after start-up of the gearing, it is caused by an accidental faulty lubrication or an overload resulting from continuous use of the machine in a manner for which it was not designed. However, if failure occurs a short time after start-up, one should check whether the amount of heat generated by the gear is compatible with the choice of the gear geometry and the choice of lubricant.

Cold Scuffing. This wear phenomenon is the result of lubricant (oil or grease) film breakage under excessive pressure action during meshing, causing a metal-to-metal contact of teeth flanks. The generation mechanism of degradation is identical to that of hot scuffing (welding and metal pull-off). In this case, it concerns only the joint action of high pressure between surfaces or extremely low sliding velocity (linear speed not over approximately 4m/s), which causes the breakage of the lubricant film between the contact profiles.

In general this type of scuffing begins with one or more adhesions on a few teeth flanks and spreads gradually nearer and nearer until the whole circumference of the gear is involved. The spreading speed of damage is a function of the type of lubricant used, the finishing process, and the hardness of the gear materials.

If we consider a gear of normalized steel or through-hardened, rough-cut finished, and grease lubricated, the destruction process of the teeth can be very quick once the lubrication becomes insufficient for the load transmitted (from a few minutes to a few hours).

We then see the formation of successive deep metal pull-offs and adherences on the flanks, which are partially rolled during meshing, thus causing an emission of metallic particles which remain blended in the grease. These particles generally initiate an abrasive wear which in a way re-establishes a uniform contact surface on the teeth, but in many cases is not sufficient to stop the scuffing.

This degradation mechanism will result in overheating of the surface in contact, which generally will remain at an acceptable level because the sliding velocities are low, and in the appearance of vibrations and meshing noises in the installation. When we stop the gear, we find teeth profiles which are often deeply altered in their geometry and which we can consider as being practically destroyed. In this case, it is essential to repair the teeth

Fig. 8 - Moderate scuffing on the flanks of a case-hardened gear. Note the streaks spread over in bands and more or less intense on the profile. This scuffing is not destructive, for it does not notably alter the profile. This gearing can operate again after a run-in period and with better lubrication and load in service conditions.

Fig. 9 - Destructive scuffing on the teeth of a spiral bevel wheel. Note the presence of deep streaks and important deformations of the profile in the contact zones. This gear wheel can be considered as destroyed.

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before planning to re-start the gear.

Whenever possible, it is preferable to proceed to a machining of the teeth at least so as to remove the particles embedded on the teeth flanks.

In the cases where we see the presence of too many metal pull-offs, we should make a more complete machining on the wheel flanks, which will require the design of a new pinion if we want to preserve an identical backlash to the one initially planned.

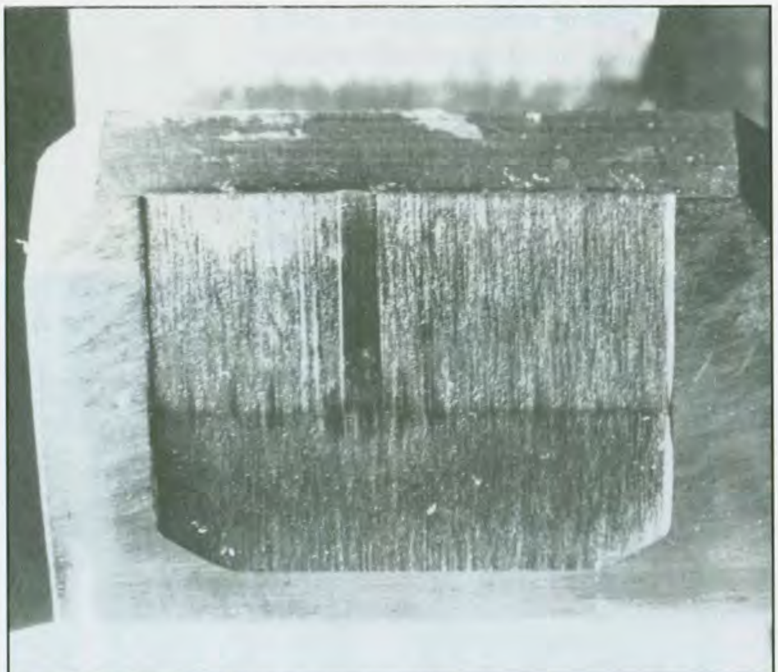

Fig. 10 - General and destructive scuffing on a gear wheel. Note an important wear of the profile with the presence of radial rough irregular streaks. Also note a metal pull-off toward the tip of the tooth.

