GEAR TECHNOLOGY

THE JOURNAL OF GEAR MANUFACTURING

MARCH / APRIL 1994

HEAT TREATING

SURFACE LAYER & WEAR BEHAVIOR OF NITRIDED GEAR DRIVES

THE FUNDAMENTALS OF GEAR PRESS QUENCHING

THIN RIM PLANETARY GEAR STRESSES

PHOTOGRAPHING GEAR FAILURES

WHAT IS ISO 9000 AND WHY YOU SHOULD CARE

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

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ISO 9000:

GLOBAL MARKET SALVATION OR A PIG IN A POKE?

ISO 9000 is the latest hot topic in marketing and manufacturing circles. Everyone seems to be talking about it, but few seem to understand it completely. Depending on whom one talks to, it's either the greatest thing to hit industry since the assembly line, another cash cow for slick consultants, a conspiracy on the part of Europeans to dominate global markets, or the next necessary step to compete in the global economy of the twenty-first century. It may be all of the above.

To help our readers sort through the wealth of information and opinion on this subject, in this issue we begin a three-part series on ISO 9000. In this issue, we will present a general overview of ISO 9000, some of its promises and problems and the impact it may (or may not) have on your business. In the next issue we will discuss the complicated business of whether you should hire an ISO 9000 consultant and how to go about it if you do. Finally, we will discuss strategies for streamlining the certification process to save money.

Like it or not, whether to become ISO 9000-certified is a question the gear manufacturing industry will have to confront. One thing that does seem certain is that it is not just another business fad, and it will not go away.

However, a head-long leap onto the ISO 9000 bandwagon may not be a good idea either. For many companies, the first clue that they need to be concerned about ISO 9000 will be a clear message from important customers that ISO 9000 certification will be part of the price of doing business with them in the future. For such companies, the question will not be whether to become certified, but when. For companies that don't receive that message from their customers, the course of action is not as clear.

For the present at least, most gear suppliers will have the time to study the issues and, if they opt for certification, to implement it in an orderly fashion. At present, two major gear industry customers, the automobile manufacturers and the defense industries, are not requiring certification.

However, there are real benefits to be gained by going through the process, whether your customers demand it or not. First is that ISO 9000 certification is

PUBLISHER'S PAGE

"A headlong leap onto the ISO 9000 bandwagon may not be a good idea."

Continued on page 48.

CALENDAR

MARCH 3-4
ANSI 1994 Annual Public Conference. ANA Westin Hotel, Washington, D. C. "Accessing Global Markets Through Standardization." Two days of workshops and seminars on key global standardization issues. Call (212) 642-4952 for more information and registration packets.

MARCH 3-5
AGMA 1994 Annual Meeting, Marriott Marco Island Resort, Marco Island, FL. The theme of the meeting will be "Perspective On World Class Manufacturing." Jerry Flaherty, Group President of Caterpillar, will be the keynote speaker. For more information, contact AGMA at (703) 684-0211 or fax (703) 684-0242.

APRIL 12-14
SME Advanced Gear Processing and Manufacturing Clinic. Holiday Inn Airport, Indianapolis, IN. Presentation of state-of-the-art solutions to today's gear manufacturing problems. Panel discussions, plant tours and tabletop exhibits. For more infomation, call SME Conferencing Division at (313) 271-1500.

APRIL 18-20
American Society for Metals International Heat Treating Conference: Equipment & Processes. Hyatt Regency Woodfield, Schaumburg (Chicago), IL. Over 100 presentations on new furnaces developments, atmosphere control technology, temperature measurement, quenching process technology advances, recycling and environmental issues, and more. Contact ASM at (216) 338-5151, x703 or fax (216) 338-4634. Use registration code IH 50.

APRIL 18-21
Material Handling Industry of America North American Material Handling Show and Forum, Cobo Hall, Detroit, MI. The most comprehensive showcase of material handling equipment, systems and technologies on the continent in 1994. Educational forums on improving productivity in manufacturing and physical distribution.
Contact MHIA at (704) 522-8644 or (800) 345-1815. Fax (704) 522-7826.

CALL FOR PAPERS
The Induction Hardening Steering Committee of the Gear Research Institute is planning an International Conference on Induction Hardened Gears and Precision Components to be held in Indianapolis, IN., May 15-17, **1995**. The committee is seeking papers on issues relating to processing, quality assurance and engineering performance, including distortion control, case depth, optimization and control, cost factors, quenching, materials selection, pretreatment, fatigue, scoring, and more. Abstracts of proposed topics are due **April 1, 1994**. For more information, contact Sharon Schaefers, GRI, (708) 241-0660 or fax (708) 241-0662.

