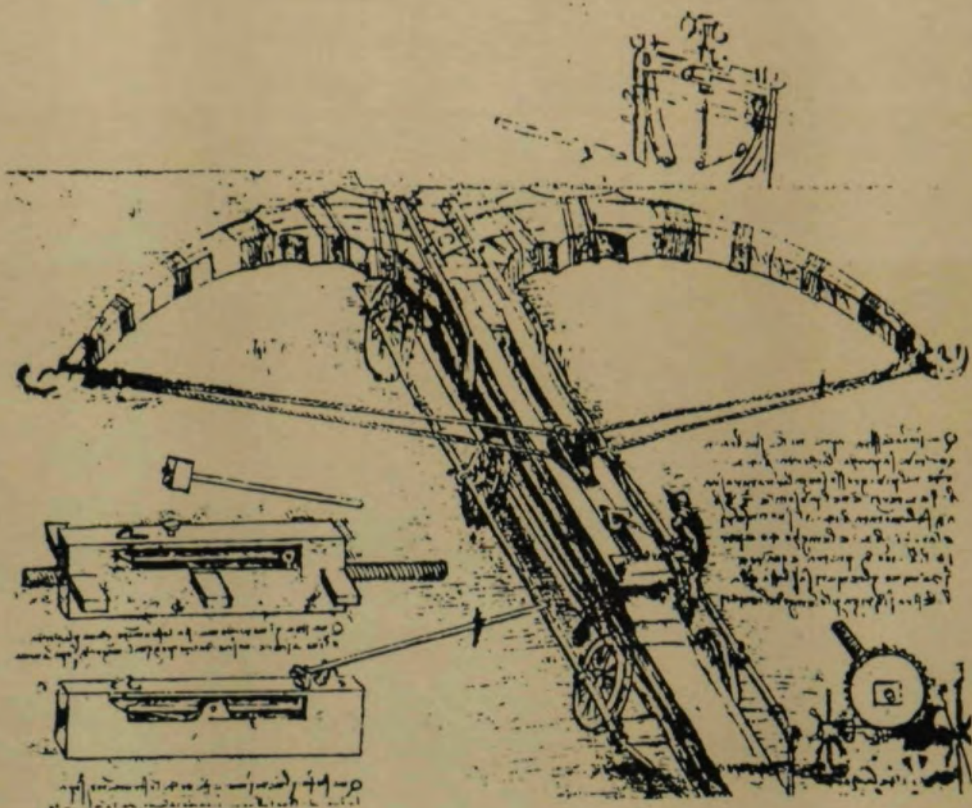
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

MARCH/APRIL 1990



Gear Noise & Silent Gears

Influence of Lubricants on Pitting & Micropitting

Dynamic Loads in Parallel Shaft Transmissions — I

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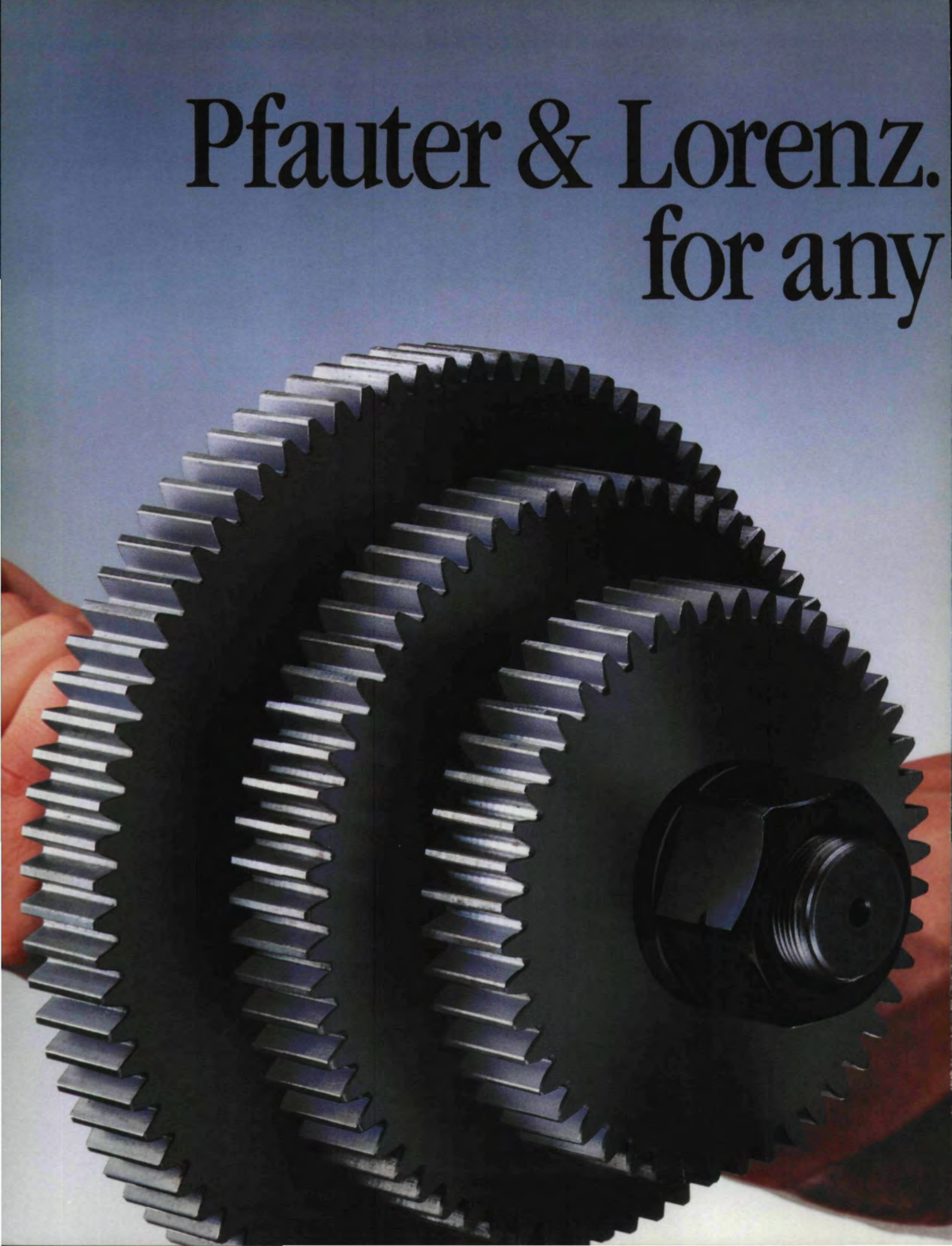
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The Advanced Technology of Leonardo da Vinci 1452-1519

COVER

Giant six-wheeled crossbow. Leonardo did a great deal of work in weaponry. This study for a crossbow is probably beyond the capacity of contemporary workshops and was never built. The bowstrings were drawn back by a sleeve attached to a worm and gear assembly, and the arrows were released by either a hammer-and-pin or by a latch-and-lever arrangement. Leonardo claimed the weapon was capable of silent operation.

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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March/April, 1990 — Vol. 7, No. 2

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Editorial

CAPITAL GAINS, SOCIETAL GAINS, OR NO GAINS AT ALL.

Taxes may be one of the only two sure things in life, but that doesn't make them popular. Nobody is happy to pay them, and the bigger the amount due, the unhappier the taxpayer. Conversely, politicians know that coming out in favor of a tax cut is the equivalent of voting for apple pie and motherhood. It's a sure-fire success at the ballot box.

Therefore, it should come as no surprise that Congress – in spite of the massive federal deficit – is again considering cutting taxes, this time on capital gains. No matter that the benefits of such a cut are debatable at best; that cutting taxes in the face of the largest deficit in our history is based on logic right out of *Alice in Wonderland*, or that a capital gains tax cut will probably benefit comparatively few Americans. Tax cuts of any kind play well in Peoria and everywhere else. According to Speaker of the House, Tom Foley, if you tell an ordinary taxpayer that he will reap $10 from a measure that will save the likes of Donald Trump an average of $25,000 in one year, he will say, "Fine, give me my $10.00."

Perhaps that's true. Certainly it makes a convenient excuse for our elected officals when they are explaining why they are going to take the easy way out – again.

Before we go farther, I think it's important to remind ourselves of a fundamental fact about tax law. Unlike the rules in engineering, mathematics, or

other kinds of accounting, tax laws do not follow any clearly apparent logic. There is no particular inevitability about them because, contrary to what may appear on the surface, the purpose of taxes is far more than just raising money to pay for running the government. The power of taxation is the power to engineer society, to redistribute wealth, to encourage certain economic practices and make others prohibitively expensive.

Given this enormous implicit power, we should think long and hard about exactly what we are doing when we support tax law changes. Before we jump on the capital gains tax cut band wagon, we should ask the all-important question, *Cui bono?* Who benefits? Will anyone? Will a cut in the capital gains tax get the money to where we want it to go to do the things we as a country want and need to have done?

My own feeling is that the answer is "No."

Some of the economic issues which we need to focus on in the next decade(s) include lowering the deficit, increasing the personal savings rate of Americans, lowering the cost of capital, and encouraging more long-term investment, all of which will improve our economic health and make us more competitive. I don't think the capital gains tax cut is the most effective way to achieve any of these goals.

This tax cut will not, in the long run, do anything to lower the deficit – probably our number one economic problem. All the voodoo economics to the contrary, somewhere down the road, the piper will have to be paid. The only way the deficit is going to come down is by raising taxes or cutting spending or – inevitably, I think – both. The logic that a cut in the capital gains tax will spur productive investment is debatable to begin with; and while it may provide an initial surge in revenues – approximately $3 to $4 billion annually for the first couple of years – after that, it would begin costing the Treasury an estimated $5 billion a year in lost receipts. That's $4 or $5 billion a year we can't afford if we are interested in cutting the deficit.

Treasury Secretary, Nicholas Brady, told the American Business Conference last year, "We have been willing to mortgage our future, to cut corners in pursuit of immediate payoff." Voting for a politically palatable tax cut now for any segment of the economy and worrying about the consequences later is another case of that currently all-too-popular American pastime – taking the money and running.

Even if any kind of a tax cut made sense, of equal importance is the fact that the capital gains tax cut does not address those other crucial concerns in our economy – encouraging savings by individuals, lowering the cost of capital, and encouraging long-term investment in improving national productivity. Whatever its benefits, the capital gains tax will impact a relatively few, relatively prosperous Americans. It will do little to encourage the average American, whose income is below $50,000, to save. Consequently, reintroducing some form of tax-deferred or tax-free savings, like the IRA, might be a better move toward the goal of encouraging Americans to save more and spend less. It would affect far more people than the capital gains tax and address a crucial societal need.

In a recent article in the *Wall Street Journal,* Peter Drucker points out that historically, tax exemptions or deferments on savings are most often taken advantage of by lower and middle income people. Even when the rate of return on such plans is small, they are popular with the majority of people – and they work. According to Drucker, the tax exempt savings account was one of the chief instruments in turning Japan from a country with one of the lowest savings rates (prior to World War II) to a country with one of the highest in the world.

Increasing savings would go a long way toward decreasing the cost of capital in the U.S. The advantage that other countries have over us in this area is one of the factors that strip us of our competitiveness in global markets. A U.S. firm might have to pay as much as 10-15% to borrow money for a project with a 10-year payoff, while a Japanese firm, only 5%.

It is true that a cut in the capital gains tax would encourage some forms of investment and some industries, but given that the taxing power has implications far beyond the mere raising of money, I wonder if these are the industries and investments we wish to encourage right now. Some of the chief beneficiaries of a capital gains tax cut would be real estate and the securities markets, and we must ask if strengthening them benefits the economy as a whole. People in these businesses do not make things; they make deals; and the question is, are more deals or more *competitive products* what the country needs now? Real estate markets are already over-inflated in many major metropolitan areas. Do the buyers and sellers of real estate need more encouragement to keep up the pace of turnover just to benefit from a capital gains tax cut? In securities trading, the question is even more pertinent. Many economists and securities traders themselves are beginning to question the benefits of leveraged buyout fever. Many viable companies have been gutted and trashed by high-powered investors seeking short-term profits. A cut in the capital gains tax, unless it applied only to assets held over some long-term period, like five years, would only encourage this kind of short-term, short-sighted profit taking.

Instead of subsidizing this kind of deal-making mania, a look at tax incentives that would encourage more saving and long term investment in the *production and manufacturing* portion of the economy would seem to set the right priorities. Those of us in capital intensive businesses like gear manufacturing are all too aware of the burden that the high cost of money places on us. And we are acutely aware of the need for more money to be spent on research, development, and purchases of resources to get and keep America's manufacturing base viable and competitive. A tax break that would lower the cost of borrowing and long term investment would be a direct benefit for us, but, more important, it would be a step in the right direction for our economy as a whole.

Altering the taxing practices and spending habits of a generation addicted to consumerism and instant gratification will not be easy, but not impossible. There are plenty of suggestions out there from reinstituting the IRA to giving tax breaks to companies engaged in long-term research, but because they are not quick, simple, or politically popular, they die on the vine for lack of interest. If the members of Congress would devote as much creativity to structuring the tax code to encourage savings and long-term investment as they do to avoiding the political consequences for voting their own pay raises, we might all be better off.

But in the final analysis, the crucial question is not really one of taxes at all. Which taxes sound like the best ones will depend to a good extent on whose pockets get emptied. As George Bernard Shaw says, "The government which robs Peter to pay Paul can always depend on the support of Paul."

The issue here, it seems to me, is whether or not we have a Congress with the courage to use the taxing power as it was meant to be used. Does it have the courage to tell the American people what it can afford and what it can't? Can it risk the short-term displeasure of the few – or maybe even of the many – for the long-term gain of the whole society? Does it have the nerve to do the right thing rather than the politically expedient thing?

I don't know. I do know that getting America back on track economically is not something that will come cheap or easy or overnight; that the power to tax is a crucial tool for making the economic and social policy required to get the job done; and that it will take leaders, not politicians, in government to use that tool properly.

EDITOR'S NOTE: As we go to press, there is talk that the President might reintroduce the capital gains tax proposal, along with a plan yet to be formulated, to encourage savings. We'll all have to watch how the President and Congress handle yet another opportunity to be statesmen instead of politicians.

Michael Goldstein

Michael Goldstein,
Editor/Publisher

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Gear Noise and the Making of Silent Gears

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Summary:

Our research group has been engaged in the study of gear noise for some nine years and has succeeded in cutting the noise from an average level of some 81-83 dB to 76-78 dB by both experimental and theoretical research. Experimental research centered on the investigation into the relation between the gear error and noise. Theoretical research centered on the geometry and kinematics of the meshing process of gears with geometric error. A phenomenon called "out-of-bound meshing of gears" was discovered and mathematically proven, and an in-depth analysis of the change-over process from the meshing of one pair of teeth to the next is followed, which leads to the conclusion we are using to solve the gear noise problem. The authors also suggest some optimized profiles to ensure silent transmission, and a new definition of profile error is suggested.

Introduction

For some nine years, our research group has been engaging in the study of gear noise and the making of silent gears. six papers in English have been published on the international conferences and periodicals (see Reference 1-6). Experiments were done on machine tool headstocks' power gears, as they represent the sort of light-load, medium-speed gears for which silent transmission is such a problem. A new single flank total composite error tester (Initiated by Huang Tonglian of China. See Reference 4.) was used to find the relation between gear geometry and the noise. The tester is able to indicate the pitch error, the profile error, the final transmission error and the change-over character (from the meshing of one pair of teeth to the next). Conclusions drawn from these experiments have also been varified by machine testing. When we reached the required accuracy, the gears offer the expected silent transmission. (Under 78 dB measured at 300mm away from pitch point, minimum 73 dB.)

For theoretical analysis, a deep investigation has been made on the meshing process of involute spur gears. A phenomenon called "out-of-bound meshing" of gears was discovered; that is, with the presence of error, the actual contact line of a pair of gears is much discrepant from the theoretical one. It usually goes along a broken line linked by a sector of the addendum circle of one gear and a part of the theoretical contact line.[4-5] With the disclosure of this phenomenon a precise and thorough statement on the change-over process of gear meshing (a geometry and kinematics of the change-over process of gear meshing) is made.

Summarizing from the above mentioned experiments and analysis we formed the following hypothesis concerning gear noise.

Dynamic Behavior and Change-Over Process of Gear Transmission

Out-of-bound meshing and change-over impact of gears. Since gear noise is largely induced from change-over impact, thorough analysis of the change-over process must be made. But before entering the study of it, two things must be mentioned.

a. *Out-of-bound meshing (OBM) of gears.* — When a gear is meshing with its pinion, theoretically speaking, the contact point will move along the common tangent 1-4 (See Table 1a), starting from 1 (the intersecting point of addendum circle of driven gear with g1g2), and ending at 4 (the intersecting point of addendum circle of driving gear with g1g2). On most occasions, there are two adjacent pairs of teeth in mesh, but, actually, there is always a difference in normal pitches of driving and driven gears. Assume $t'_0 > t_0$ as in the case of Table 1a, the contact point, after passing 4, will continue the meshing and the gap at 1' will be narrowed until at point 4", the next pair of teeth come into meshing at e' and take over the transmission. So the actual contact line is a broken line e'44". There is a shock at the instant of take-over, called "leaving impact," of OBM. Table 1b shows the other case, $t'_0 < t_0$, where the contact line is 2"2e. The impact thus produced is called approaching impact.[4-5]

b. *The single flank total composite gear error tester.* — It is the main measuring instrument we have been using in this

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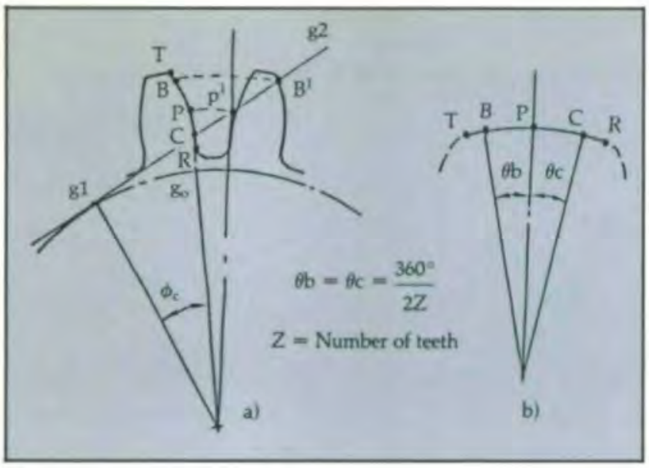


Fig. 1

research work. It is an all-round gear tester developed from the ordinary single flank gear transmission error tester.[4] The result coming out is the transmission error of every individual pair of teeth (from tip to root), but overlapped into a polar coordinate, where ϱ of the polar coordinate represents the transmission error, while θ of the polar coordinate represents the angular position of gear, which determines the position of contact point on the profile, expressed by its generating angle ϕ_c. (See Fig. 1a) The greatest merit of this tester is its thorough exposure of the change-over characteristic. Fig. 2 shows the expression of different errors on the tester's record. Fig. 1 indicates the characteristic points on the profile. T and R are the tip and root points of the whole profile. The contact line is g1g2. The pitch point is p'. The generating angle of point c is ϕ_c, which also indicates the angular position of the gear when the contact point passes c. B C is the basic section of profile which offers a contact ratio $=1$. Fig. 1b shows the location of the points on the tester's record, where $\theta_b = \theta_c = 360° / (2z)$, where z = the number of teeth. Dot lines show the OBM. TB and CR are sections generally considered for tip/root relief.

Now let us see how the change-over takes place. From Table 1a, 1b, one may think that the way of change-over could be either as shown in Table 1a or Table 1b. But remember that both Table 1a and Table 1b are based on the assumption that the profile is an involute curve with error in pressure angle (generated from a base circle with error in diameter), but without tip (root) relief, (TR in Fig. 3a.) Here the change-over point falls on the OBM zone. Now let's see Fig. 3. If the profile has tip relief 1-2 large enough to cover the normal pitch error, then the change-over point e' will fall within the profile 21R. It is called inside change-over as shown in Table 1c. It is a change-over without OBM. The convex profiles (See Fig. 14, Fig. 15) also belong to this group.

Fig. 3b shows the case with profile shape error. It is a typical saddle form profile. The pressure angle is too small, with a relative relief, and the outside change-over will be transformed into inside change-over.