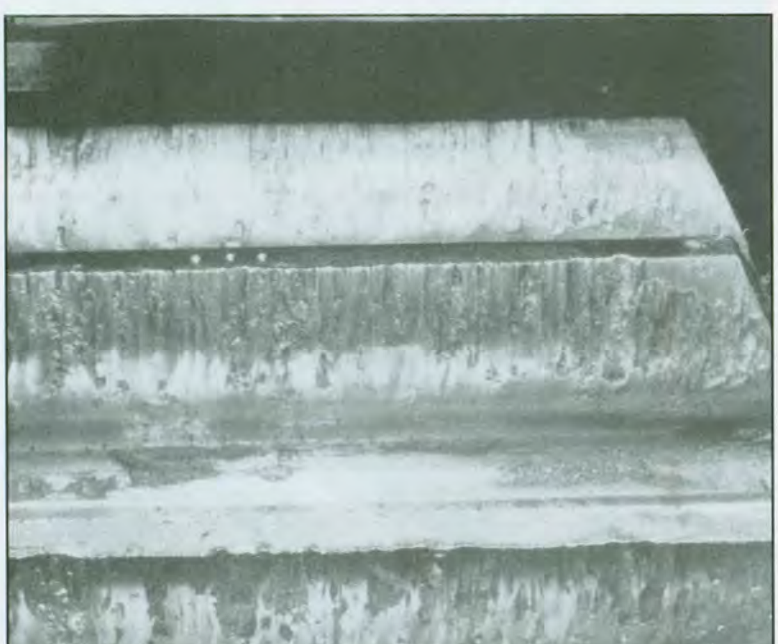

Fig. 11 - Cold scuffing on the teeth of a winch wheel.

When this scuffing happens on large, open gears used for driving ovens, crushers, or winches, the repair has to be done most of the time *in situ,* for it is often impossible to remove such gears. (Fig.11)

We can, as a first operation, remove by manual grinding all welded metallic particles that are too thick in relation to the initial profile of the teeth. Then we must run-in the teeth before thinking of restarting of the gear to obtain a good contact pattern on the flanks.

For this operation, we can use an abrasive compound of grain size structure adapted to the size of the teeth or special synthetic graphite grease. The running-in should not be too harsh, otherwise we risk deforming the profile of the teeth by abrasive wear.

If we have a surface-hardened and ground gear often used at the low speed stage of a reducer, the metal pull-off on the flanks is often accompanied by plastic micro-distortion of flanks due to embedding and micro-welding of large particles, which may damage the hardened surface layer because it has to support very heavy local pressure for which it has not been designed.

The surface overheating generated by the micro-weldings may be high and can also sometimes generate surface cracks, which may evolve rapidly in service through the hardened surface layer and lead to the rupture of one or more teeth.

For such gears in general, the appearance of cold scuffing is brutal and often results in their breaking during use.

The possible causes of cold scuffing are insufficient lubrication or a severe overload of the gearing over a sufficiently long period. (These two causes can occur simultaneously).

This type of damage is generally destructive for the teeth and develops most often on grease-lubricated teeth. The surface roughness of the teeth is of great importance, and when it is great, it encourages the formation of micro-welding and the development of deep scuffing. ■

Part II of this article will cover wear with three bodies, interference wear, and varying intensities of gear tooth wear.