GEAR SCHOOLS
Dates for **AGMA Training School for Gear Manufacturing** for the remainder of the school year are **March 7-11, May 16-20, September 12-16 and November 14-18.** All sessions will be held at Daley College in Chicago, IL. Contact AGMA at (708) 684-0211 for a registration packet or more information.

What is ISO 9000 and Why Should I Care?

Amy Zuckerman

What follows is the first of three articles we will be running on ISO 9000 and what it means for the gear industry. This first article will cover what ISO 9000 is, what some of its benefits – and problems – are, and whether your company should be a candidate for this certification process. In our next issue, we will consider the important question of how, when, and if to hire an ISO 9000 consultant. The final article in this series will discuss ways to save money while streamlining the certification process in your company.

NAFTA is a reality. The GATT treaty seems certain to become reality.

As the world focuses on treaties and agreements, the thorny question of how to make international trade really work has been receiving scant attention. Which brings us to ISO 9000, the international quality standards program that has been touted as a replacement for all quality assurance standards worldwide.

ISO fits neatly into the "how-to" issues of foreign trade. Companies who buy into the ISO 9000 program – companies from Minneapolis to Mombasa – are supposed to be able to select among ISO 9000 certificate bearers worldwide and feel confident that they are dealing with a "quality" company. By offering this assurance of "quality," ISO 9000 is expected to breed credibility in the world market and thereby make overseas working arrangments far easier.

If boosters of ISO 9000 talk about creating a "quality" company, many world market players consider standards programs like ISO 9000 a key to international trade. This is especially true in Europe, where there is general recognition that the purpose of quality assurance – the backbone of ISO 9000 – is domination of global markets. European ISO officials understand that he who dictates how companies operate will dictate the rules of the game. They understand that ISO 9000 is big business and that the registrars that are the most respected will end up on top.

With the glow of overseas markets beckoning

MANAGEMENT MATTERS

internationally many companies – especially those with overseas interests – are coming to understand the connection between standards and trade. Some have responded to the ISO 9000 promise for market purposes, while others have been coerced into ISO 9000 compliance by multinational customers. Whatever the reason for pursuing ISO 9000, a total of 40,000 ISO 9000 certificates have been issued to companies throughout the world. And there are now 95 countries worldwide that have accepted the ISO 9000 standards on a voluntary basis.

At present, but subject to change, the ISO 9000 Series is comprised of subdivisions 9001, 9002 and 9003, which relate to a company's scope of activity at any particular site. Standards are introduced

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throughout a company to increase quality performance and awareness in all company functions. As such, the standards apply to service companies as well as manufacturers. It should

Amy Zuckerman
is co-principal of IN/EX Information Export, an international marketing consulting firm in Pelham, MA, and the author of ISO 9000 Made Easy. A Self-Help Guide to Certification.

be well understood that ISO 9000 is a for-profit, open market program.

Also note that the main buzzword of the U.S. ISO 9000 program is "quality assurance," not "product assurance." Quality assurance is only tangentially connected to product assurance. One widespread criticism of the ISO 9000 program is that it is not connected directly enough to product standardization. In effect a certified company could manufacture a poor product, but in a very efficient manner.

Key to ISO 9000 implementation is internal auditing whereby all company functions – even office procedures – are constantly monitored. Documentation is the primary auditing tool. The standards are necessarily quite general and flexible: Applications will vary appreciably from company to company.

In general, certified companies claim increased productivity, fewer accidents, and better employee morale and communication. Most certificate holders market the fact for competitive advantage.