Table 1c shows a profile with normal pressure angle, but with tip relief to obtain inside change-over. Since the tip relief should be large enough to cover the normal pitch error, a fairly large intersecting angle θ results, and consequenently, little benefit could be gained in reducing noise. Table 1d shows another way of transforming outside change-over into inside change-over. It is done by adapting the pressure angle to obtain correct normal pitch (normal pitch $t_o = t \cdot \cos\alpha$, varying α, we can get right t_o), expressing on the tester's record, the geometric expression of right t_o is that the adjacent curves will link up very well. So if the pressure angle can be adapted to make the adjacent curves meet very well, a very small "relative relief" can guarantee the inside change-over,

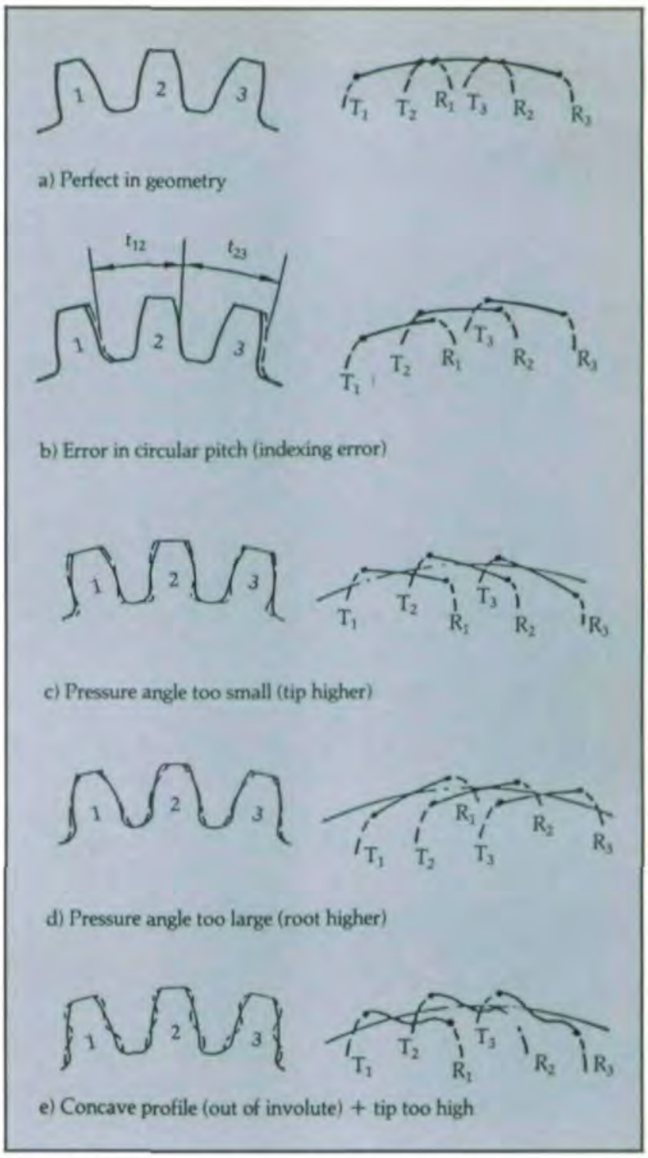


Fig. 2 – Geometric errors expressed on tester's record.

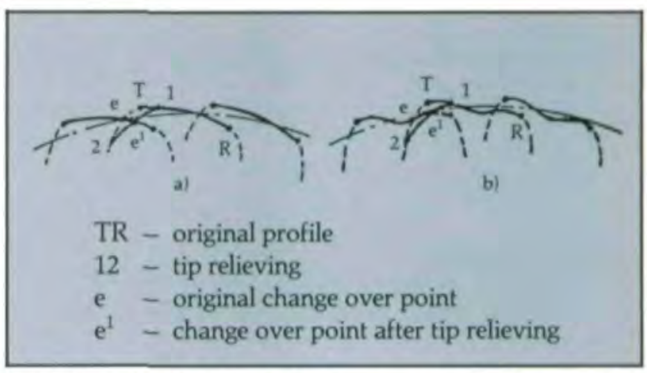
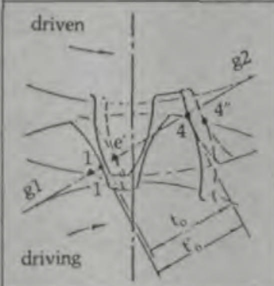
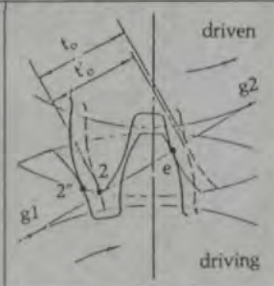
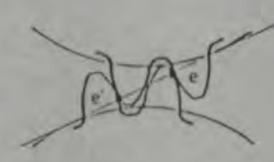
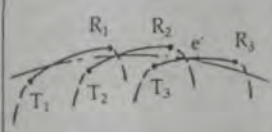
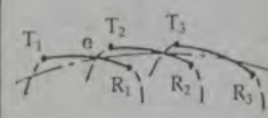
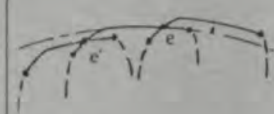
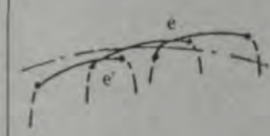
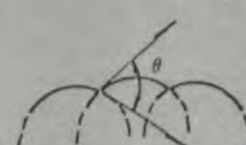


Fig. 3 – Transformation of outside change over to inside change over by tip relieving

Table 1 Change-over character of gears				
	Outside change-over		Inside change-over	
Figure of contact				
Contact line	e'4-4"	2"-2-e	e'-e	
Tester's record	 Type (a)	 Type (b)	 Type (c)	 Type (d)
Impact happens	at leaving	at approaching	at change-over point e' e	
Brief Description	Change-over impact induced by too small pressure angle	Change-over impact induced by too large pressure angle	Pressure angle normal, inside change-over obtained by tip (root) relief which should be great enough to cover the indexing error (error in circular pitch)	Inside change-over obtained by adapting pressure angle to obtain correct normal pitch (in tester's record to link up the adjacent curves), with slightest tip (root) relief to avoid OBM
Characteristic parameter for gear noise	Normal pitch difference $\Delta t_o = t_o$ (driving)-t_o(driven) t_o= normal pitch of gear		Intersecting angle θ at change-over point 	

while intersecting angle θ, and hence the impact can be kept very small. It is apparent that this way of obtaining inside change-over (Table 1d) is much more favorable than the previous one (Table 1c).

Outside change-over is generally considered as unfavorable. This is because, in the OBM process, the tip of one gear tooth actually does not mesh with the profile of the other gear. There is no common tangent for them. The tip edge of one gear tooth just scrapes the profile of other gear, and what is more, *the OBM curve in the tester's record is steep.* (See Fig. 2c.) This results in a very large intersecting angle, causing impact. So a gear with approaching OBM is a typically noisy gear.

But leaving OBM acts quite differently. This is due to the phenomenon of losing contact in gear meshing. As shown in Fig. 5, it goes along the dot line. (For detail see Fig. 16c.)

Experiments show that a gear system with normal pitch of the driving gear is a little bit larger than the normal pitch of the driven gear offers fairly silent transmission.

So there are four typical forms of change-over as listed in Table 1. For the outside change-over, the characteristic parameter is the normal pitch difference Δt_o; and for inside change-over, it is the intersecting angle θ which determines the impact and hence the noise. (See Fig. 4.)

Some discussion on low noise gear profile and the definition of profile error. In the preceding section, we have discussed the gear noise at the instant of take-over. Now let us see what happens during the whole process of meshing of a pair of tooth profiles (from root to tip). They work just like a pair of cams. According to the theory of conjugation, they *should be made into accurate involute curves (not concave* nor convex on tester's record) to ensure smooth transmission during this period. The characteristic parameter for noise in this case is the reciprocal of the radius of curvature of the curve on the tester's record. (See Reference 4.) But as mentioned above, in order to ensure smooth change-over, pressure angle should be adapted to link up the adjacent curves. So we suggest redefining the profile error, splitting it into two items:

a) Discrepancy from involute curve; that is, draw the two closest involute curves of the same base circle (not necessarily the nominated base circle). The normal distance of them, Δf

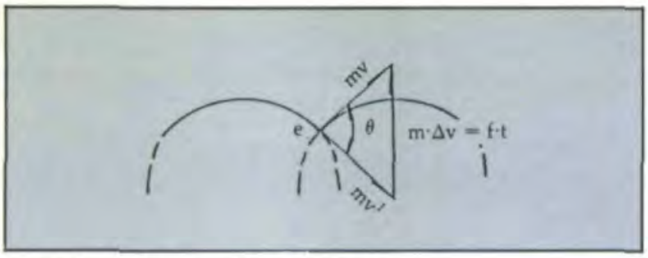


Fig. 4 – Vector diagram of change over impact.

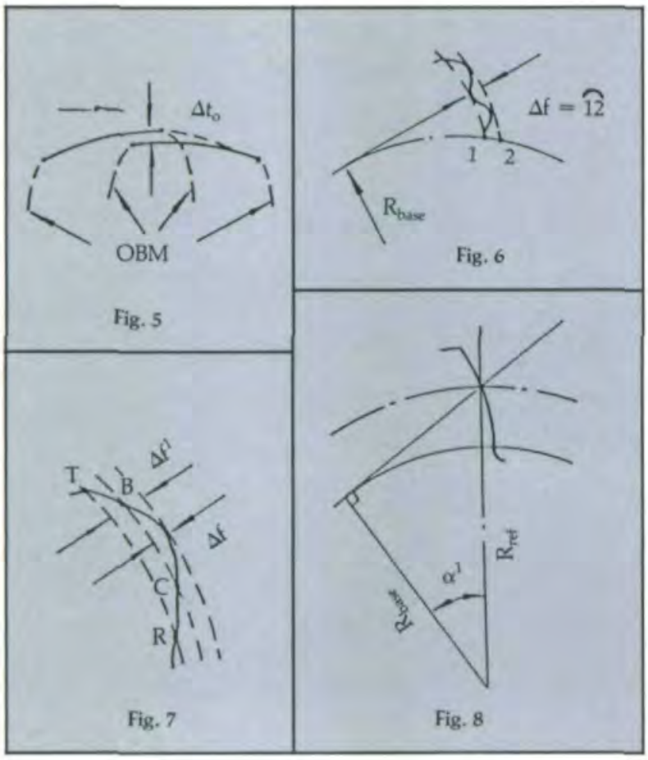


Fig. 5

Fig. 6

Fig. 7

Fig. 8

is defined as the profile error. (Fig. 6). If the BT/CR section is used for tip/root relief, then only the shape error of BC section will count. Tip/root relief should not be confused with profile error. As shown in Fig. 7, profile error is Δf instead of $\Delta f'$.

b) The actual pressure angle in this case is determined by the radius of base circle R_{base}; that is, $\alpha = \text{Cos}^{-1}(R_{base}/R_{ref})$. (See Fig. 8.)

Experimental Research on Gear Noise

Perhaps what interests the gear manufacturers most is the results of experimental research. They wish to know what items of accuaracy influence the noise most and how close tolerances must be on errors to ensure silent transmission. The purpose of experimental research was to find the answer for them, but we also did some other experiments with the purpose of investigating the phenomenon of gear noise.

Influence of gear error on the noise level. The experiment was done on m=2.5 z=50 B=20 lathe headstock gears. They were hobbed, shaved (without shave profile correction), hardened, and then subjected to a "controllable electrochemical honing"(3) with the intention obtaining samples with different magnitudes of errors. The profile of gear samples thus obtained are of typical saddle form, but different in concavity Δf. It is a pity that in this batch of gear samples, there was no convex profile, and some error in pressure angle was deliberately made. Gear noise and total transmission error were carefully measured. Altogether 242 noise levels were taken. They were listed on noise level sequence. Table 2 shows the two extremities, Nos. 1-26 are silent gears (below

Table 2 Gear noise and its geometric error

No.	dB	Driving gear number	Driven gear number	Δf_c	Δt_{oc}	No. of inverse gearing	Remark
1	75.7	1641	2161	−1.5	−1.5	2	
2	75.8	2161	1641	−1.5	+1.5	1	
3	76.8	2161	3041	−1	+1	27	
4	77.8	3041	0941	−1.5	+1	42	
5	77.6	22251	1641	−2.5	+4.5	7	
6	78	2161	2261	−1.5	−0.5	18	
7	78	1641	2251	−2.5	+1.5	5	
8	78	1641	1741	−2.5	−0.5	15	
9	78.2	2251	3041	−2	+4	53	
10	78.2	2161	1741	−2	+1	43	
11	78.3	3041	1641	−1.5	+0.5	14	
12	78.4	2251	1741	−3	+4	25	
13	78.5	1641	1321	−5	+3	50	
14	78.5	1641	3041	−1.5	−0.5	11	
15	78.7	1741	1641	−2.5	+0.5	8	
16	78.8	2251	2161	−2	+3	21	
17	78.8	1641	0941	−2	+0.5	73	
18	79	2261	2161	−1.5	+0.5	6	
19	79	0331	2971	−7.5	+3	65	
20	79.1	1331	1741	−4	−1	68	
21	79.2	2161	2251	−2	−3	16	
22	79.2	1641	0341	−3	+1.5	95	
23	79.2	3041	1321	−4.5	+3.5	124	
24	79.5	0341	0941	−3	−1	49	
25	79.5	1741	2251	−3	−4	12	
26	79.5	1261	0951	−9	+2.5	34	
223	84	0151	1331	−5.5	−6	142	
224	84	3042	2032	−8.5	−5.5	171	
225	84	0342	1642	−13	−2	172	
226	84	1432	2252	−9.5	−7	110	
227	84	0312	2032	−10	+1	228	
228	84.1	2032	0312	−10	−1	227	
229	84.2	0151	1332	−8	−7	159	
230	84.2	2032	0332	−11	+6	231	
231	84.3	0332	2032	−11	−6	230	
232	84.5	0332	1642	−13	−1	180	
233	84.5	0152	1332	−8	−7	160	
234	84.6	0151	2251	−4.5	−11	139	
235	84.6	1332	2031	−9.5	−8.5	210	
236	84.7	0152	0332	−8	−5	184	
237	84.8	1332	2032	−11	−4	166	
238	85	0151	0311	−8	−11	217	
239	85	0152	1331	−5.5	−6	155	
240	85	1432	2032	−10.5	−8	211	
241	85.5	0152	2252	−8	−10	173	
242	85.6	0331	2031	−8	−10	220	

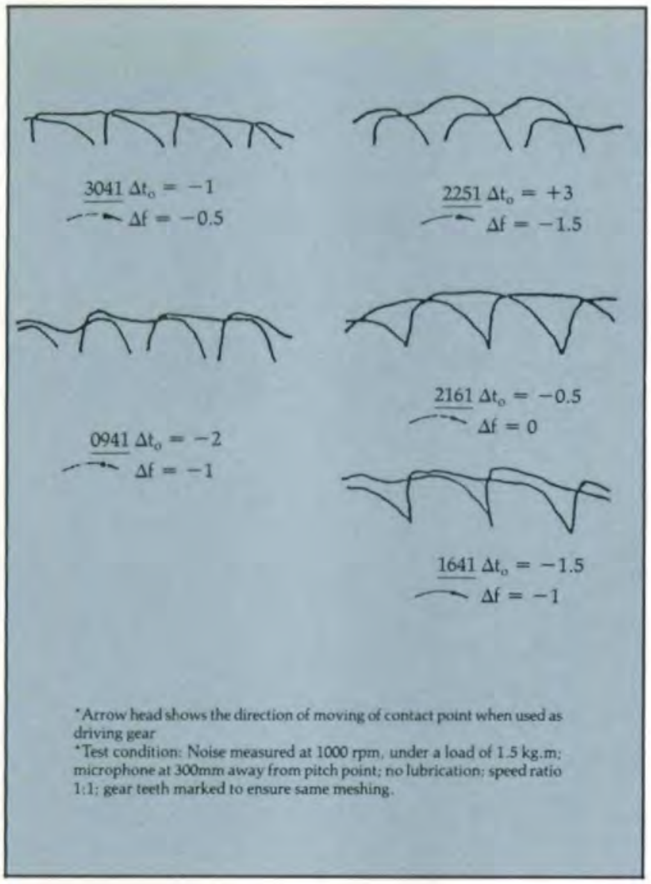


Fig. 9 – Tester's records of gears involved in the first 5 noise tests

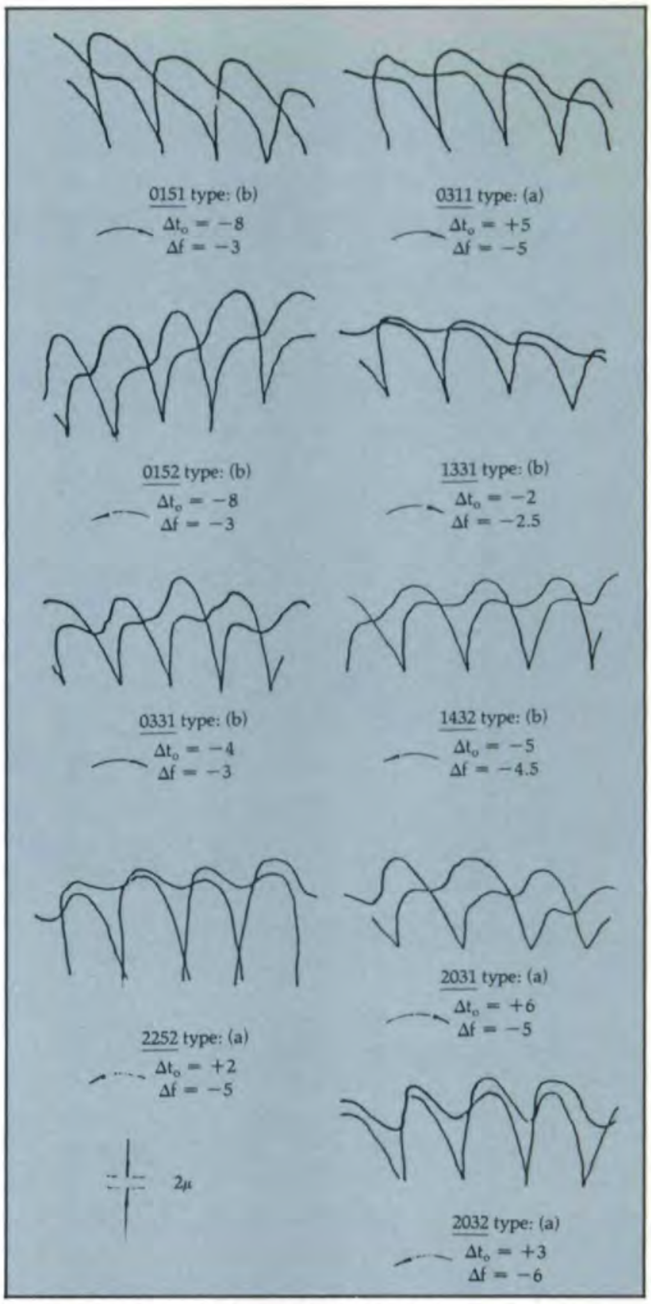


Fig. 10 – Tester's records of gears involved in the last five noise tests

80dB). Nos. 223-242 are the most noisy ones, over 84dB. The conclusion is that for medium-module, medium-speed, light-loaded spur gears, the main errors influencing noise are the normal pitch difference and the profile concavity. Positive Δt_o (normal pitch of driving gear larger than that of driven gear) offers "leaving impact" change-over (Table 1a), which is apparently more favorable than negative Δt_o (approaching impact change-over, Table 1b). From the average value of the extremities, increasing rate of noise could be roughly estimated as every increment of 0.66 microns in negative normal pitch difference (normal pitch of driven gear minus that of driving gear), plus a composite concavity increment of 0.62 microns will cause an increment of 1db in gear noise. The tester's records of the gears involved in first five noise tests are shown in Fig. 9 (1641, 2161, 2251, 0941, 3041), while the tester's records of gears involved in last five noise tests (0151, 0152, 0311, 0331, 1331, 1432, 2031, 2032, 2252) are shown in Fig. 10. Fig. 11 shows gears ground on Chinese worm wheel

Table — 3 An index to find tester's record with known number of gear

Number of gear	0151	0152	0311	0331	0941	702L	703L	706L	709L	710L
Number of Fig.	10	10	10	10	09	13	11	12	13	11
	1432	1641	2031	2032	2161	2251	2252	3041	711L	1331
	10	09	10	10	09	09	10	09	12	10

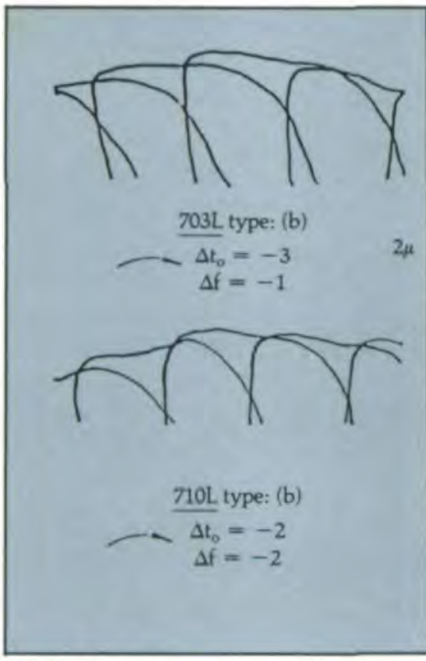


Fig. 11 — Ground on Chinese worm wheel gear grinder 7232A

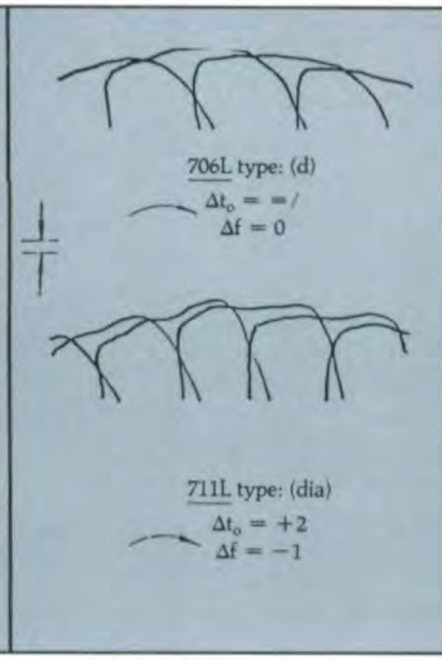


Fig. 12 — Ground on Swiss worm wheel gear grinder NZA

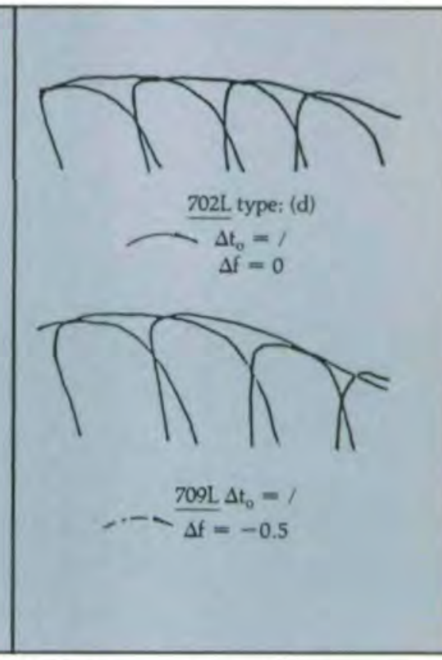


Fig. 13 — Ground on Chinese worm wheel gear grinder 7232A plus final free grinding

gear grinder 7232A (703L 710L). Fig. 12 shows gears ground on Swiss gear grinder NZA (706L 711L). Fig. 13 shows gears ground on Chinese gear grinder 7232A with final free grinding. (702L, 709L). Table 3 is an index to find the tester's record of known gear number.