Acknowledgements: *Published as AGMA document 88 FTM 4 and in Europower Transmision, April, 1991. Reprinted with permission.*

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ADVERTISER'S INDEX

Management By Walking Around

Keep on top of your business by keeping in touch with what's happening in the office and on the factory floor.

Richard G. Ensman, Jr.

Have you ever been confronted by a thorny business problem, only to discover - belatedly - that it had been creeping up on you for months, or even years?

In today's fast-paced business world, emerging problems can easily go unnoticed for long periods of time. Managers can become so engrossed with the pressing concerns of day-to-day operations that they fail to observe the tell-tale signs of upcoming difficulties.

If you want to develop early warning signs of business problems, you have to work at it. While you may be able to keep abreast of trends through formal reports and meetings, these information systems won't tell you the whole story. To find out what's really happening in your business, you must be willing to invest a little time and effort "behind the scenes." Here's how:

Walk the beat. Once a day, take some time to walk through your place of business. Stop and talk to your employees every once in a while. Ask them what's going on. Ask them to explain the kind of work they're doing and the problems they're encountering. And ask them about their concerns and accomplishments.

Recruit a mystery customer. Ask a friend or colleague to visit your place of business. You might ask your "mystery customer" to offer you a casual, candid reaction to the performance of your sales force or customer service people - or you might ask him to pose specific problems to your staff and observe their reactions.

Set up a simple focus group. Draw together a random group of customers, employees, or vendors and ask them to offer candid comments on your product lines, management practices, or other items of concern to you. Whatever issue you bring to this group's attention, don't forget to ask members to assess the image you and your business firm enjoy in the community.

Write a letter. Under a fictitious name and address, write a letter to your own

MANAGEMENT MATTERS

company asking for information about a product, help with a customer service problem, or a quick price quotation. Note your staff's response: How long a time before the letter is answered? How does the answer come? By letter? By telephone? What's the overall quality of the response?

Take a file test. Next time you need detailed information on some subject, keep track of how long it takes to get a response. How many hours or days elapse before someone on your staff brings you the needed information? Is the information accurate and complete? Are the records you obtain in tip-top shape?

Ask candid questions. During one of your casual

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Richard G. Ensman, Jr.
is a free-lance writer from Rochester, NY. He specializes in topics for business and trade magazines.

visits to front line management, ask supervisors a few candid, but seemingly inconsequential, questions. Ask, for instance, about the work schedules of various employees; about the day-to-day responsibilities and accomplishments of the people under their supervision. You'll end up understanding how well your front line people grasp the intricacies of day-to-day operations.

Play the memo game. Want to find out how quickly you can get action on some issue? Next time you need to send out a memorandum asking for verification of some piece of information, an analysis of a problem, or a recommendation on some current issue facing the company, keep track of the response time. When - and how - does the response find its way to you?

Call your office. Some afternoon, when you're out of the office, call in. How many times does the telephone ring before the receptionist picks up the receiver? How pleasant does the receptionist's voice sound? How are you greeted? If your call is placed on "hold," how long are you kept waiting? And how helpful is the receptionist when your call is answered?

Keep an eye on the junk. In the course of one of your walk-arounds, make note of all the discards from your assembly floor or waste paper from your office. What's being discarded? How much money do you suspect is lost in waste that day? Do your people have any suggestions for reducing waste - and improving efficiency - in the future?

Yell "help" to the computer. Ask for computer data that you know is only a few days old. Accounts receivable data from a very recent sale makes a good test, as does mailing list information that's come in over the last few days. Can your staff get you the information? If not, how far behind is your company in data entry? Why? What can your computer people or office staff do to speed up data entry and processing?

Check the money. Pose a hypothetical question to your top financial person: How much money is left over in your budget, say for the purchase of new office furniture? Or how much profit does he or she see accruing to the company in the current quarter? Can you get quick answers to these and other budgetary questions? Can the money person put his or her hands on financial records quickly and efficiently? Can he make fast and accurate financial projections?