Some Growing Pains: Some Politicalization

There are problems with

Certified Organizations Categorized by Industry, Type of Certificate and "Certificate Age"

		Certificate < 1 year	
	Category	ISO 9001	ISO 9002
1	Industry		
2	Food and Drinks		2
3	Fabrics		
4	Clothing		
5	Leather/Shoe		
6	Wood/Furniture	1	
7	Paper/Packaging		4
8	Publishing		
9	Oil/Chemical	16	25
10	Rubber/Synthetics	3	2
11	Building Material	2	2
12	Metal (basic)	1	2
13	Metal Products	5	11
14	Machines	6	1
15	Electronics	14	9
16	Transportation (related)		45
17	Instruments/Optical	3	1
18	Other Industries	7	5
19	Industrial Trade		12
20	Mineral Exploration		
21	Building (related)	4	1
22	Trade		
23	Logistics/Communication		2
24	Banks/Insurance		
25	Professional Services	9	13
26	Other Services		
27	Laboratories		2
	TOTAL	71	139

how the program is being administered internationally. ISO 9000 consultants are not regulated; the American Registrar Accreditation Board (RAB) has insufficient funding to maintain records, and U.S. registrars are not always accepted overseas. The RAB has recently issued "Conflict of Interest" rules for registrars.

Technically, the primary criticism of ISO 9000 is that it is too vague to be of significant value for industries in general. For example, the U.S. Big Three auto-makers include complete ISO 9000 standards in their supplier quality guidelines, but have supplemented them, expanding 15 of 20 ISO 9000 sections. For the time being, the Big Three recommend that suppliers need only meet their standards.

Members of the ISO and European national accreditation boards are looking at modifying ISO 9000 along industry-specific lines. They, along with American boards, are also considering more stringent regulations for registrars.

The major industry complaint is the cost of certification. The base cost of registration fees runs about $35,000. However, consultants, employee time and documentation costs can easily drive total costs up well into six figures. The National Association of Manufacturers' Small Manufacturers Forum is creating a cross-industry council to lobby for ISO 9000 cost concessions for smaller manufacturers.

Many industry leaders are questioning the need for ISO 9000 certification at all at this time. Dean Beachler of the National Tooling & Machining Association and C. Daniel Whelan, quality control manager of Ford Motor Co., are both concerned with the lack of regulation surrounding the U.S. program, as well as the high cost of certification. Neither the NTMA nor automotive leaders are urging their suppliers to become ISO 9000 certified.

Other industry leaders are just as adamant that their suppliers seek certification. In the chemical

Chart compiled by Dutch ISO 9000 consultant Gerrit Zijlstra.

Certificate > 1 year			
ISO 9001	ISO 9002	ISO 9003	TOTAL
			2
			1
	3		7
	2		2
8	14		63
2	7		14
2	1		7
	1		4
10	15	1	42
4	7		18
17	2		42
1	20		66
2	3		9
	2	1	15
2	5		19
4	2		11
	3		5
7	2	1	31
1	1		4
60	90	2	362

Who's Who in ISO 9000

INTERNATIONAL ISO 9000 PROGRAM

ISO or International Organization for Standardization

• **Purpose** – To create and promulgate ISO 9000 quality assurance standards internationally. Does not regulate the ISO 9000 program. Based in Geneva, Switzerland.

• **Status** – Non-profit, though some ISO board members operate ISO 9000 consulting firms.

• **Products for Sale** – Subscriptions to *ISO 9000 News*, memberships.

National Accreditation Bodies

• **Purpose** – Charged by the ISO to accredit companies in the private sector to carry on ISO 9000 work in the field and to authorize ISO 9000 registration (certification in Europe). Some accreditation bodies (mostly European) actively regulate and monitor ISO 9000 activities within their boundaries.

• **Examples** – United States Registrar Accreditation Board (RAB), Dutch RvC, British BSI, etc. . . .

• **Status** – Most accreditation bodies are non-profit organizations that charge fees to companies seeking registrar/certifier status. In Europe, many accreditation bodies are government-regulated. This is not the case in the U.S.

European Organization for Testing and Certification (EOTC)

• **Purpose** – Established by the European Community (EC) to promote harmony between European and international ISO 9000 organizations.

• **Status** – Non-profit, 70% funded by the EC. Will have to become self-supporting within the decade.

Registrars (Certifiers in Europe)

• **Purpose** – To offer ISO 9000 registration (or certification). These are the entities that national accreditation body regulate.

• **Status** – Most registrars/certifiers operate for profit, though some are non-profit. Some offer ISO 9000 consulting services, an issue currently being debated in the U.S. and abroad.

ISO 9000 Consultants

• **Purpose** – Individuals or companies that offer ISO 9000 assistance in the field.

• **Status** – Most are for-profit; unregulated.

U.S. ISO 9000 PROGRAM

American National Standards Institute

• **Purpose** – To create standards (product and quality assurance) for U.S. industry. One of several such organizations throughout the country. Not yet recognized as the official U.S. standards agency. Offices in New York City.