Comparison of noise level of gears ground on different gear grinders. Experiments were done mainly on worm abrasive wheel gear grinders because they are productive, and because they show promise for manufacturing silent gears. Table 4 shows the noise level measured at 300mm from the pitch point of the meshing gears. From these test results, one sees that with the addition of a final worm wheel free grinding, the noise level was dropped from an average of 80.7dB to 76.7dB. Fig. 14 shows the tester's records from Swiss grinder AZA. The machine tool is accurate, but because the wrong diamond dressing wheel was used the profile thus obtained is a convex form.

Some gear makers think that a convex profile may offer silent transmission, perhaps because most gear making methods produce a concave profile and typically noisy gears. Therefore, it is understandable to suppose that a convex profile will offer silent transmission. However, we have shown experimentally that this is not the case. A batch of some 20 gears with profiles as shown in Fig. 15 were made. They turned out to be very noisy (over 85dB).

Experiment for investigating the phenomenon of losing contact in gear meshing. Gear noise researchers have long noticed the phenomenon of losing contact. We did some experiments on this phenomenon as well. The gear noise tester was isolated between driving and driven gears. A signal

Table 4 — Comparison of noise of gears ground on different gear grinders

NO.	Type of gear grinder	Way of profile generation and indexing	No. of noise test	Average noise	No. of Fig. of tester's record
1	Chinese horizontal gear grinder 7132A	steel belt and rolling disc with virtual rolling radius adjustable, indexing by change gear and worm wheel		83dB	(data offered by factory)
2	Chinese worm wheel gear grinder 7232A	Combine profile generation and indexing just like hobbing	28	80.7dB	Fig. 15
3	Swiss worm wheel gear grinder NZA	"	32	78.0dB	Fig. 16
4	Swiss worm wheel gear grinder AZA	"	4	81.5dB	Fig. 18
5	Chinese worm wheel free grinder	"	34	76.7dB	Fig. 17

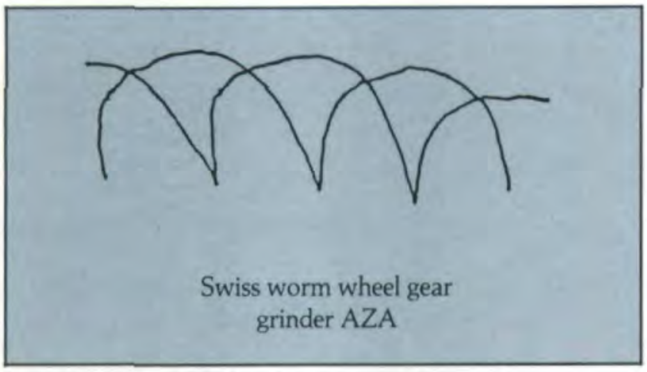


Fig. 14 — Tester's record of gear ground on AZA

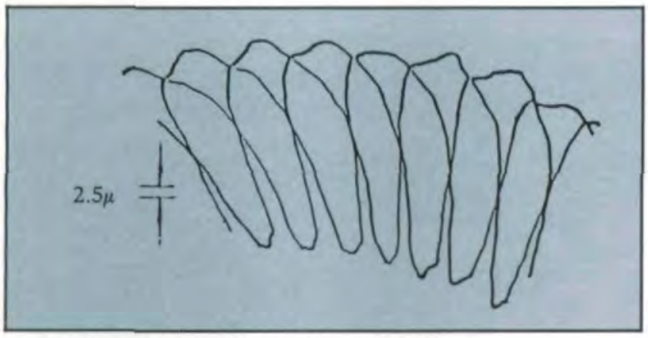


Fig. 15 — Tester's record of gear with convex profile

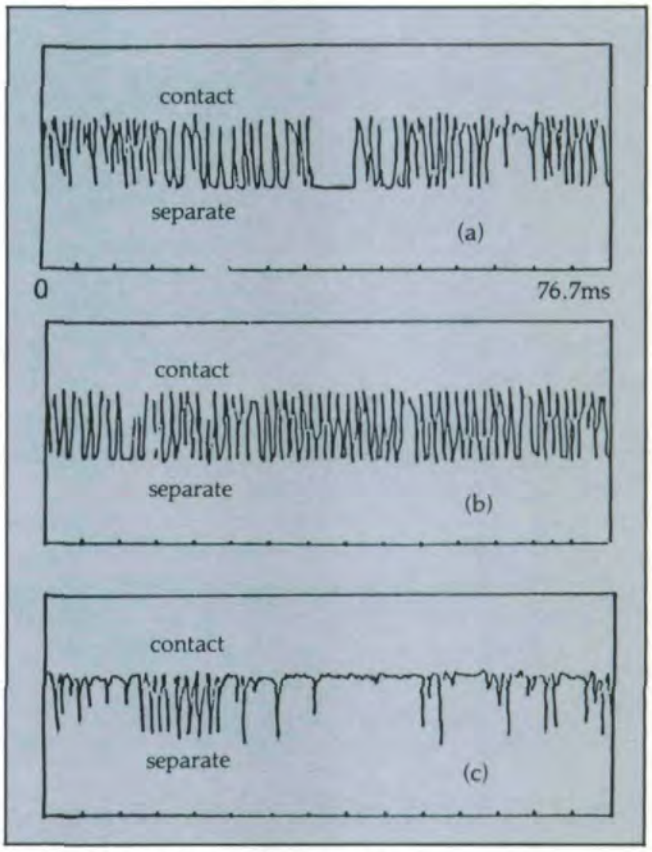


Fig. 16 — Time series curves of losing contact

voltage with some resistors in series was connected to the gears. If the gears were in contact, the signal voltage was short circuited to low voltage. If they lost contact, high voltage appeared. Our experiment not only confirmed the existance of the phenomenon, but also defined some of its characteristics. Three sorts of losing contact were observed. (See Fig. 16.)

a) Jump over teeth — The driving gear loses contact with the driven gear; then, after crossing several teeth, it comes again into contact.

b) Frequently lose contact for short instant within the period of one tooth, lose contact and come into contact again several times.

c) Jump over teeth and come to frequently losing contact alternatively.

Experiment for investigating the influence of contact ratio on noise level of gears. A pair of non-standard high precision gears were made with very high thin teeth, which could provide a contact ratio of more than two. By varying the center distance on the tester, a relation between the contact ratio and the noise level was found as shown in Fig. 17. Minimum noise seems to fall on a contact ratio a little bit less than 2 or 1. This indicates that even under the slight load, the influence of torsional stiffness is still of some importance.

A tentative suggestion on the optimized geometry for low noise gears. For the time being, we can suggest only the type (d) in Table 1. That is, doing best to reach better indexing (circular pitch), for the existing error, adapt pressure angle α to ensure correct normal pitch, so that on the tester's record the adjacent curves will meet very well, and a smooth change-over can be realized with slightest tip/root relief. Table 5 shows an example of silent gear pair.

Research on the Making of Silent Gears

From the previous sections, it is clear that gear noise is produced:

a) at the instant of change-over,
b) during the process of meshing of a pair of gear teeth.

The demand for accuracy is also directed at reducing noise, that is:

a) The normal pitch difference Δt_o as small as possible. If it is not possible to obtain this figure, a positive Δt_o is preferable.
b) The profile form error Δf should preferably be as close to the theoretical involute as possible with minimum relief needed.

From Table 5, it can be seen that to guarantee ideal silent gear transmission, the demand on key items of accuracy is remarkably high.

1) To ensure final accuracy, the key technology is in the finishing method after hardening.

2) To reach high accuracy in normal pitch, free finishing methods are very favorable. They can correct the normal pitch error of the gears with the accurate normal pitch of the tool (gear hone, worm-shaped, abrasive wheel, internal gear hone, etc. In Table 4, it has been shown that an additional free grinding to the worm wheel gear grinder cut down the average noise level by 4dB. Another factory in Dalian succeeded in cutting down the gear noise by some 5-6dB by internal honing. (The gear hone is with a shape of an internal gear.) Further theoretical analysis can prove mathematically that free finishing can guarantee the type of gear shown in Table 1d, while grinding with constrained motion of wheel and job will always lead to the geometry shown in Table 1c.

3) To obtain the ideal profile accuracy, a finishing method with controllable metal removing rate along the full profile (from tip to root) is keenly wished. The field controlled ECH of gears is one of the new technologies we are developing with this intention. (See Reference 3.)

Conclusion

As the introduction of this article has stated, all we are writing here is just an idea formed in our minds by the results of experiments. It can basically explain the phenomenon we can see now, but any disclosure by further experiments may make it necessary to revise the explanation. Besides, there is still a distance between experiment and application in production to cover. But so far as we can see now, a comparatively silent gear (say below 78dB) looks feasible. For further reducing noise, other measure such as damping and absorbing, and sound isolation, should be considered too.

Table — 5 Example of silent gear pair

Driving gear		Driven gear		
Gear No.	Fig. No.	Gear No.	Fig. No.	Noise level (average)
702L	Fig. 17	709L	Fig. 17	75.6dB
709L	Fig. 17	702L	Fig. 17	74.2dB

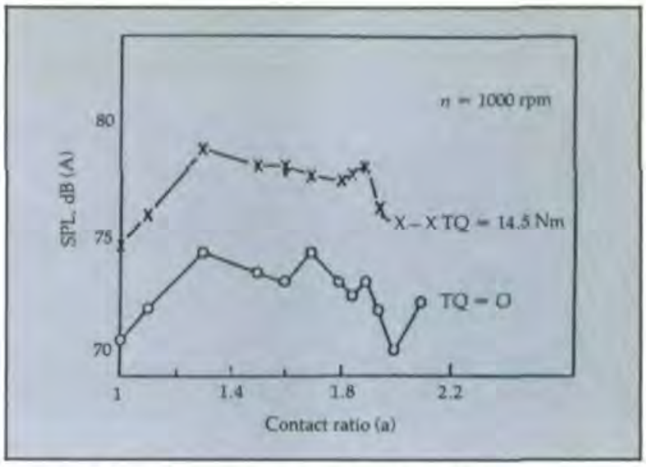


Fig. 17 – Relation between contact ratio and noise level

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Influence of Lubrication on Pitting and Micropitting Resistance of Gears

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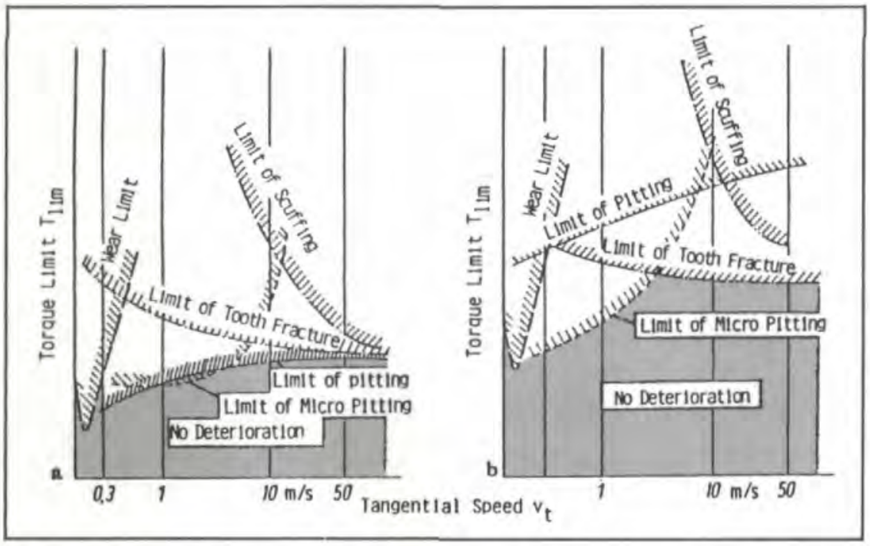

Fig. 1a – Main limits of the load capacity of through-hardened steel gears.

Fig. 1b – Main limits of the load capacity of case-carburized steel gears.

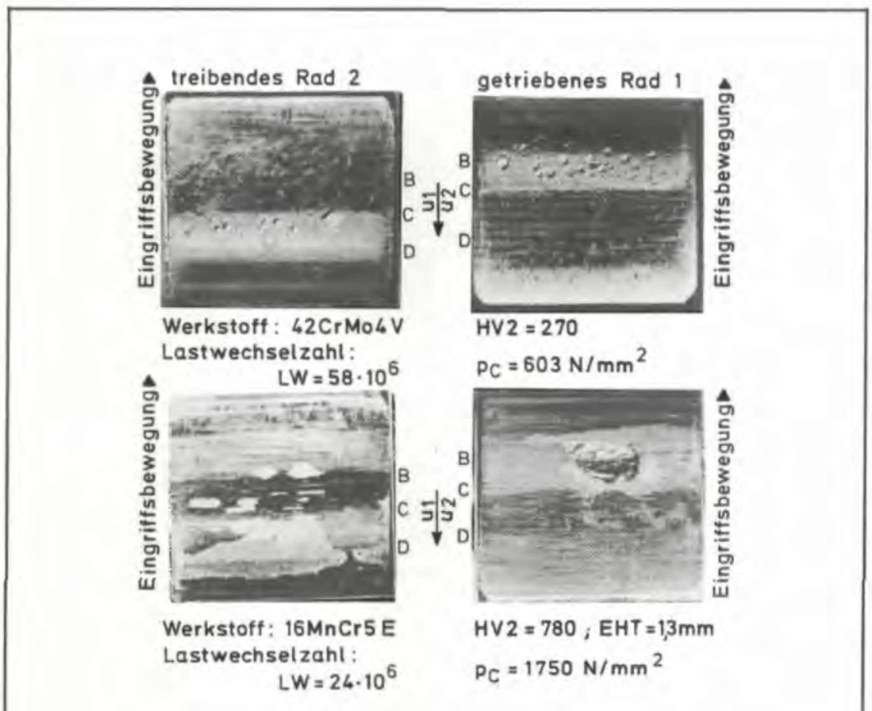

Fig. 2 – Pitting damaged tooth flanks. Application conditions: a = 200 mm; m = 9 mm; z_1/z_2 = 21/22; injection lubrication using mineral oil; v_t = 11.5 m/s.

Abstract:

Pitting and micropitting resistance of case-carburized gears depends on lubricants and lubrication conditions. Pitting is a form of fatigue damage. On this account a short time test was developed. The test procedure is described. The "pitting test" was developed as a short time test to examine the influence of lubricants on micropitting. Test results showing the influence of case-carburized gears on pitting and micropitting are presented.

Introduction

Pitting and micropitting are essential failure limits of the surface strength of gears, which are not only affected by material, but also by lubricants and lubricating conditions.

Fig. 1 shows the increasing torque limit of pitting and micropitting as a function of increasing tangential speed. The tooth fracture limit may be raised by increasing the module, so that the ruling factor is the pitting limit, even for case-carburized gears. (See Fig. 1b.) Therefore, the surface strength has to be determined by a life time test with a gear test rig. Because life time tests are very expensive, other test procedures for pitting and micropitting load capacity are worked out.[4-8]

Problems of Surface Strength

Fig. 2 shows typical pitting damage of two gear pairs. The damage appears below the pitch circle at pinion and wheel.

Pitting causes the failure of the gear pair within the testing procedure. In general, micropitting occurs during surface strength tests of hardened gears. Operative conditions (especially pressure, tangential speed, and temperature), tooth geometry (especially curvature and sliding conditions), quality of the surface, material, and lubricant features determine stress and deterioration of the gear pair. For the most part, case-carburized gears show serious pitting damage only at a few flanks, which limit the life of the gear pair. The pitting damage varies over the gear periphery.

Because failure shows fatigue characteristics, repeated tests under similar conditions show gear lives scattered in a wide range.

The method of calculation in ISO 6336/2[3] covers the influence of the lubricant only by the lubricant factor Z_L, which takes the nominal viscosity of the lubricant and the strength of material into account. (See Fig. 3.)

The range of uncertainty is very large because varying additives of the lubricant cannot be taken into account.

Fundamental rolling tests using through-hardened steel disks have systematically examined the influence of friction coefficient and lubricant film (referred to the surface roughness) on pitting resistance. (See Fig. 4.[10]) The transfer of absolute strength values from these tests to the gear pair is uncertain, because conditions differ.

The Determination of Flank Load Capacity Using FZG Back-To-Back Test Rigs

Systematic, statistically proven tests simulating real-life conditions should only be performed on a cost-effective basis, depending on the reliability of the gears required.

Back-to-back test rigs are normally used, and Wöhler curves are determined by running tests. Fig. 5 shows the test rig and testing conditions. By varying gears and conditions, many applications can be closely simulated.

Failure limit of pitting depends on application conditions and characteristic features of the gear. Fig. 6 shows the deterioration development during a fatigue test as a function of flank pressure and load cycles.

Failure criteria of case-carburized gears are defined as 4% pitted area

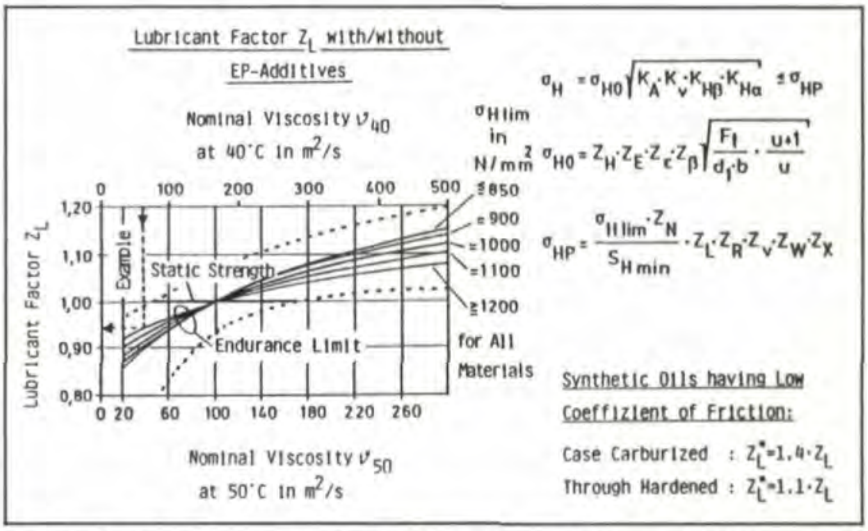

Fig. 3 – Influence of lubricants on pitting load capacity – calculation method reference ISO 6336/2.[1,3]

Surface Strength of Through-Hardened Disks (Material: 42CrMo4)

Following Factors Influencing the Life Time are Tested Separately:
– p_H Hertzian Pressure; – μ Coefficient of Friction; – ϑ_{B1} Flash Temperature;
– v_2 Tangential Speed; – La Lubricant-Film-Thickness / Roughness.

The following equation is the result of these investigations:

$$\frac{LW_1}{LW_2} = \left(\frac{p_{H1}}{p_{H2}}\right)^{-7} \cdot \left(\frac{\mu 1}{\mu 2}\right)^{-3} \cdot \left(\frac{\vartheta_{Bl\,1}}{\vartheta_{Bl\,2}}\right)^{-2} \cdot \left(\frac{v_{2\,1}}{v_{2\,2}}\right)^{1.2} \cdot \left(\frac{La_1}{La_2}\right)^{2.2}$$

If $La_{1,2} > 1.0 \rightarrow La_{1,2} = 1.0$.

Surface Strength of Through Hardened Gears (Material: 42CrMo4)

The dominant influence of the coefficient of friction, apart from the influence of the pressure, is also proven at Gears.