Put yourself in the other guy's shoes. Make it a point to spend an hour or two with a single employee - selected at random - every so often. During the time, see if you can learn the essentials of his job: recording cash receipts, processing customer service problems, assembling products before distribution, or whatever. Ask as many questions as you can during these random learning sessions - and don't hesitate to follow up later on with questions about efficiency, quality control and work procedures with supervisors.

Read. Ask one of your employees or managers to give you a random batch of outgoing correspondence - memoranda, letters, and reports - some morning. Read it, all of it. What's the overall quality of this material? Is it clear and concise? Does it appear pleasing to the eye? Do you notice any glaring typographical errors? What kind of image do you think your correspondence conveys to the public? If you could make any improvements in the correspondence, what would you do?

Be a doorstop. At the end of the day, stand near the door. Listen to the buzz of informal conversation going on around you. Are people eager to get out of the office or plant? Do you notice some employees staying late to complete important projects? Do you hear comments indicating job satisfaction - or dissatisfaction - from your people? What does the overall tenor of the conversation tell you about the attitudes and morale of your employees?

Complain. Ask a friend or colleague to make a complaint about one of your firm's products or services. How effectively and graciously is the complaint handled? How do your customer service people offer to make amends for the problem or address the issues brought to their attention? Nothing will give you better insight to your customer service capability than the comments of a complaining customer.

Formal reports, meetings and information systems all help you understand your firm's "big picture." But the time and effort you spend around the desks of your employees, the water cooler, and the copy machines will give you the tools you need to better understand the day-to-day strengths and weaknesses of your company.

Ultimately, this time and effort will help you identify nagging problems - and solve them before they threaten the profit or stability of your business. Management by walking around may not always be enjoyable but, if it gives you insights into troubles you didn't know you had, it may well be profitable. ■

MANAGEMENT MATTERS

Meetings and reports may convey the "big picture", but talk around the watercooler can fill in important blanks.

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NOVEMBER 9-10
Workshop on Applied Induction Heat Treating for Industry, Detroit, MI. Sponsored by the Center for Industrial Heat Treating Processes, University of Cincinnati, this workshop will cover the fundamental principles of electromagnetic induction, equipment and power supplies, induction heat treating metallurgy, coil design, quenching, quality control, and troubleshooting.

NOVEMBER 11-12
Workshop on Quenching Processes and Control in Heat Treating, Detroit, MI. This University of Cincinnati program will cover methodology for evaluation and comparison of quenching parameters and intensity, practical approaches to minimizing residual stresses and distortion, prediction of hardness distribution in as-quenched parts, and implementing statistical process control to improve quality. For more information on these two workshops, contact Dr. A. H. Soni, University of Cincinnati, (513) 556-2710, FAX (513) 556-3390.

DECEMBER 2-4
Fundamentals of Gear Design. University of Wisconsin, Milwaukee, Milwaukee, WI. Annual course in history, types of gears, theory of gear tooth action, manufacturing methods, inspection techniques, failure modes, and lubrication. Taught by Ray Drago. For more information, contact Richard G. Albers, (414) 227-3125.

MARCH 3-4, 1993
Dudley Gear Seminar. San Diego, CA. Two-day course covering history, gear failures, rating practice, materials & heat treatment, lubrication, vibration, manufacturing, and other special topics. Call Dudley Technical Group, Inc. (800) 354-5178 of fax (619) 487-4893 for futher information.

MARCH 8-11, 1993
National Manufacturing Week℠. National Design Engineering Show & Conference. McCormick Place North, Chicago, IL. Sponsored by the National Association of Manufacturers, this is a comprehensive design engineering event featuring design systems, OEM components, CAD/CAM systems and services for product design engineering. Held in conjunction with the National Plant Engineering & Maintenance Show & Conference and the International Control Engineering Exposition & Conference. For more information, contact NAM at (203) 964-0000 or FAX (203) 964-8489.