Coefficient of Friction at EHL*-Conditions

Mineral Oil : $\mu \sim p_H^{0.6}$ $\mu \sim v^{-0.25}$ $\mu \sim Ra^{0.2}$;

Synthetic Oil: The coefficient has to be evaluated by tests.

*EHL : Elasto-Hydrodynamic Lubrication

Fig. 4 – Effects of EHL-parameters on pitting load capacity – summary.

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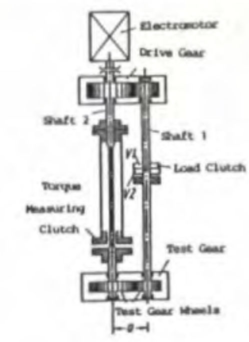
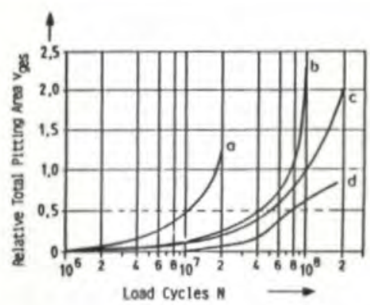


FZG Back-To-Back Test Rig:

Standard Test Gear Wheel and Testing Conditions:

a = 91.5; 112.5; 140; 200mm;
v_t = 8...16 m/s;
Mineral Oil: ν_{50} = 50...100 mm^2/s;
Spur Gears : b = 10...30 mm;
m = 2.25...8 mm; Z_1/Z_2 = 37/45...12/18;
Balanced Sliding (Gear Type: C);
DIN-Quality 4...6, Grinded Flanks;
Wöhler Curves Basing upon 7...25 Test Points.

Fig. 5 – Determination of pitting load capacity using FZG test rigs.[6]



ref. to FZG/Rettig

Evaluation of a Wöhler Curve

Material : 15CrNi6 ;
Module : 5.5 mm ;

a. $p_C = 1.20 \cdot p_{CD}$;
b. $p_C = 1.02 \cdot p_{CD}$;
c. $p_C = p_{CD}$;
d. $p_C = 0.92 \cdot p_{CD}$;

p_C : Flank Pressure at the Pitch Point;

p_{CD} : p_C Endurance Limit Occurs.

Fig. 6 – Deterioration development as a function of flank pressure and number of load cycles.

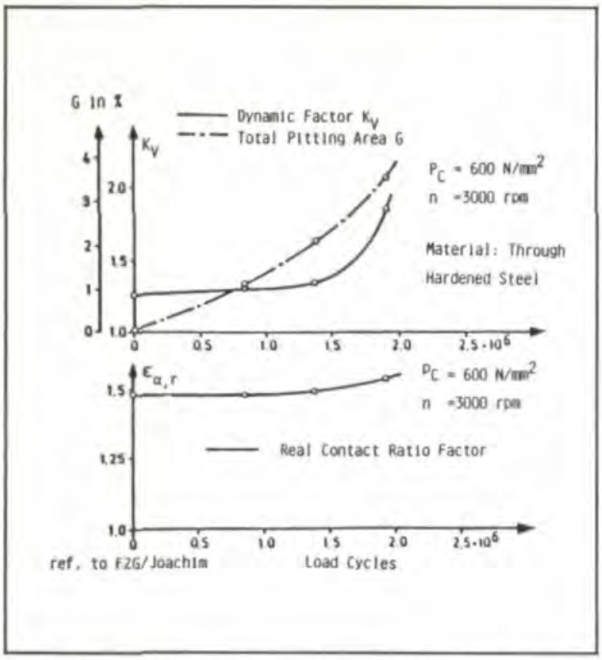


Fig. 7 – Dynamic factor, total pitting area, and real contact ratio factor as functions of load cycles.

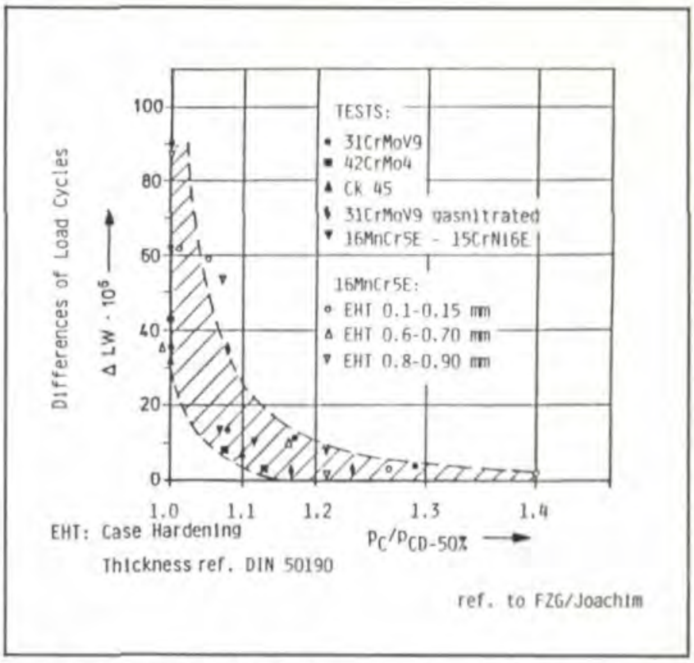


Fig. 9 – Differences of load cycles between maximum and minimum running numbers.

on one flank, or as the surface of all pitted areas exceeds 1% of the active flank area. When reversals reach $5 \cdot 10^7$ or 10^8 cycles, endurance range will be assessed.

Experimental investigations show that dynamic tooth forces progressively increase, exceeding the limited area. (See Fig. 7.)

Because gear life times until pitting occurs vary greatly, a large number of tests are required to determine fatigue strength and endurance limit. (See Fig. 8.) Then a strength value with a certain probability (e.g., Weibull-or Probit-probability curve) can be in-

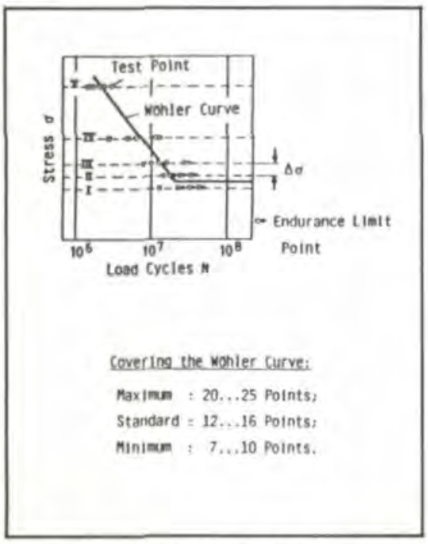


Fig. 8 – Determination of pitting load capacity by using a Wöhler curve.

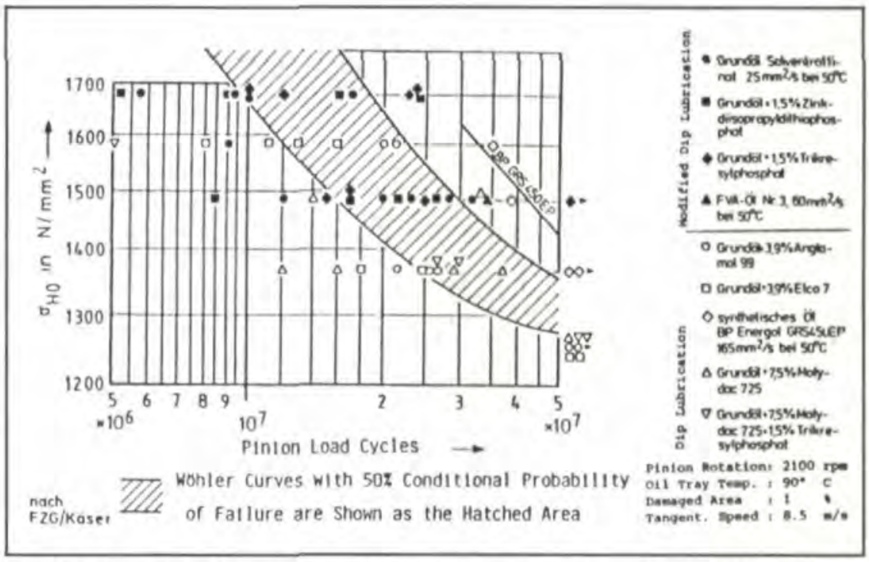


Fig. 10 – Influence of additives on pitting load capacity of case-carburized gears (16MnCr5E).

FZG Back-To-Back Test Rig ref. DIN 51354:

Case Carburized and Ground Test Wheels, Gear Type: C, Dip Lubrication with Constant Oil Temperature: 90 C, Tangential Speed: v_t = 8.3m/s, Driving Pinion; Defined Running In, Repeated One Load Stage Test (p_c = 1654 N/mm²) until Reaching Failure Limit;

Failure Criterion: 4% Pitting on One Flank of the Pinion or 1% Pitting on Total Active Area of the Pinion.

Fig. 11 – Test conditions of the FZG pitting test.

dicated. One gear pair is necessary to achieve two test points, and one test point requires about 500 test rig hours to reach the endurance limit. The execution of a S-N curve is very time-consuming. To prove the fatigue stress statistically at the range of limited life fatigue, shorter running times are possible.

Fig. 9 shows the difference of load cycles between the area of limited life fatigue strength and endurance limit at pitting tests. It is evident that even at the low range of limited life fatigue ($p_C/p_{CD} \approx 1.1$), a distinct contraction of the running time differences is to be expected.

Fig. 10 shows S-N curves which examine the influence of additives on pitting resistance of case-carburized gears. Only a limited statistical proof is possible.

The "Pitting Test" as a Short Time Test for Pitting Resistance

Based on results of the influence of additives on pitting resistance, the Forschungsvereinigung Antriebstechnik e.V., Frankfurt, (FVA) has carried out a round robin test.

The short time "pitting test" was defined and executed by five laboratories. The test conditions are shown in Fig. 11.

FZG gear test rigs having similar geometry (center distance: a = 91.5mm) to the standardized FZG scuffing test rig (DIN 51354) were used.[2]

Gears (case-carburized 16MnCr5, Maag 0° grinding) and lubricant from one charge were placed at disposal. The results of all test rigs concerning two tested lubricants are shown in Fig. 12.

The FZG has taken part in this round robin test. Fig. 13 shows the share of the FZG. Although the load cycles widely scatter, a distinct differentiation between the two lubricants having different additives is possible.

The relative scatter (incline of the Weibull curve) of both test series is almost corresponding. If the FZG test is evaluated sequentially for the use of the test gear pairs, one can see that, despite the increasing number of running tests, the statistical mean total endurance LW_{50} hardly differs.

It would need about five or six test runs to achieve a correct value if the tests are executed carefully. (See Fig. 14.)

In the course of time the FZG has carried out further pitting tests using different gears (equal geometry, different material).

Quality of manufacture and heat treatment of these gears were identical to those used at the round robin test. Reduced load cycles have been the results of these tests. This is proven by a few comparison tests shown in Fig. 15. Especially in pitting tests, the constancy of test gear features is very important. Besides the difference of endurance, the two lubricants show a cardinal difference in the appearance of the flank surface. The flanks have a general blank look using FVA oil doped with A99, while using FVA oil doped with ZDP effects strong micropitting and heavier flank wear. (See Figs. 16-17.)

Pitting and micropitting affect each other. Depending on intensity, micropitting results in continuously increasing crater wear at the root of the flanks and in reduced pitting resistance.

Micropitting and Micropitting Resistance

Description of damage: To the naked eye, areas where micropitting has occured appear gray. (See Fig. 18.) Micropitting usually starts at the root of the pinion and spreads over the whole flank. It rarely begins at the teeth tip of the wheel.[7,9]

There is continuous loss of tooth surface material, which results in craters similar to wear craters and deteriorates the profile. Scanning electron microscope investigations prove that the failure shows fatigue symptoms. There are a lot of broken out, partly conchate particles (micropitting!) which give the gray look to the flanks.

Operative conditions: Fig. 19 shows the main conditions of the micropitting resistance comprehensively. All factors affecting the lubricating film between the flanks affect the tendency to micropitting.[6]

By use of a "thick" EHL film and "smooth" surfaces, micropitting can be eluded. Lubricants doped with different additives react to test conditions individually. Unlike ZDP additives, sulfurphosphorus (S-P) additives are advantegeous to micropitting resistance.

The Micropitting Test: A short time test to determine micropitting resistance. The micropitting test was developed by the FZG implementing a FVA

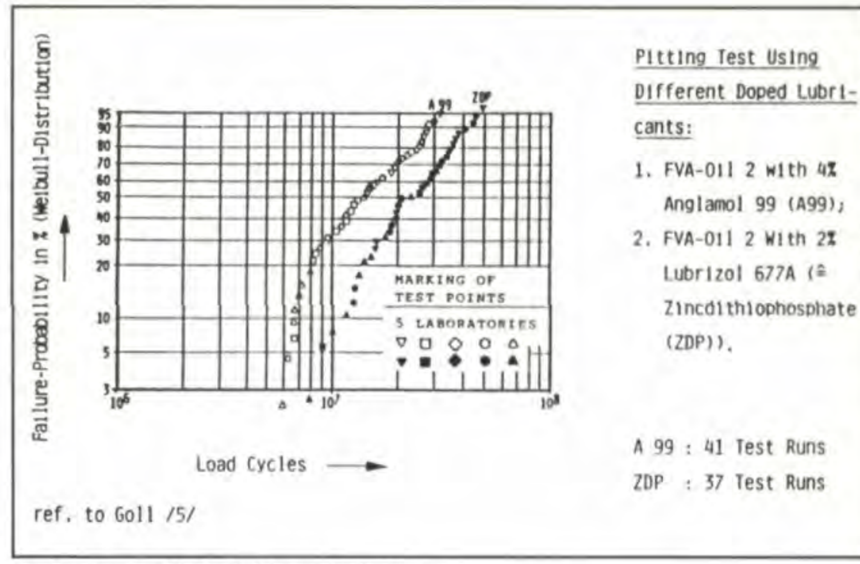
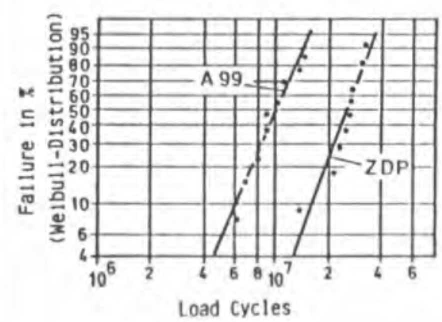


Fig. 12 – Results of the first FVA round robin test.



Lubricant	LW_{50}
FVA2+A99	$10.4 \cdot 10^6$
FVA2+ZDP	$26.0 \cdot 10^6$

Numbers of Load Cycles until Reaching the Failure Limit in Millions:

A 99	ZDP
6.3	14.0
6.9	21.6
8.1	23.1
8.3	25.1
9.0	26.4
9.1	26.9
10.3	27.5
11.1	30.0
11.2	31.2
13.9	32.4
14.7	
14.8	

Fig. 13 – Results of the FZG pitting test.

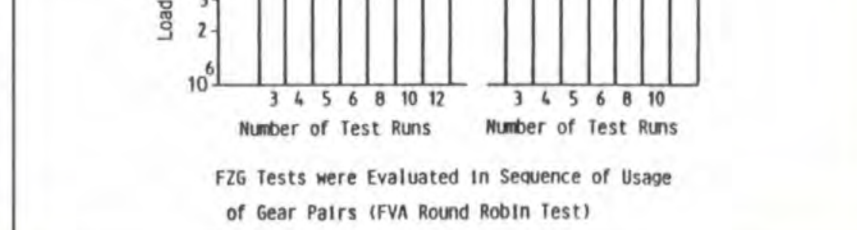


Fig. 14 – Influence of the number of test runs.

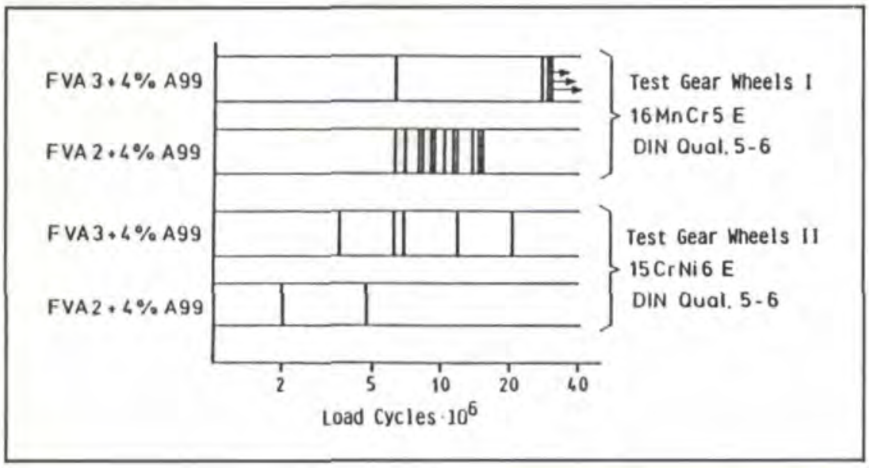


Fig. 15 – Influence of test gear wheels on pitting tests.

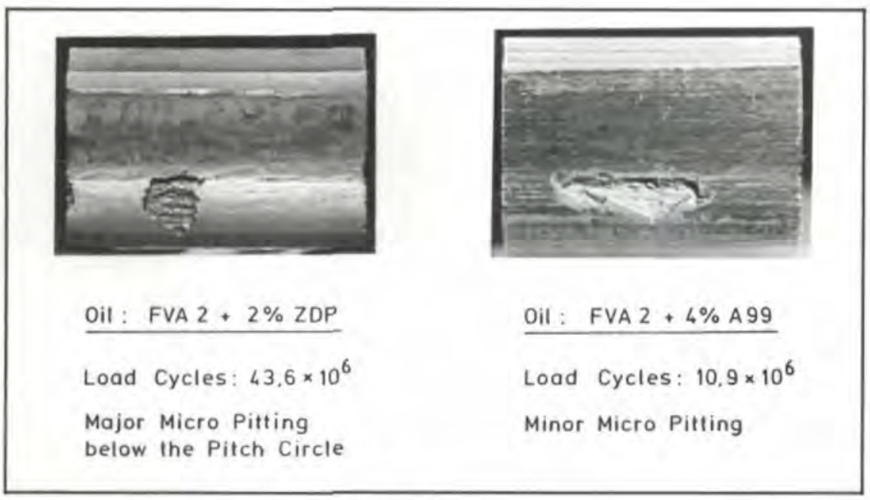


Fig. 16 – Examples of gear teeth used on pitting tests.

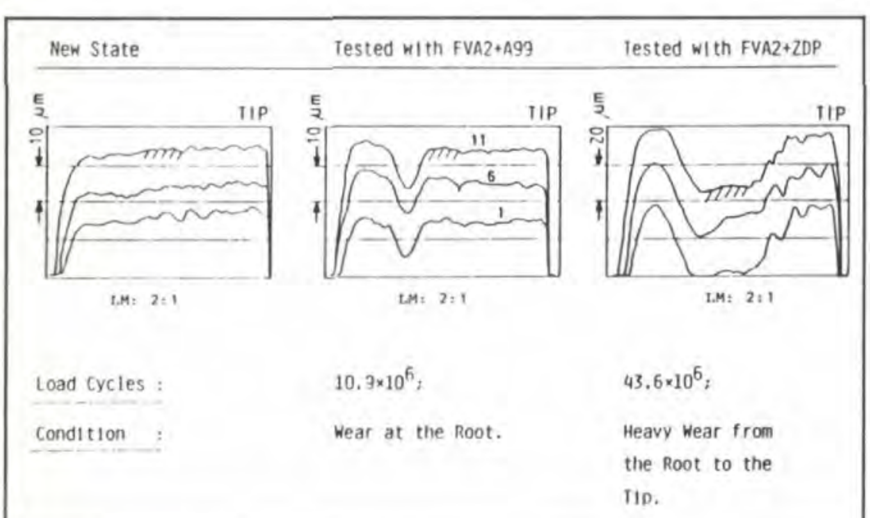


Fig. 17 – Tooth profiles of test gears.

project. To test the influence of lubricants, a modified FZG test (DIN 51354) and a gear with balanced sliding at the tip of the pinion and of the wheel are used.[7,8,9] The load is raised by load stages until the amount of the profile error exceeds twice the amount of error at the beginning. To get reliable judgment and information about deterioration development (partly degressive) at longer running times, a permanent test run (10 . . . $50 \cdot 10^6$ load cycles) is executed afterwards.

Results of micropitting test runs: Fig. 20 shows results of a micropitting test using three lubricants doped with different additives.

Obviously there is a relation between loss of weight and the failure causing increased amount of profile error. It is remarkable that the lubricants' GFT-low and GFT-high base on the same oil. The lubricant GFT-low is doped with ZDP additive, and GFT-high with a S-P additive. Thus, the strong influence of additives on micropitting is obvious. In certain cases the torque limit of micropitting is in contradiction to the pitting limit.

Summary and Outlook

Pitting and micropitting resistance of case-carburized gears depends on lubricants and lubricating conditions. Pitting is a form of fatigue damage. Therefore, pitting endurance has to be determined by running tests, and the results shown as S-N curves. Because these tests need statistical evaluations, they are expensive, therefore, a short time test was developed. The test procedure is described. The pitting test determines the influence of lubricants doped with different additives on pitting resistance.

Test runs have shown that micropitting often occurs during pitting tests of case-carburized gears. The damage micropitting is described. The micropitting test was developed as a short time test to prove the influence of lubricants doped with different additives on micropitting resistance of case-carburized gears.

Lubricants doped with S-P additives show a higher micropitting resistance than lubricants doped with ZDP additives.

Because qualitative results of both test procedures are possible only to the respective damage of this test, the FZG aims at a combination of both procedures considering economic aspects.

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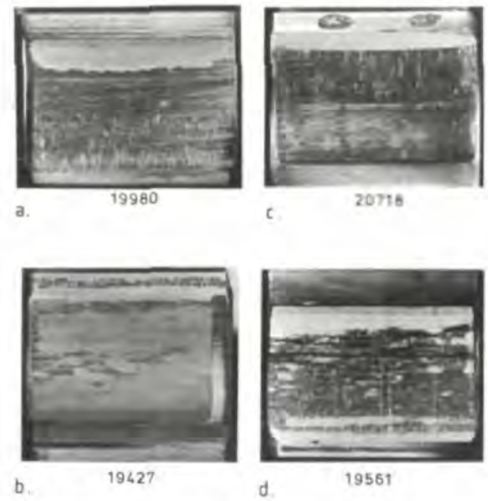

a. At the Root of the Driving Pinion;
b. Totally Damaged Flank;
c. At the Tip of the Driven Wheel;
d. Partly Damaged Flank.

Fig. 18 – Examples of gear teeth showing micropitting.

Influencing Variable	Range
Flank Roughness: Reduction from $6\mu m$ to $3\mu m$	1:3
Material, Heat Treatment (Advantageous Percentage of Austenite)	1:2.8
Additives (Equal Viscosity of Base Oil)	1:2
Doubled Working Viscosity	1:2
Halved Coefficient of Friction	1:1.7
Tangential Speed (Advantageous: High Speed)	1:1.3
Lowering Oil Temperature ($\Delta\vartheta = 20K$)	1:1.3

Fig. 19 – Estimated range of the influence of conditions on micropitting load capacity.

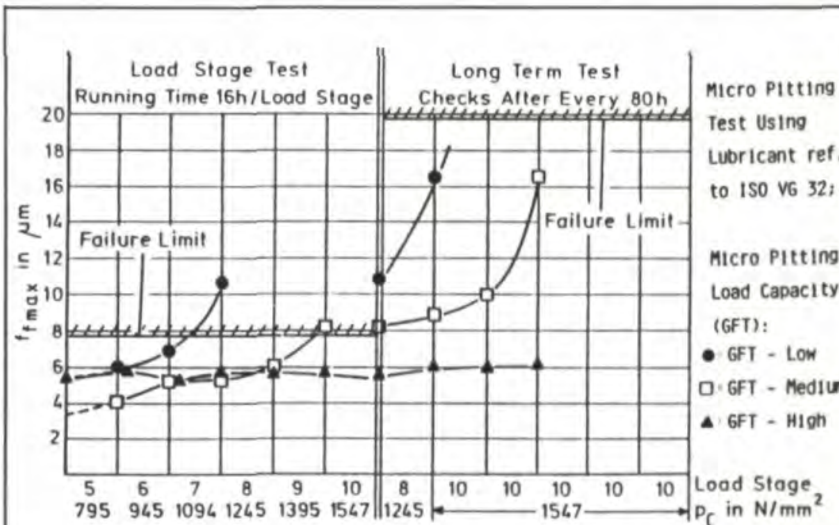

Fig. 20 – Increase of profile error at micropitting tests with different oils.

Dynamic Loads in Parallel Shaft Transmissions Part 1

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Introduction

Recently, there has been increased interest in the dynamic effects in gear systems. This interest is stimulated by demands for stronger, higher speed, improved performance, and longer-lived systems. This in turn has stimulated numerous research efforts directed toward understanding gear dynamic phenomena. However, many aspects of gear dynamics are still not satisfactorily understood.

For example, in industrial settings, a high performance gear system is often obtained by overdesigning and by sacrificing costs, materials, and compactness. In aerospace and military application where weight is a premium, gear systems are often designed under conditions very close to the failure limits, thereby introducting uncertainties in performance and life prediction. They are often prematurely replaced to prevent in-service failure. Moreover, gear systems are often designed by using static analyses. However, when gear systems operate at high speed, there are several factors which affect their performance. These include shaft torsional stiffness, gear tooth loading and deformation, gear tooth spacing and profile errors, rotating speeds, mounting alignment, dynamic balance of rotating elements, gear and shaft masses and inertia, and the masses and inertias of the driving (power) and driven (load) elements.

There is no agreement among researchers on the best methods for evaluating dynamic load effects. Hence, gear designers are often confronted with conflicting theories. They generally have to rely on past experience, service safety factors, and experimental data with a limited range of applicability.

The objective of this report is to provide more insight into the factors affecting dynamic loads.

Research efforts on gear system dynamics have been conducted for many years. In 1892 Lewis(1) recognized that the instantaneous tooth load was affected by the velocity of the system. In 1925, Earle Buckingham(2) headed an experimental research effort, endorsed by ASME, to measure dynamic effects. A report published in 1931 represented the first authoritative document on gear dynamics. It presented a procedure for determining the so-called dynamic load increment due to mesh dynamics and gear tooth errors.

In 1959, Attia(3) performed an experiment to determine actual instantaneous loading. He found that Buckingham's results were conservative.

In 1958, Niemann and Rettig(4) found that larger masses caused higher dynamic loads, but as the average load became larger, the effect of larger masses became less important. They also found that very heavily loaded gear systems showed no appreciable dynamic load increment, whereas in lightly and moderately loaded gear systems, there were considerable dynamic load increments. In 1958, Harris(5) suggested that for gear systems isolated from external stimuli, there are three internal sources of dynamic loads:

(1) Error in the velocity ratio measured under the working load.

(2) Parametric excitation due to stiffness variation of the gear teeth.

(3) Nonlinearity of tooth stiffness when contact is lost.

In 1970, Houser and Seireg(6) developed a generalized dynamic factor formula for spur and helical gears operating away from system resonances. The formula took into consideration the gear geometry and manufacturing parameters as well as the dynamic characteristics of the system.

In 1972, Ichimaru and Hirano(7)analyzed heavy-loaded spur gear systems with manufacturing errors under different operating conditions. They found that the change in tooth profile showed a characteristic trend to decrease dynamic load. In 1978, Cornell and Westervelt(8) presented a closed form solution for a dynamic model of a spur gear system and showed that tooth profile modification, system inertia and damping, and system critical speeds can have significant effects upon the dynamic loads. In 1981, Kasuba and Evans(9) presented a large scale digitized extended gear modeling procedure to analyze spur gear systems for both static and dynamic conditions. Their results indicated that gear mesh stiffness is probably the key element in the analysis of gear train dynamics. They showed that the gears and the adjacent drive and load systems can be designed for optimum performance in terms of minimum allowable dynamic loads for a wide range of operating speeds.

In 1981, Wang and Cheng(10) developed another dynamic load response algorithm. They reported that the dynamic load is highly dependent on the operating speed. This model is later modified by Lewicki(11) to account for the nonlinear Hertzian deformation of meshing gear teeth. The gear dynamic load found from the revised model showed little difference from the original model, since the Hertzian deflection was relatively small in comparison with the total gear tooth deflection. Nagaya and Uematsu(12) stated that because the contact point moves along the involute profile, the dynamic response should be considered as a function of both the position and speed of the moving load. In 1982, Terauchi, et al.,(13) studied the effect of tooth profile modifications on the dynamic load of spur gear systems. According to their results, the dynamic load decreased with proper profile modifications.

In this first part of the article, we present a model of a parallel shaft transmission. We consider the effects of shaft stiffness and inertia, load and power source inertias, tooth stiffness, local compliance due to contact stresses, load sharing, and friction. A parameter study is provided in the second part of the article.

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Modeling

Fig. 1 depicts a model of the transmission. It consists of a motor or power source connected by a flexible shaft to the gear system. The gear system consists of a pair of involute spur gears. They are connected to the load by a second flexible shaft as shown. Symbolically, the model may be represented by a collection of masses, springs, and dampers as in Fig. 2.

Let θ_M, θ_1, θ_2, and θ_L represent the rotations of the motor, the gears, and the load. Then by using standard procedures of analysis, the governing differential equations for the rotations may be written as

$$J_M\ddot{\theta}_M + C_{s1}(\dot{\theta}_M - \dot{\theta}_1) + K_{s1}(\theta_M - \theta_1) = T_M \qquad (1)$$

$$J_1\ddot{\theta}_1 + C_{s1}(\dot{\theta}_1 - \dot{\theta}_M) + K_{s1}(\theta_1 - \theta_M) + C_g(t)[R_{b1}\dot{\theta}_1 - R_{b2}\dot{\theta}_2] + K_g(t)[R_{b1}(R_{b1}\theta_1 - R_{b2}\theta_2)] = T_{f1}(t) \qquad (2)$$

$$J_2\ddot{\theta}_2 + C_{s2}(\dot{\theta}_2 - \dot{\theta}_1) + K_{s2}(\theta_2 - \theta_1) + C_g(t)[R_{b2}\dot{\theta}_2 - R_{b1}\dot{\theta}_1] + K_g(t)[R_{b2}(R_{b2}\theta_2 - R_{b1}\theta_1)] = T_{f2}(t) \qquad (3)$$

$$J_L\ddot{\theta}_L + C_{s2}(\dot{\theta}_L - \dot{\theta}_2) + K_{s2}(\theta_L - \theta_2) = -T_L \qquad (4)$$

where J_M, J_1, J_2, and J_L represent the mass moments of inertia of the motor, the gears, and the load; C_{s1}, C_{s2}, and $C_g(t)$ are damping coefficients of the shafts and the gears; K_{s1}, K_{s2}, and $K_g(t)$ are stiffnesses of the shafts and the gears; T_M, T_L, $T_{f1}(t)$, and $T_{f2}(t)$ are motor and load torques and frictional torques on the gears; R_{b1} and R_{b2} are base circle radii of the gears; t is time; and the dots over θ indicate time differentiation.

In developing Equations 1 to 4 several simplifying assumptions are employed.

(1) The dynamic process is studied in the rotating plane of the gears. Out-of-plane twisting and misalignment are neglected.

(2) Damping due to lubrication of the gears and shafts is expressed in terms of constant damping factors.

(3) The differential equations of motion are developed by using the theoretical line of action.

(4) Low contact ratio gears are used in the analysis. Specifically, the contact ratio is taken between 1 and 2.

Analysis

A major task in the analysis is to determine the values of the stiffness, damping, and friction coefficients appearing in Equations 1 to 4. Another task is to determine the ratio of load sharing between the teeth during a mesh cycle. These factors depend on the roll angle of the gears. Thus, Equations 1 to 4 are made nonlinear by these terms.

STIFFNESS

Gear stiffness. – Consider first the stiffness coefficients K_g, K_{s1}, and K_{s2}. Let the tooth surface have the form of an involute curve. Let W_j be the transmitted load at a typical point j of the tooth profile. Let q_j be the deformation of the tooth at point j in the direction of W_j. Then the gear stiffness K_{gj} for the gear teeth in contact at j is

$$K_{gj} = \frac{W_j}{q_j} \qquad (5)$$

In general q_j will depend on the following: (1) the bending of the tooth, the shear deformation of the tooth, and the axial compression of the tooth; (2) the deflection due to the flexibility of the tooth foundation and fillet; and (3) the local compliance due to the contact stresses.

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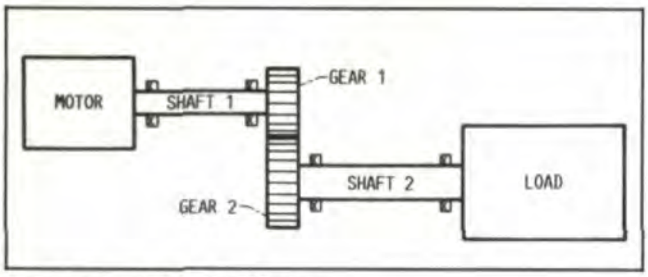

Fig. 1 – A Simple Spur Gear System.

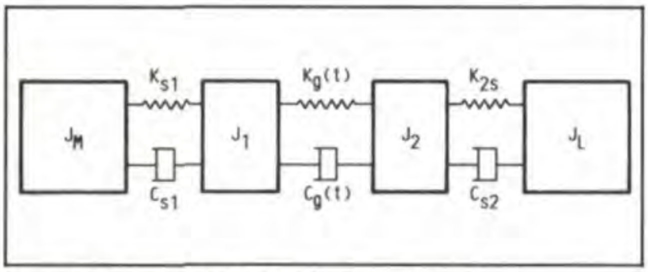

Fig. 2 – A Mathematical Model of the Transmission System.

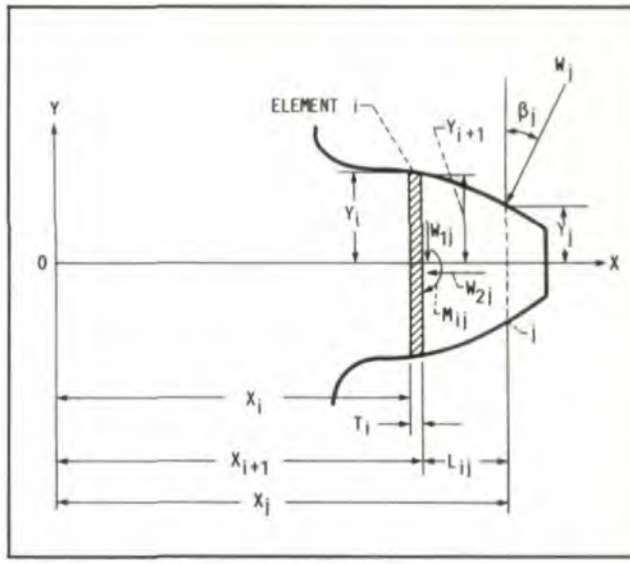

Fig. 3 – Element Modelling of a Gear Tooth.

To determine q_j let the tooth be divided into elements as shown in Fig. 3. Let i be a typical element with thickness T_i, cross section area A_i, and second moment of inertia I_i. Let L_{ij} be the distance between element i and point j along the x-axis. Let β_j be the angle between W_j and the y-axis. (See Fig. 3.)

Consider the tooth to be a nonuniform cantilever beam. Let q_{bj} be the contribution to q_j by the bending, shear, and axial deformation of the tooth. The q_{bj} may be represented as the sum of the deformation in the elements i beneath point j. That is,

$$q_{bj} = \sum_{i=1}^{n} q_{bij} \qquad (6)$$

where n is the number of elements beneath j and q_{bij} is the deformation of element i due to the load W_j.

(continued on page 30)

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DYNAMIC LOADS . . .
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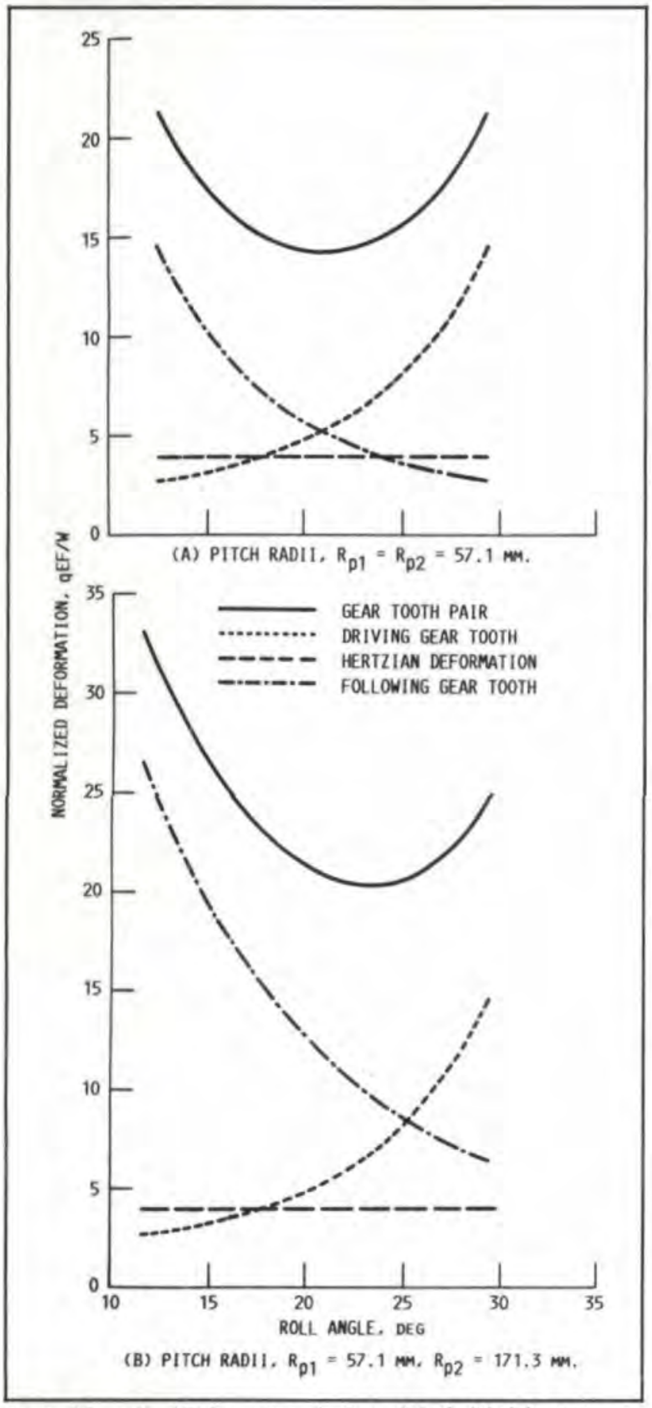

Fig. 4—Normalized Deformation of a Pair of Teeth. Module, 3.18 mm; Pressure Angle, 20°; Face Width, F = 25.4 mm; Modulus of Elasticity, E = 207 GPa; Applied Load, W = 105 KN/m.

With standard analysis q_{bij} is found to have the value:[8, 14, 15]

$$q_{bij} = (W_j/E_e)\Big\{\cos^2\beta_j\Big[\Big(T_i^3 + 3T_i^2L_{ij} + 3T_iL_{ij}^2\Big)\Big/3I_i\Big] - \cos\beta_j\sin\beta_j\Big[\Big(T_i^2Y_j + 2T_iY_jL_{ij}\Big)\Big/2I_i\Big] + \cos^2\beta_j[12(1+\nu)T_i/5A_i] + \sin^2\beta_j(T_i/A_i)\Big\} \qquad (7)$$

where Y_i is the half-tooth thickness at element i (See Fig 3.), ν is Poisson's ratio, and E_e is the "effective elastic modulus" depending upon whether the tooth is wide (plane strain) or narrow (plane stress). Specifically, for a "wide" tooth, where the ratio of the width to thickness at the pitch point exceeds 5,[14] E_e is

$$E_e = \frac{E}{(1-\nu^2)} \qquad (8)$$

where E is Young's modulus of elasticity. For a "narrow" tooth (width-to-thickness ratio less than 5), E_e is

$$E_e = E \qquad (9)$$

Expressions similar to Equation 7 hold for q_{fj}, the contribution to q_j for the deformation due to the flexibility of the tooth fillet and foundation.[15]

Let q_{cj} be the contribution to q_j from the local compliance due to contact stresses. With the procedures of Lundberg and Palmgren[16] q_{cj} may be expressed as

$$q_{cj} = \frac{1.275}{E^{0.9}F^{0.8}W_j^{0.1}} \qquad (10)$$

where F is the width of the tooth.

Hence, by superposition, the deformation at j in the direction of W_j is

$$q_j = q_{bj} + q_{fj} + q_{cj} \qquad (11)$$

The above expressions were used to calculate the deformations for two different gear pairs. The results are shown graphically in Figs. 4a and b.

Shaft stiffness. — The shaft stiffness K_s is given by the standard expression

$$K_s = \frac{JG}{l} \qquad (12)$$

where G is the shear modulus, l is the shaft length, and J is the polar moment of area given by

$$J = \frac{\pi D^4}{32} \qquad (13)$$

where D is the shaft diameter.