APRIL 21-28, 1993
Hannover Fair, Hannover, Germany. The world's largest exhibit of industrial equipment, components, and systems by companies from around the world. There will be an AGMA Group Pavilion in the Power Transmission and Control Sector of the show. For information about attendance or exhibiting, contact Kurt Medert at AGMA Headquarters, (703) 684-0211.

ADVERTORIAL

CNC Controlled High Precision Gear Grinding

HÖFLER NOW OFFERS A COMPLETE RANGE OF FULLY AUTOMATIC, CNC CONTROLLED GEAR GRINDING MACHINES

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CNC technology controls the generating and indexing paths (eliminating the change and index gear train), workpiece diameter setting, helix angle setting, grinding slide stroke and grinding wheel slide. In addition, these machines feature a 4-axes, CNC-controlled grinding wheel dresser.

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- NOVA CNC,
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Every HÖFLER gear grinding machine delivers a high performance production accuracy of AGMA 14 under normal production conditions.

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If you would like to know more about these and other HÖFLER gear grinding products and innovations, please contact them at one of the following addresses at your convenience. HÖFLER is looking forward to hearing from you.

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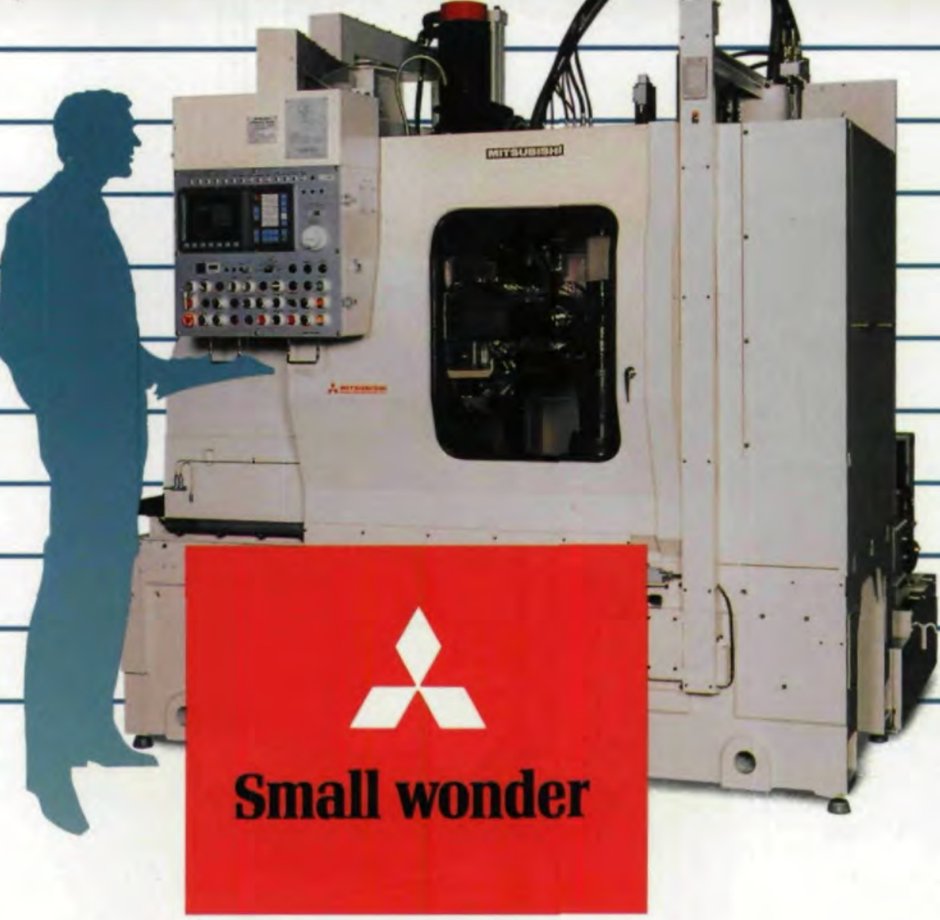

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