DAMPING

Shaft damping. — Next consider the damping coefficients C_{s1}, C_{s2}, and C_g. Damping in the shafts is due to the shaft material. In Equations 1 to 4 the coefficients C_{s1} and C_{s2} are taken to have the form

$$C_{s1} = 2\xi_s\left(\frac{K_{s1}}{\left\{\dfrac{1}{J_M} + \dfrac{1}{J_1}\right\}}\right)^{1/2} \qquad (14)$$

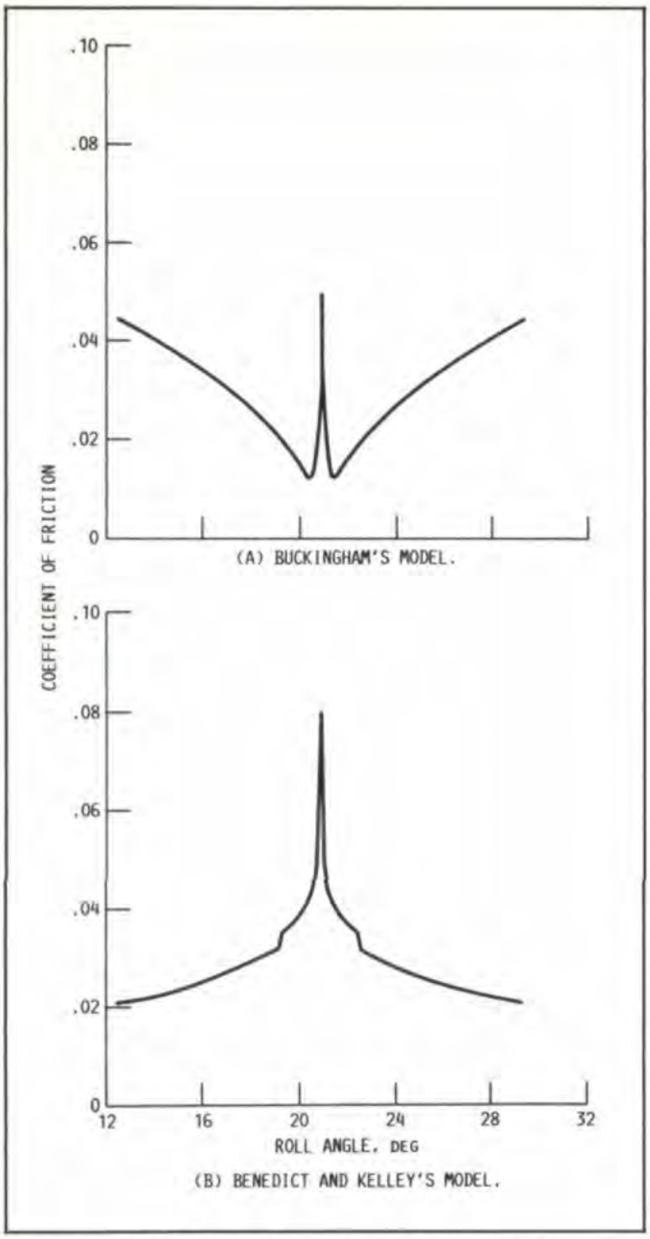

Fig. 5 – Friction Coefficient of Gears in Figure 4a at 1500 RPM.

and

$$C_{s2} = 2\xi_s \left(\frac{K_{s2}}{\frac{1}{J_L} + \frac{1}{J_2}} \right)^{1/2} \qquad (15)$$

where ξ_s represents the damping ratio. Experiments have shown that ξ_s has values between 0.005 and 0.075.[17]

Mesh damping. – Similarly, the effect of damping of the gear mesh is taken as

$$C_g = 2\xi \left[\frac{K_g R_{b1}^2 R_{b2}^2 J_1 J_2}{R_{b1}^2 J_1 + R_{b2}^2 J_2} \right]^{1/2} \qquad (16)$$

where, as before, ξ is the damping ratio. Measurements have shown ξ to have values between 0.03 and 0.17.[9-10]

Friction. – Equations 1 to 4 contain terms T_{f1} and T_{f2} which represent the frictional moments of the driving and driven gears. These moments occur because of the relative sliding of the gear teeth. Buckingham[18] has recorded a

semiempirical formula for the friction coefficient f of boundary lubrication as

$$f = 0.05e^{-0.125V_{sl}} + 0.002\sqrt{V_{sl}} \qquad (17)$$

where V_{sl} is the sliding speed measured in in./sec. An analogous expression for elastohydrodynamic lubrication has been developed by Benedict and Kelley[19] and by Anderson and Loewenthal[20] as follows:

$$f = 0.0127 \log (C_i W / F\mu_o V_{sl} V^2_R) \qquad (18)$$

where

C_i = 29.66 (SI units)
= 45.94 (English units)
W = the applied load, N/m (lb/in.)
F = face width, mm (in.)
V_R = rolling velocity, mm/sec (in./sec)
μ_o = lubricant viscosity, N-sec/m^2 (lb-sec/in.^{2})

Figures 5a and b show graphs of the friction coefficient as given by Equations 17 and 18 as a function of the roll angle. Figs. 6a and b show the resulting effect upon the friction torque.

MESH ANALYSIS

Fig. 7 illustrates the motion of a pair of meshing teeth. The initial contact occurs at A, where the addendum circle of the driven gear intersects the line of action. As the gears rotate the point of contact will move along the line of action APD.

Nomenclature

A_i	cross section area of ith element of gear teeth, mm^2 ($in.^2$)
C_g	damping coefficient, gear tooth mesh, N-sec (1b-sec)
C_s	damping coefficient of shaft, N-m-sec (in.-lb-sec)
E_e	effective modulus of elasticity, N/m^2 ($lb./in^2$)
F	*tooth face width, mm (in.)*
G	shear modulus, N/m^2 ($lb/in.^2$)
I_i	second moment of inertia of ith element of gear teeth, mm^2 ($in.^2$)
J_L	polar moment of inertia of load, m^2-Kg
J_M	polar moment of inertia of motor, m^2-Kg
J_1	polar moment of inertia of gear 1, m^2-Kg
J_2	polar moment of inertia of gear 2, m^2-Kg
K_g	stiffness of gear tooth, Nm/rad-m
K_s	stiffness of shaft, N-m/rad
L_{ij}	distance between elements i and j, mm
q_b	gear tooth deformation due to beam deflection, mm
q_c	gear tooth deformation due to contact deformation, mm
q_f	gear tooth deformation due to foundation flexibility, mm
$q=q_b+q_f+q_c$	total gear tooth deformation, mm (in.)
R_b	base radii of gears, mm (in.)
R_p	pitch radii of gears, mm
t_i	thickness of element i, mm
T_L	torque on load, N-m
T_M	torque on motor, N-m
T_{f1}	*torque on gear 1, N-m*
T_{f2}	torque on gear 2, N-m
V_{s1}	sliding velocity during tooth mesh, mm/sec
V_R	rolling velocity, mm/sec
W	applied load, N/m (lb/in.)
x_i	x-coordinate of element i, mm (in.)
y_i	y-coordinate of element i, mm (in.)
β	load angle, rad
δ	backlash, mm (in.)
θ	angular displacment, rad
$\dot{\theta}$	angular velocity, rad/sec
$\ddot{\theta}$	angular acceleration, rad/sec^2
μ_o	lubricant viscosity, N-sec/m^2 (lb-sec/$in.^2$)
ν	Poisson's ratio
ξ	damping ratio

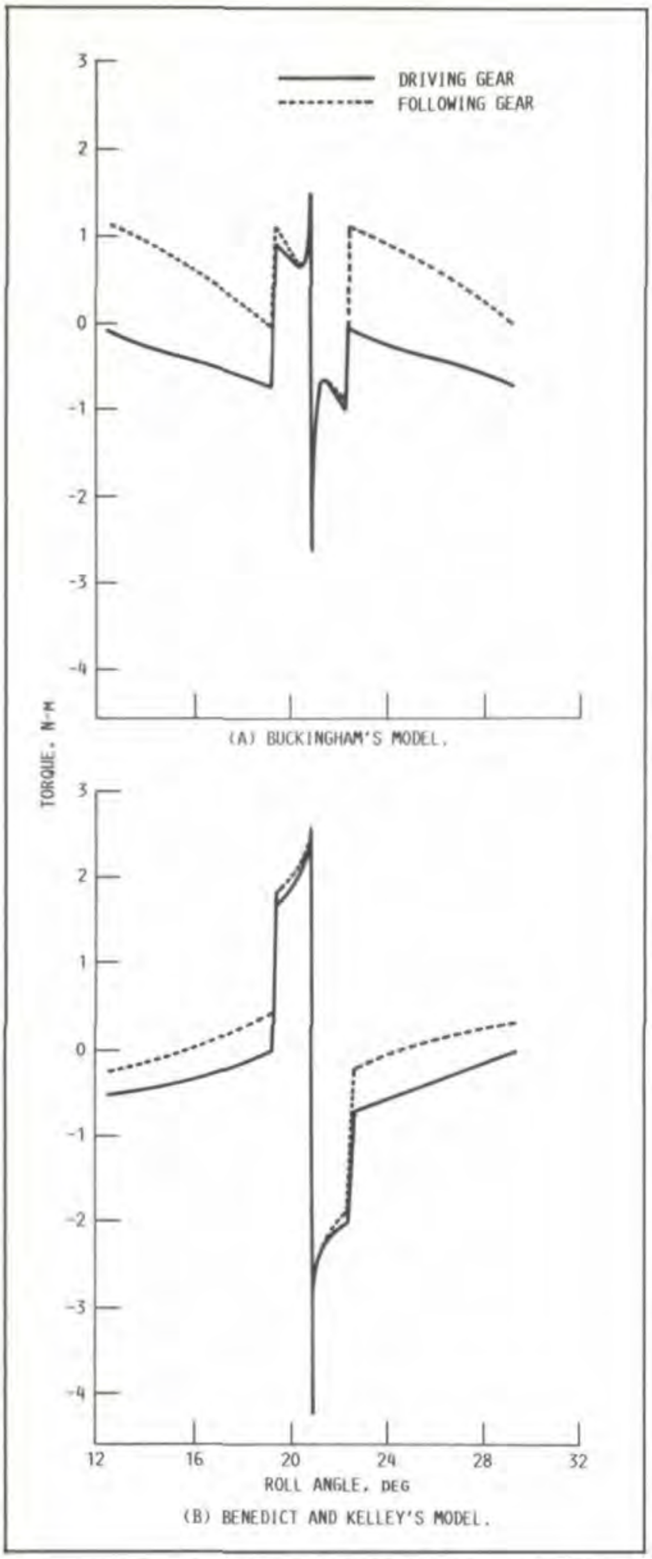

Fig. 6 – Friction Torque Variation Along the Contact Path for Gears in Figure 4a at 1500 RPM.

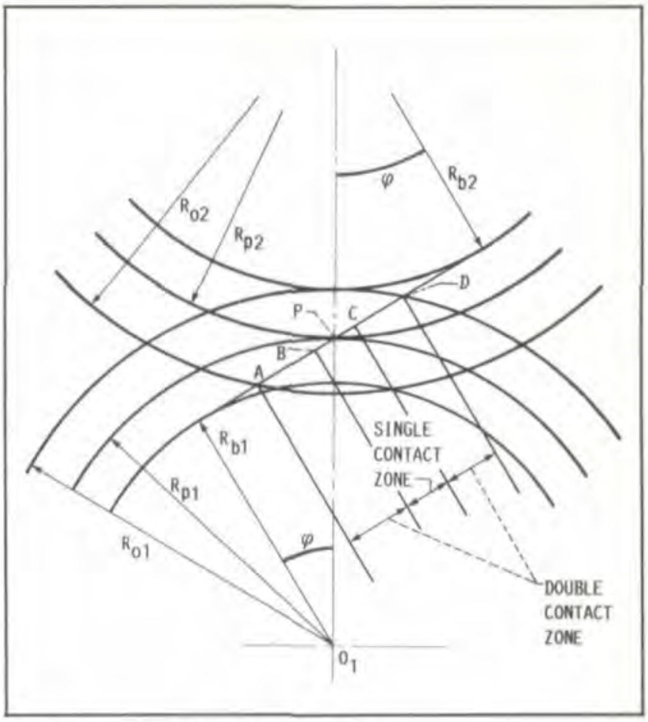

Fig. 7 – Illustration of Gear Meshing Action.

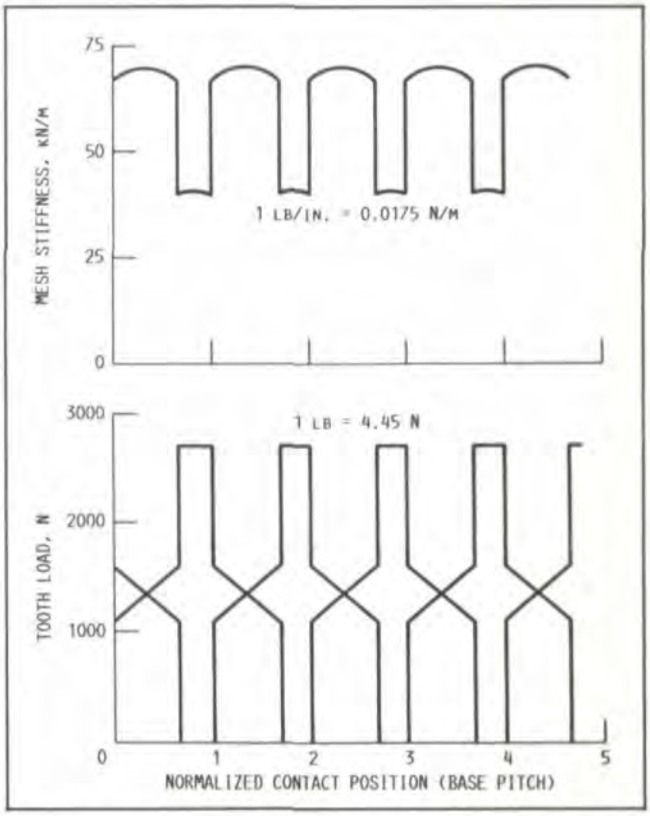

Fig. 8 – Typical Stiffness and Load Sharing of Low Contact Ratio Gears of Figure 4a.

When the tooth pair reaches B, the recessing tooth pair disengages at D leaving only one zone. When the tooth pair reaches point C, the next tooth pair begins engagement at A and starts another cycle.

In the analysis, the position of the contact point of the gear teeth along the line of action is expressed in terms of roll angles of the driving gear tooth.

Fig. 8 shows typical stiffness and load sharing characteristics through a mesh cycle. Let a series of mating tooth pairs be denoted as a, b, c, d and let points A, B, P, C, D be the same as those in Fig. 7. Then AB and CD represent the double contact regions, BC represents the single contact region, and, as before, P is the pitch point.

The stiffness values at double contact regions are clearly much higher than those at single contact regions. When gears rotate at appreciable speed, this time-varying stiffness as shown in Fig. 8 is the major excitation source for the dynamic response of the system.

Discussion

The objective of this analysis is to establish the governing differential equations and to present a procedure for solution. As noted, the equations themselves are nonlinear. However, they may be efficiently solved by using the following linearized-iterative procedure.

The linearized equations may be obtained by dividing the mesh cycle into n equal intervals. Let a constant input torque T_M be assumed. Let the output torque T_L be fluctuating because of damping in the gear mesh, because of friction, and because of time-varying mesh stiffness.

Let initial values of the angular displacements be obtained by preloading the input shaft with the nominal torque carried by the system. Initial values of the angular speeds may be taken from the nominal operating speed of the system.

The iterative process is then as follows: the calculated values of the angular displacements and angular speeds after one period are compared with the assumed initial values. Unless the differences between them are sufficiently small, the procedure is repeated by using the average of the initial and calculated values as new initial values.

Finally, observe that the term $(R_{b1}\theta_1 - R_{b2}\theta_2)$ in the equation of motion represents the relative dynamic displacement of the gears. Let δ represent the backlash. Let gear 1 be the driving gear. The following conditions can occur:

Case 1

The normal operating case is

$$R_{b1}\theta_1 - R_{b2}\theta_2 > 0 \tag{19}$$

The dynamic mesh force F is then

$$F = K_g(t)\, R_{b1}\theta_1 - R_{b2}\theta_2) + C_g(t)(R_{b1}\dot{\theta}_1 - R_{b2}\dot{\theta}_2) \tag{20}$$

Case 2

$$R_{b1}\theta_1 - R_{b2}\theta_2 \leqq 0 \text{ and } |R_{b1}\theta_1 - R_{b2}\theta_2| \leqq \delta \tag{21}$$

In this case, the gears will separate and the contact between the gears will be lost. Hence,

$$F = 0 \tag{22}$$

Case 3

$$R_{b1}\theta_1 - R_{b2}\theta_2 < 0 \text{ and } |R_{b1}\theta_1 - R_{b2}\theta_2| > \delta \tag{23}$$

In this case, gear 2 will collide with gear 1 on the backside. Then,

$$F = K_g(t)[(R_{b2}\theta_2 - R_{b1}\theta_1) - \delta] + C_g(t)\,(R_{b1}\dot{\theta}_1 - R_{b2}\dot{\theta}_2) \tag{24}$$

Conclusion

A low contact ratio spur gear transmisson model is devleoped. The model includes inertias of load and power source, stiffness of shaft, time-varying mesh stiffness, and damping and friction inside gear transmissions.

Governing equations of the model are derived and a linearized iterative procedure for the solution is presented. Parameter study including rotating speed, diametral pitch, applied load, damping, stiffness, and inertia will be presented in Part 2 of this article.

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Shaper Cutters–Design & Application Part 1

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Field of Application

Gear shaping is one of the most popular production choices in gear manufacturing. While the gear shaping process is really the most versatile of all the gear manufacturing methods and can cut a wide variety of gears, certain types of gears can only be cut by this process. These are gears closely adjacent to shoulders; gears adjacent to other gears, such as on countershafts; internal gears, either open or blind ended; crown or face gears; herringbone gears of the solid configuration or with a small center groove; racks; parts with filled-in spaces or teeth removed; and gears or splines with thick and thin teeth, such as are used in some clutches. Fig. 1 graphically illustrates the flexibility of the shaping process, and Fig. 2 shows some examples of gears which are mainly suitable to the shaping process, as well as some which may suit alternate methods, such as the rack. External spur and helical gears and pinions are the other conventional gears widely manufactured by the shaper cutter process.

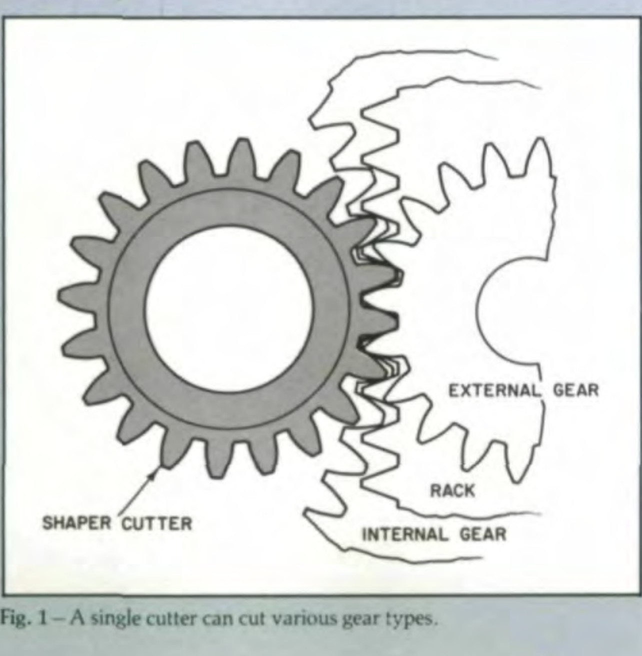

Fig. 1 – A single cutter can cut various gear types.

Other special involute and non-involute forms that can be shaped are roller chain sprockets, silent chain sprockets, and timing belt pulleys, as well as ratchets, saws, parallel key splines, involute splines, and many miscellaneous items.

Another shaper cutter type tool is used with a special machine to produce worms and similar screw items. In this configuration the tools are called thread generators.

The Shaping Method

The machining or cutting of gears with gear shaper cutters is a planing or shaping process involving a reciprocating motion of the tool, with chips being removed only on the forward direction of the stroke. On the return portion of the stroke no metal is cut, and the cutter and work must be separated so that the cutting edges are not dulled or damaged by dragging them backwards through the cut. This is a positive relieving action and is controlled by both the direction and amount of relief and must be established for each different application. On internal gears, with the enveloping of the gear around the cutter, it can be critical. The direction of work and cutter rotation also influence the direction or angle of the relieving motion and must be considered. On some machines the angle of relief is controlled by a cutter offset setting, and on others by a cam guide. The motions to develop the relieving action also can vary on different machines, with some moving the work away from the cutter and others moving the cutter away from the work.

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W.L. JANNINCK *is a consultant for ITW – Illinois Tools, a division of Illinois Tool Works, Inc. He has nearly 40 years' experience in an engineering and manufacturing environment in a cutting tool plant working in the design, development, and application of gear shaper cutters and gear hobs used in the generating process of gear cutting. He has served on various committees of AGMA and MCTI and is past chairman of the AGMA Cutting Tools Committee. He has also served on the SAE – ANSI Involute Spline Committee and ASME – Committee on Power Transmission Chains and Sprockets. He has written several articles on tool applications, gaging, gear design, and gear inspection.*

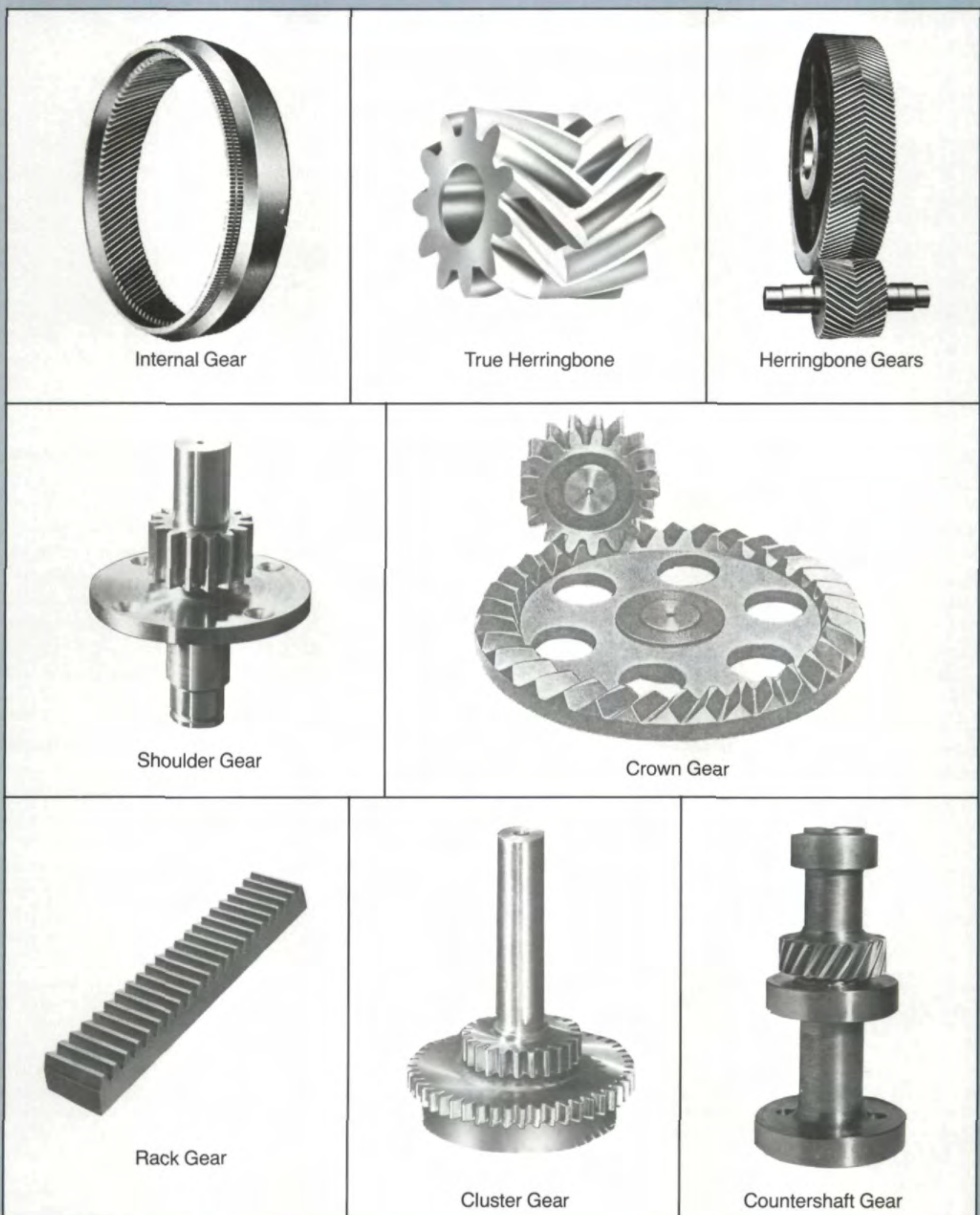

Fig. 2 – Group of gears cut by the shaping method.

For every stroke of the cutter ram, the cutter travels down, taking a cut, retracts, then returns to the top of the stroke cycle, and advances back into depth positioned for the next cut.

The cutting tool, that is, the shaper cutter, resembles a pinion or gear in appearance. It is relieved to create cutting edges and is usually constructed of one of the various tool steels. The machine contributes the reciprocating or stroking action and also steers the cutter in a timed rotary relation to the gear being cut, indexing the cutter-gear pair in a ratio according to the tooth numbers in cutter and tool and providing the plunging and feeding control.

The cutting edges of the cutter, if reciprocated along the cutter axis, sweep out an enveloping gear surface as seen in the phantom view in Fig. 3. This represents the cutting domain of the cutter.

The action provided by the gear cutting machine provides the motion needed to roll the cutter and blank together so that a generating action is developed. The generated result from a gear shaped involute cutting tool is a mating involute gear.

The length of stroke of the cutter must be just long enough to sweep from above the top face of the gear to just beyond the bottom face to allow the chip to separate, and the machine is adjusted to suit this requirement.

As the cutter approaches the circular blank at the initiation of the cut, it is gradually plunged into depth, while at the same time the circular indexing is taking place. The cutting process gradually removes material, and teeth are formed progressively on the blank periphery, until that point is reached where the teeth overlap and a complete gear is seen. Usually a second small infeed takes place, and a light finishing cut is made, duplicating the cycle and passing completely around the gear again. At this time the cutting action is relieved from most of the cutting load and freed from some of the blank material stresses.

It is not unusual when cutting herringbone gears that several roughing passes are made before the last final cut.

Body Types of Shaper Cutters

There are four basic types of gear shaper cutters in use.

Disk Type. This is the most common body type for shaper cutters. It is an arbor type cutter in the form of a disk with a central mounting hole, which is counterbored. It represents the optimum configuration for accuracy and use of tool steel in its construction. See Fig. 4.

Deep Counterbore Type. This is the second most used type of gear shaper cutter and is similar to the disk type, except it has a greater overall axial length to permit the use of a deeper counterbore for complete retention of the locking nut throughout the cutter life. See Fig. 5.

Deep counterbore cutters are used to cut into a clearance groove adjacent to a shoulder, to clear a raised hub or obstruction in the center of an internal gear, or to extend the

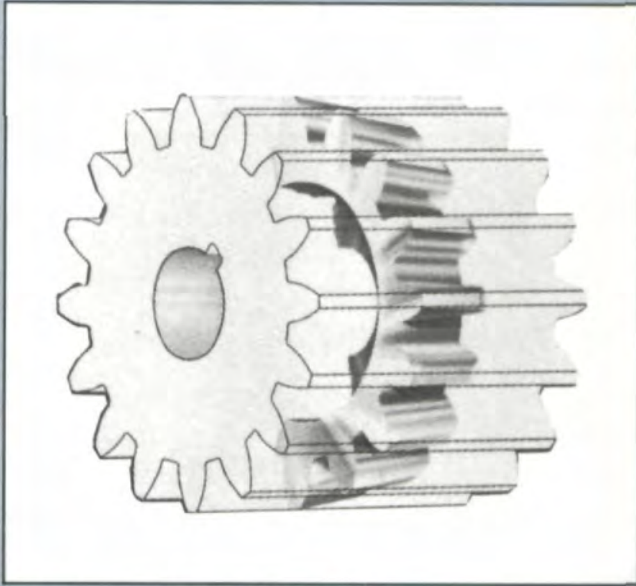

Fig. 3 – Envelope of reciprocated cutter.

Fig. 4 – Disk type cutter.

Fig. 5 – Deep counterbore cutter.

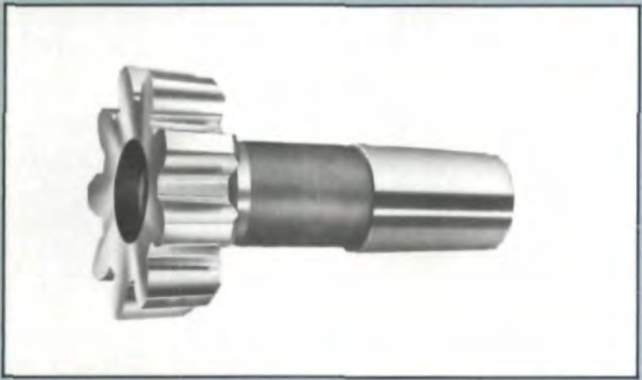

Fig. 6 – Shank type cutter.

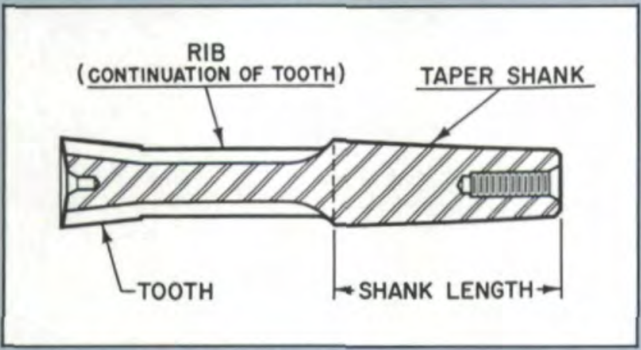

Fig. 7 – Shank type cutter with ribbed neck.

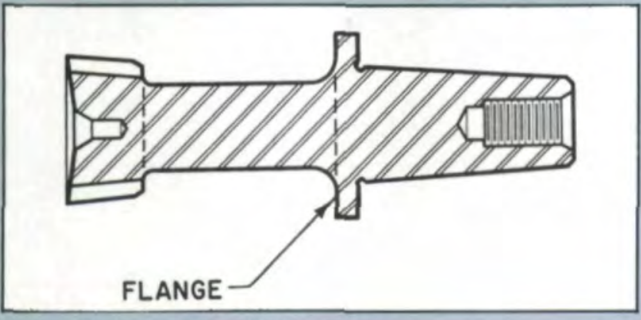

Fig. 8 – Shank type cutter with flange.

Fig. 9 – Straight shank cutter.

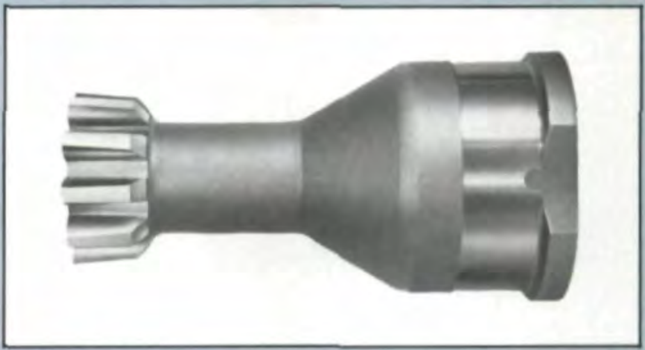

Fig. 10 – Hub type cutter.

reach of the cutter spindle. This cutter requires more tool steel to manufacture and slightly larger tolerances than a disk type.

Shank Type. This configuration is generally selected to cut a gear in a restricted space or for internal gears. The number of teeth is generally small, and the shank diameter size is picked accordingly from one of the four different standardized taper shank sizes. See Fig. 6.

If the cutter neck becomes too small and is weakened, a fluted or ribbed neck is used. The fluting lies just inside of the cutter form at the back face, thus increasing cutter strength. See Fig. 7.

Flange type shank cutters have a secondary mounting surface that comes into contact with the face of the cutter spindle, providing extra strength and stability of the cutter mounting. Taper diameter and flange must be sized closely so that the taper seats just prior to the flange making contact. See Fig. 8.

Most all taper shank cutters are made with a suitable standard sized tapped draw bar hole for securing the cutter in the spindle.

While straight or cylindrical shanks may be used, they are not a popular method of holding a shank type cutter. See Fig. 9.

Hub Type. This design is particularly adaptable to those sizes of cutters that fit between shank and hole type, and it can be mounted directly on the machine spindle without the use of a shank adapter. See Fig. 10.

Spur and Helical Cutters

Spur and helical gear shaper cutters look like spur and helical gears and are identified the same way. Spur cutter teeth have zero helix angle, and the teeth are aligned parallel to the cutter axis. Helical cutters have their teeth inclined at a helix angle to the cutter axis and follow along a helical path.

A right hand (RH) helical cutter is described the same way as a RH helical gear; that is, the teeth twist away from the observer in a clockwise direction. Likewise, on a left hand (LH) helical cutter, the teeth twist away from the observer in a counterclockwise direction.

A RH cutter produces LH external gears or RH internal gears. A LH cutter produces a RH external gear or a LH in-

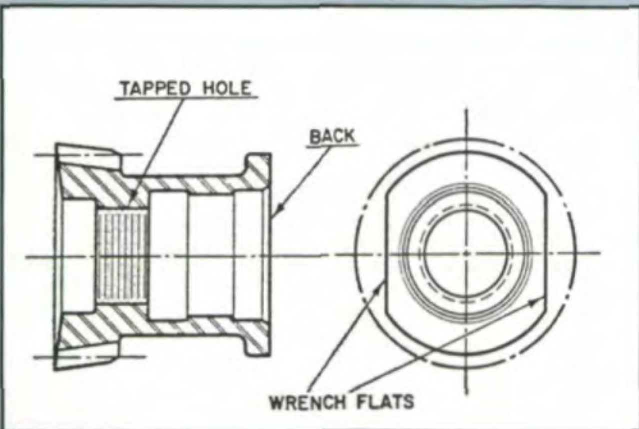

Fig. 10

Fig. 11 – RH helical cutter cutting a LH helical gear.

ternal gear. Fig. 11 shows a RH cutter engaged with a LH helical gear.

Herringbone Cutters

These are disk type cutters and are used in matched pairs of one RH and one LH cutter to generate true herringbone gears with solid continuous teeth. Sometimes a thin clearance groove is cut in the center of the gear face, which reduces the critical cutter stroke settings required with the solid style.

These cutters are also known as Sykes cutters in reference to the firm that originated this gear cutting process. See Fig. 12.

The gear geometry requires that the cutter face be flat, and a special groove and lip sharpening is used to have equal shear faces do the cutting.

Special herringbone machines are used to cut this type of true herringbone gear.

Cutter Blank Sizes

A source for some fundamental data on shaper cutter blank sizes is included in the ANSI Gear Shaper Cutters Standard. Besides giving the tolerance levels on the various elements, it gives suggested blank dimensions on nominal PD, bore, minimum counterbore, web, thickness, and nominal tooth length for spur and helical, disk and deep counterbore cutters for various diametral pitches. It also gives similar blank dimensions for herringbone cutters and, besides giving the four basic taper shank sizes, it gives some values for overall cutter length, pitch diameter, and tooth length based on the cutter diametral pitch.

These blank sizes are not fixed, but only a suggestion for a starting point and are usually fitted to the special circumstances encountered. At least if they are referenced, some realistic dimensions can be determined. Since the sizes usually minimize blank material, it is prudent to layout the planned dimensions for assurance of suitable strength and fit.

Fig. 12 – Set of RH and LH helical herringbone cutters.

Tooth Shape Change

As a shaper cutter is used and sharpened back, it changes in size. We can compare the profile at the front with that at the back and see a difference. Not only does the outside diameter become smaller, but also due to the side clearance, the tooth becomes thinner, and the tip width changes. Fig. 13 shows the profile of a spur cutter compared when new and at end of life.

The base circle does not change, but the cutter does use a lower segment of the involute curve. The base pitch remains constant, so during the entire life of the cutter a correct involute is cut.

If the base circle lies above the root of the cutter at the back, it is possible that the flank portion of the form below the base may come into action on the gear and cause a tip trimming or tip relief on the gear tooth. Fig. 14 illustrates an example of this action for a cutter with a small number of teeth at end of life.

If the base circle lies below the cutter root no involute trimming will occur at any time during the cutter life. Fig. 15 shows a cutter with these conditions. This is the preferred form for a good operating shaper cutter if other elements of geometry and circumstances permit.

Another area that is affected by the change in cutter size is the shape of the root fillet produced on the part. On an external gear the larger diameter, new cutter will produce a fillet with a larger radius of curvature. At end of life, the smaller diameter cutter produces a smaller radius fillet. Under normal circumstances the difference in actual fillet size is usually small and can be tolerated or, if necessary, adjustments made in the design parameters, such as cutter useful life, to minimize the effect.

Tooth Forms

Intentional modifications to the involute profile of the cut-

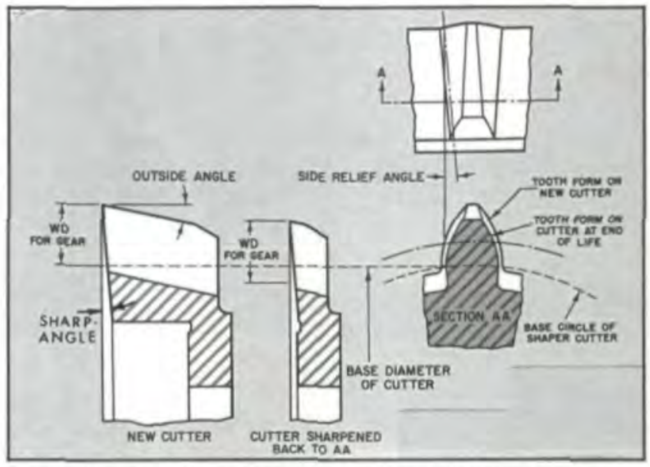

Fig. 13 – Illustrates change in profile from front to back on a spur cutter.

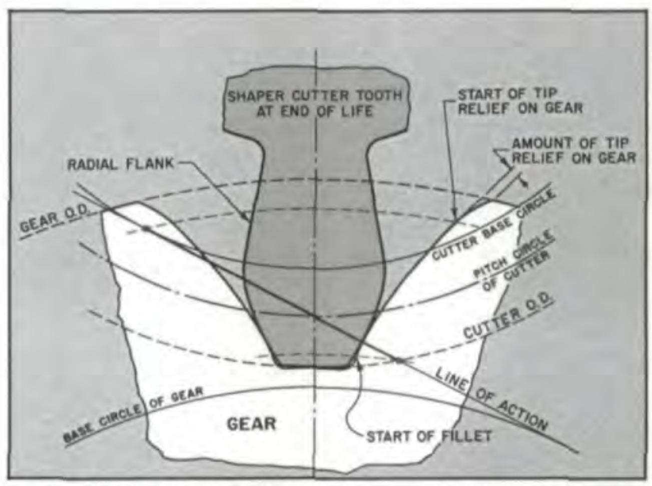

Fig. 14 – The effect of a radial flank on the form cut by trimming away part of the involute at the gear tip.

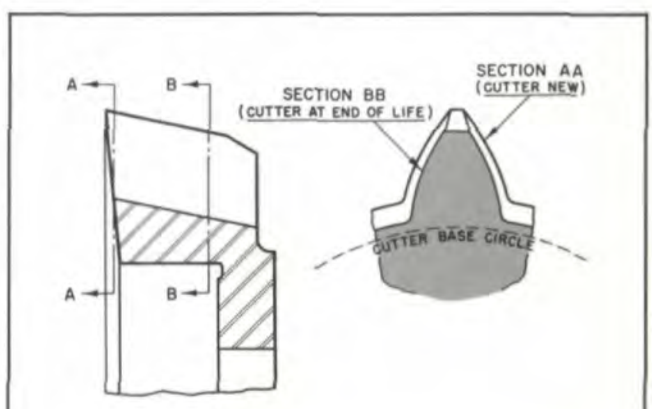

Fig. 15 – This sketch shows the case of the cutter base circle and involute lying above the cutter root diameter throughout the cutter's life. This is a preferred configuration.

ter to produce an alteration on the gear profile can fall into several categories.

Gear Tip Relief. Gear tip relief as shown in Fig. 16 can be produced by a cutter in three ways. The first method is by a constant approach built into the flank of the cutter that will cut a uniform tip relief throughout the cutter life. This is a controlled preferred method and is illustrated in Fig. 17.

The second is by using a straight flank tangent to the involute just above the base circle. This modification produces a variable amount of tip relief during cutter life. See Fig. 18.

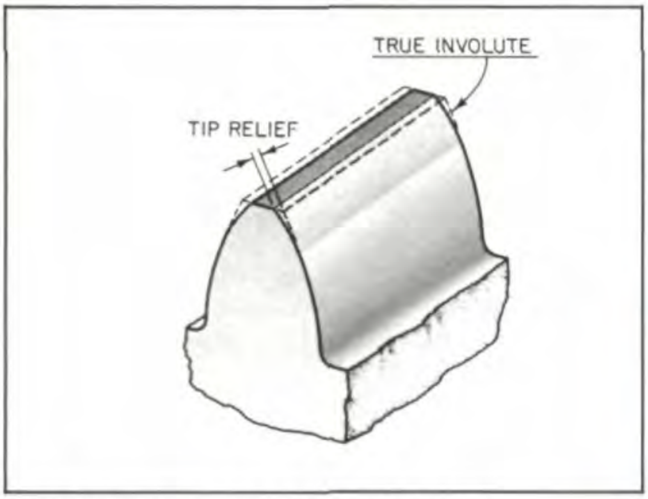

Fig. 16 – Isometric view of a gear tooth with a tip relief.

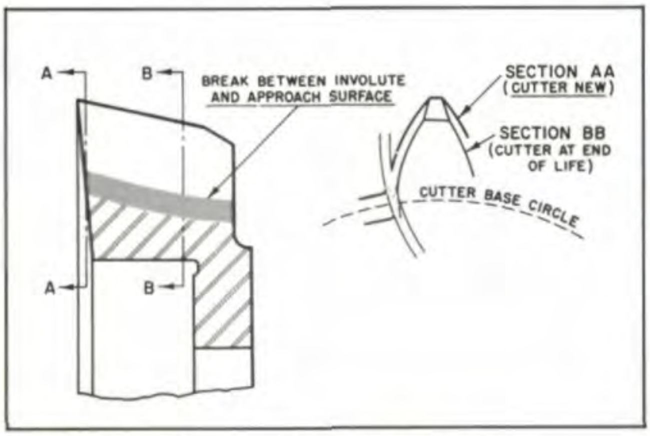

Fig. 17 – Diagram of a cutter with a "constant approach".

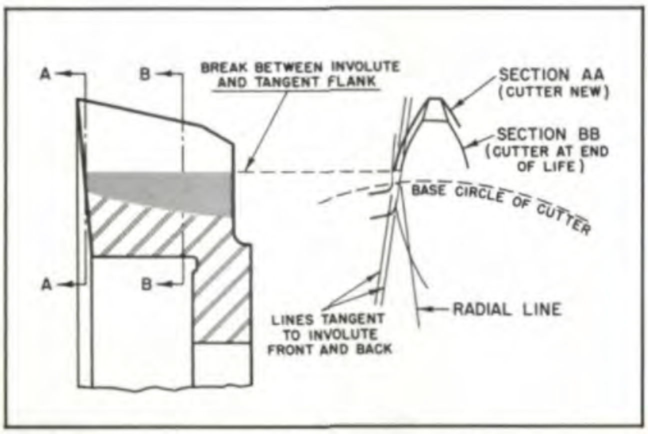

Fig. 18 – Diagram of a cutter with a "tangent approach".

The third is by using a radial flank tangent to the form at the base circle, which may produce no tip relief when the cutter is new and then gradually more as the cutter is sharpened back. See Fig. 19.

The latter two modifications are applied with a flat wheel generating grinder doing one flank at a time. The third way is not really used as a method of producing a specified relief, but can occur naturally because of the construction of some shaper cutters, especially those with low numbers of teeth and/or low pressure angles.

Gear Tip Chamfer. Whenever it is necessary to produce a tip chamfer on the gear being cut, such as is shown in Fig. 20, a semi-topping tooth form is used. A ramp is added to the cutter root flank and is located so a reasonably uniform chamfer is produced throughout the cutter life. See Fig. 21.

Gear Flank Undercut. When further processing is done on a gear after shaping, such as shaving, grinding, skiving, or cold rolling, the gear is prepared by undercutting the flank in-

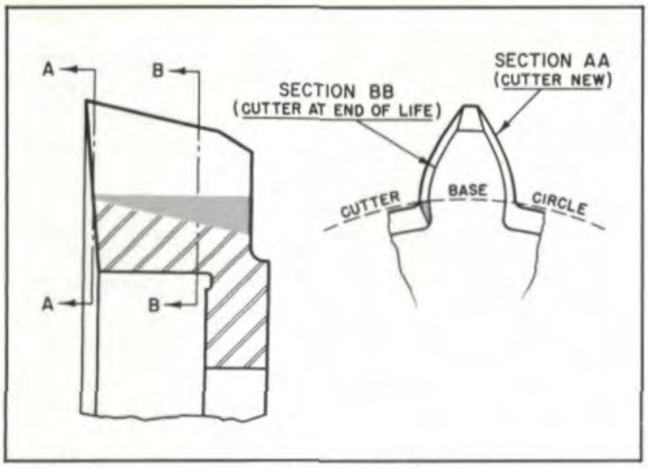

Fig. 19 – Diagram of a cutter with a radial flank.

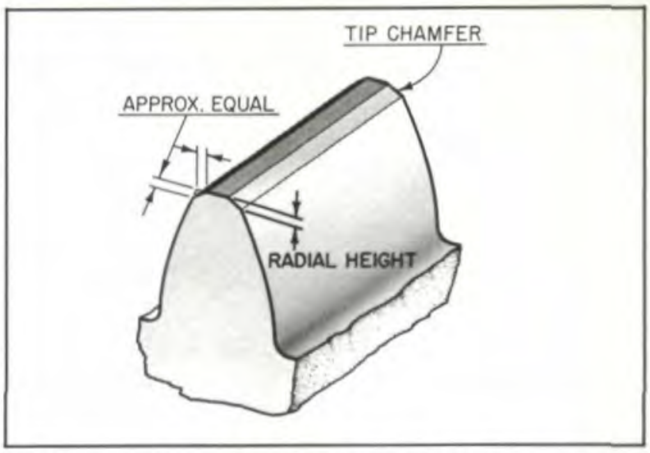

Fig. 20 – Isometric view of a gear tooth with a tip chamfer.

volute near the gear root. This is accomplished by the use of a protuberance on the cutter form. See Fig. 22.

Topping. Occasionally it is necessary to shape a gear, cutting the outside diameter at the same time that the gear is cut. Such a topping cutter is restricted to spur cutters and helical cutters with a circular sharpening.

Full Fillet Root. The tip of the cutter tooth generates the root fillet on the gear teeth. If the gear fillet is to be a full radius, then the cutter tip radius also will be full. For gears without a full radius a smaller corner radius or, in some cases, a small corner chamfer is used on the cutter tip.

Cutting Internal Gears

One of the very important applications of shaper cutters is the generation of internal spur and helical gears. Some extra considerations have to be given to the rather confining conditions of having the work part surrounding or enveloping the cutter.

The normal practice is to use as large a cutter as possible and still avoid the special geometry problems of internal gear cutting. Table 1 presents a listing of both a maximum and a minimum number of teeth suggested for the cutter for three different tooth form systems. Generally if a cutter is picked

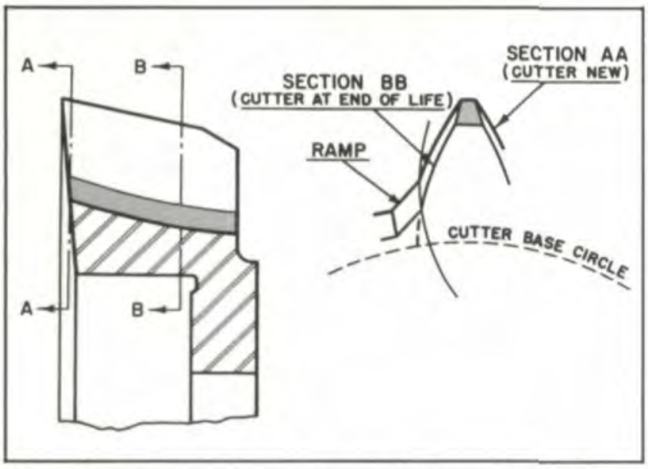

Fig. 21 – Diagram of a semi-topping cutter with a ramp which cuts the tip chamfer.

Table I
Size Guide for Selection of Shaper Cutters
For Internal Gears
All values in number of teeth

Internal Gear	20° Stub			20° Full Depth			25° Full Depth		
	Max. for Cutter	Max. in Pinion	Min. for Cutter	Max. for Cutter	Max. in Pinion	Min. for Cutter	Max. for Cutter	Max. in Pinion	Min. for Cutter
22	10	14	10	–	–	–	10	16	10
24	11	16	11	10	14	10	11	18	10
26	12	18	11	10	16	10	12	20	10
28	13	20	12	11	18	11	13	22	11
30	14	22	13	12	20	12	14	24	11
32	16	24	14	14	22	13	16	26	12
36	19	28	16	16	26	15	19	30	13
40	22	32	18	19	30	16	22	34	14
44	25	36	19	22	34	18	25	38	15
48	28	40	21	25	38	20	28	42	16
52	32	44	22	28	42	21	32	46	17
60	38	52	24	35	50	23	38	54	18
70	47	62	26	43	60	26	47	64	19
80	56	72	28	51	70	28	56	74	20

within the limits for the number of teeth being cut, it will be suitable. If there is any departure from standard proportions or if long or short addendums are used, the chart can only act as a guide.

There are four specific problem areas to be investigated.

Feed Fouling. Fig. 23 shows an example of a cutter that in effect overhangs part of the finish gear profile, and as it is fed radially into the part, it will machine away that part of a tooth flank. If this occurs, it generally means the cutter used had too many teeth. It was too big.

Rubbing Interference. Fig. 24 is an illustrated case of the cutter outline overlapping the cut teeth when the cutter is retracted for the return stroke. This would drag the cutter back, rubbing on the gear teeth. Cutter and part damage is likely. One solution is to first try a change in the relieving direction to see if a retraction path can be found to clear the interference. Another is to consider using a cutter with fewer teeth.

Flank Trimming. Fig. 25 shows a case of flank trimming on the addendum of the gear being cut during generation. In actuality a tip relief is produced, and if it is not acceptable on the product, it means the cutter may be too small or even undersize. It generally is corrected by using a larger number of cutter teeth or increasing cutter diameter.

Tip Fouling. Fig. 26 shows a geometrical phenomenon that also is unique to internal gears. As the cutter tip passes through a cycle of entering, generating, and leaving a tooth space, if its path crosses the corner of the gear tooth, it will machine away the interfering area. All the gear teeth are affected by this action if it occurs. In this case the cutter is too

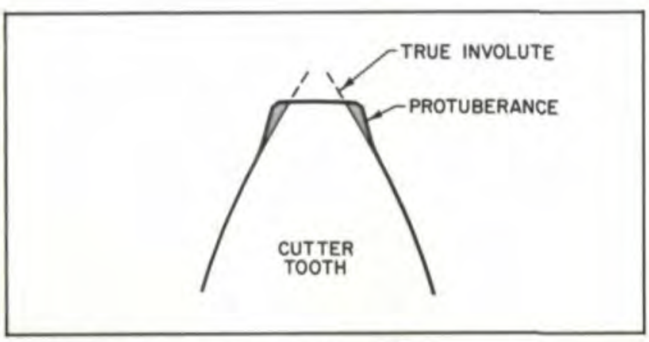

Fig. 22 – Gear shaper cutter tooth with protuberance.

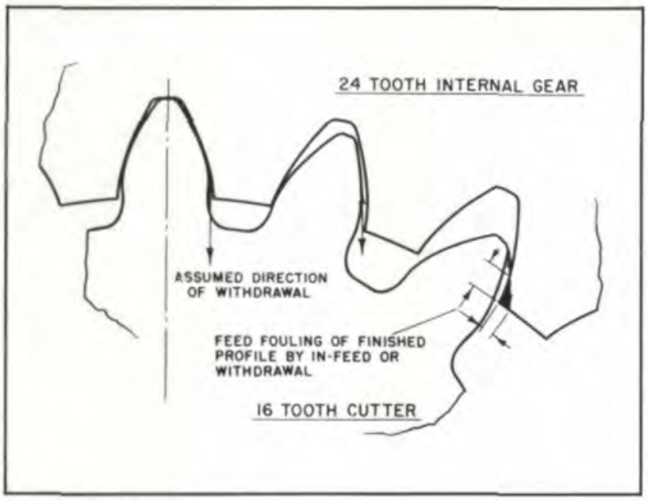

Fig. 23 – Feed Fouling.

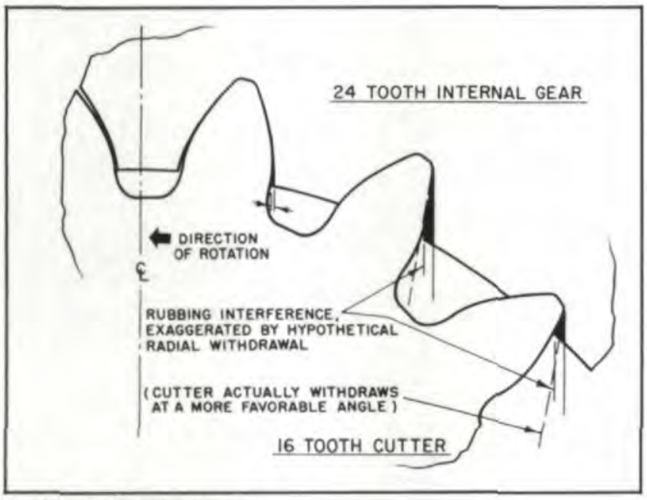

Fig. 24 – Rubbing Interference.

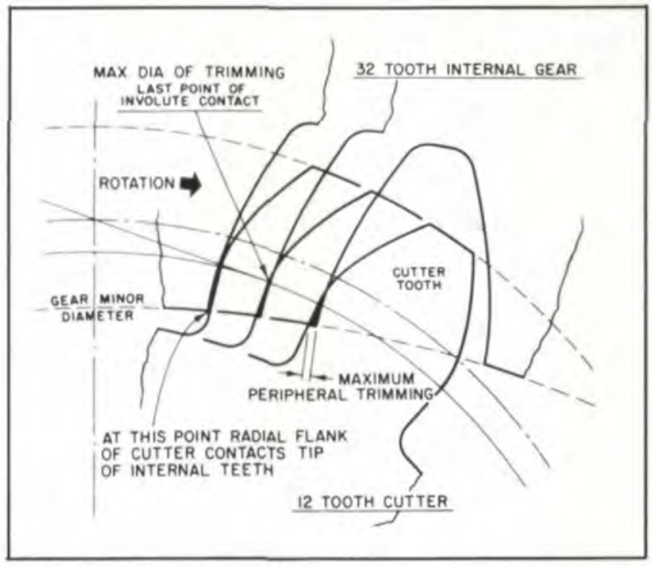

Fig. 25 – Flank Trimming.

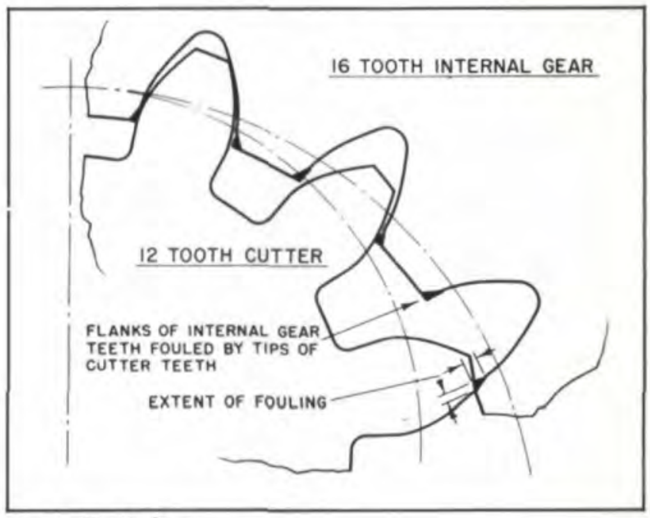

Fig. 26 – Tip Fouling.

large, and a smaller number of teeth should be considered.

If for an urgent job, a cutter must be selected from an available stock list, and it violates the suggested tooth numbers, it is best to pick a lower cutter tooth number. There are less damaging problems with a smaller cutter.

In the above four cases, as well as the case of the influence of cutter oversize or undersize on the part/cutter relationship, the investigation of internal cutter design is a perfect problem for computer aided design. To be sure a cutter will work properly an analysis or mathematical model must be made. This assures all critical specifications, including gear root fillet, requirements are met.

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K_v	dynamic factor
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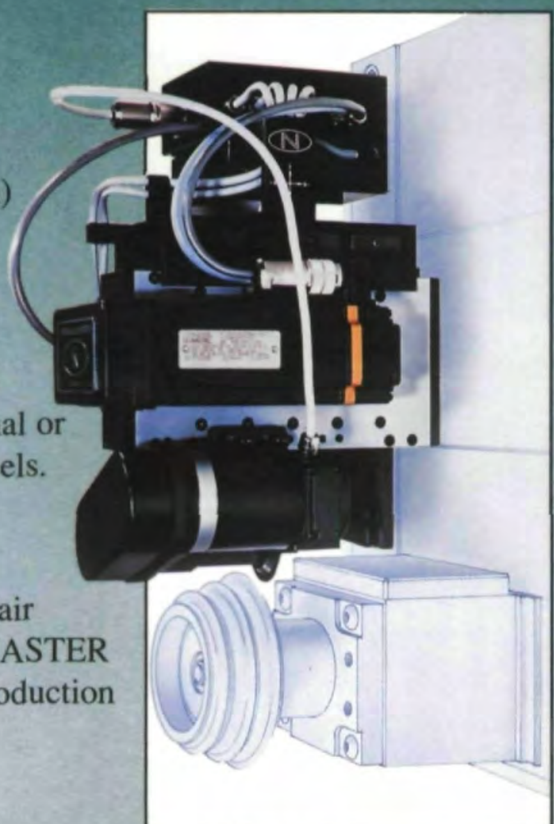
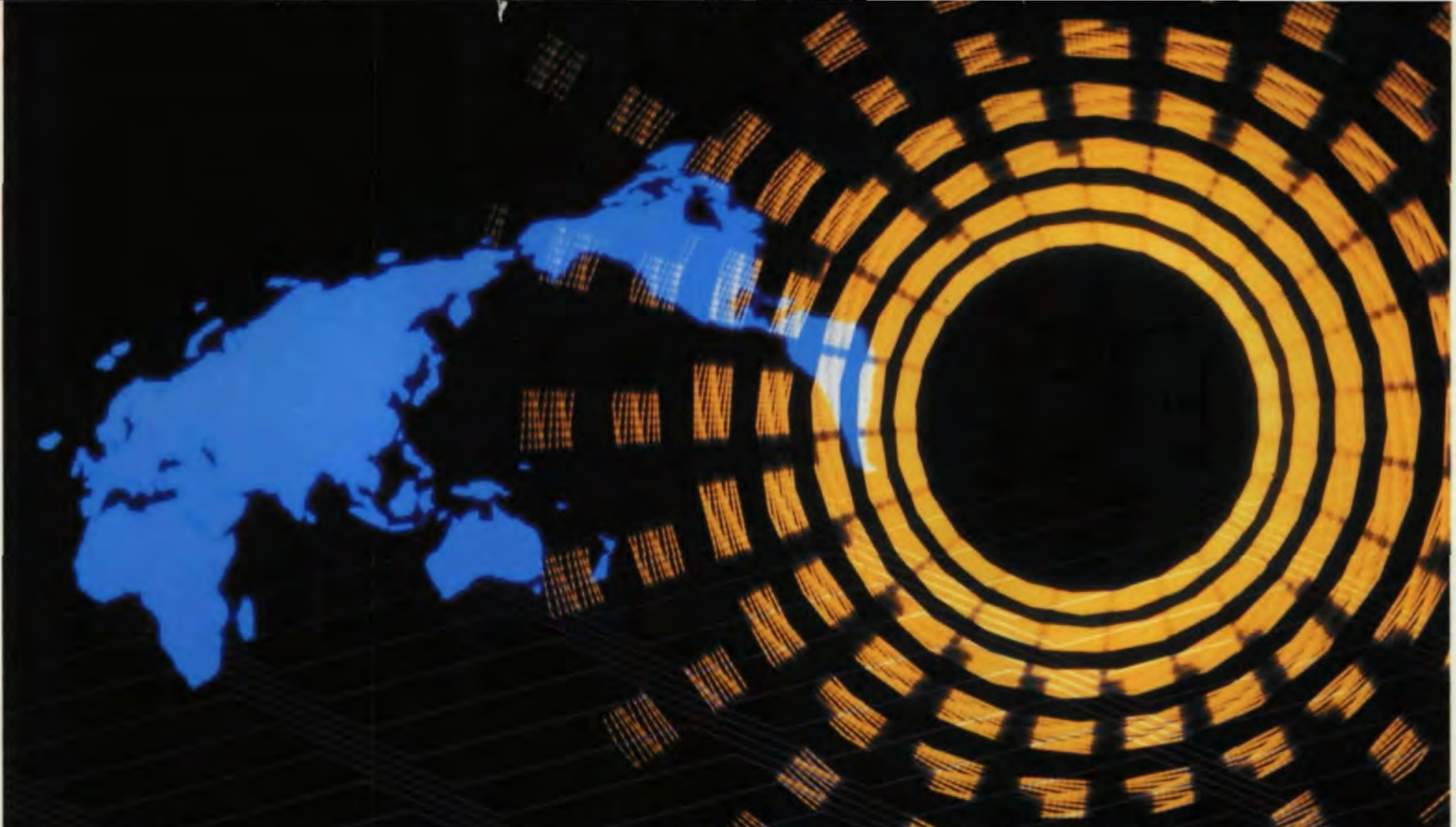
W	load

W_a	allowable tangential load
W_t	transverse or tangential component of load
W_n	load component normal to helix in pitch plane
W_N	load normal to surface
W_r	radial component of load
w	load per unit length
M	bending moment
θ	involute polar angle
P	transmitted power
P_{ac}	allowable power
P_{sc}	service horsepower
r_t	throat-form radius
ρ_o	relative radius of curvature
m_o	modified contact ratio
m_N	load sharing ratio
s	stress
s_t	calculated bending stress
s_c	compressive stress
s_s	shear stress
s_a	allowable contact stress number
s_W	working contact stress number
s_{at}	allowable bending stress
y	tooth-form factor for circular pitch
Y	tooth-form factor for diametral pitch
T	torque
T_G	torque of gear
T_p	torque of pinion
v_N	velocity normal to surface





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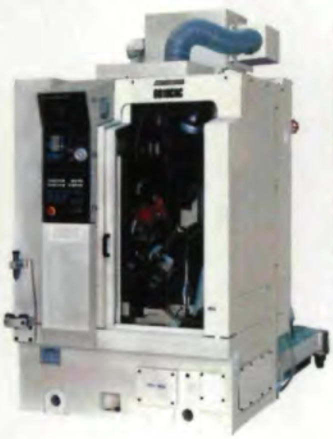

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