• **ISO 9000 Role** – Coordinator of national ISO 9000 program. U.S. representative at the Geneva-based ISO. May soon be directly involved with the U.S. RAB in the registrar accreditation process.

• **Status** – Non-profit.

• **Products for Sale** – Includes standards series for ISO 9000 and other programs, as well as memberships.

American Society for Quality Control (ASQC)

• **Purpose** – "A society of individuals and organizations dedicated to the on-going development, advancement and promotion of quality concepts, principles and technologies . . . [with the aim of] facilitating continuous improvement and increased customer satisfaction by identifying, communicating and promoting the use of quality principles, concepts and technologies, and thereby be recognized throughout the world as the leading authority on and champion for quality." (From ASQC mission statement) Based in Milwaukee, WI.

• **ISO 9000 Role** – "Parent" agency to the U. S. Registrar Accreditation Board.

• **Status** – Non-profit.

• **Products for Sale** – Books on quality issues, videos, audio tapes. Maintains a promotional catalog that lists mugs, posters and other such items for sale.

United States Registrar Accreditation Board

• **Purpose** – A subsidiary of ASQC, the RAB oversees accreditation of ISO 9000 registrars. May soon share these duties with ANSI. Based in Milwaukee, WI.

• **Status** – Non-profit, though some board members operate for-profit ISO 9000 consulting companies.

CEEM

• **Purpose** – To disseminate ISO 9000 publications, sponsor ISO 9000 seminars. Based in Fairfax, VA.

• **ISO 9000 Role** – Maintains records of U.S. companies that have been ISO 9000-registered, as well as names of ISO 9000 registrars operating in the U. S.

• **Status** – For-profit publishing and seminar house. Has published the works of U. S. RAB Chairman Robert Peach (an ISO 9000 consultant), and other well-known ISO 9000 consultants. Will offer information to the media *gratis*, but sells information to the general public.

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industry, for example, ISO 9000 is fast becoming a condition of doing business with a giant like DuPont. And there have been recent reports that General Electric ordered all suppliers to become ISO 9000 certified if they want to continue to do business with GE.

What are companies to make of these kinds of contradictory messages?

Industry leaders and experts in the ISO 9000 field recommend companies keep an eye on their industry leaders (see chart). If industry leaders and heads of industry associations are advising against ISO 9000 certification, the best course is to learn as much about the process as possible while sitting tight.

For companies serving several industries, it may be necessary to keep abreast of ISO 9000 developments in all industries where you have a customer base. If one or more of these industries is pushing ISO 9000, you should be prepared to enter the certification process. At the very least, conduct a baseline audit of your company to determine where you stand vis-a-vis the ISO 9000 standards. Find a registrar that suits your company's needs and be prepared to hire him or her if and when the pressure is on to seek certification.

Not all companies will need to become ISO 9000 certified. Companies that are regional in nature, with no connection to major corporations or exporting, may never feel the pressure to become ISO 9000 certified.

Controlling Your ISO 9000 Destiny

Good upfront organization and planning as well as employee involvement are key to savings in any quality program. But there are other ways to shave thousands of dollars from the ISO 9000 process. The marketplace is responding to cost-saving pressures; some international registrars facing mounting competition are bowing to lower cost demands. Some large customers have financially assisted small suppliers in obtaining certification.

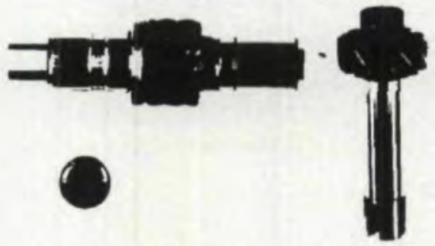
Companies of all sizes should negotiate carefully with potential registrars and consultants. Question those who say or imply that "A quality company should pay whatever it takes to become certified." Learn from companies that have been certified; ask what their costs were; where they felt they could have saved money. Ask registrars and consultants what portion of their incomes are derived from ISO 9000 services.

Be aware that ISO 9000 as implemented in today's world is big business – but it is not oriented towards improving any manufacturers's bottom line. It is up to the aspiring company to make such certification worthwhile. ■

For information on and copies of the ISO 9000 standards, contact ANSI at (212) 642-4900.

Investigation of Surface Layer and Wear Behavior of Nitrided Gear Drives

Claus Razim, Mercedes Benz AG
Stuttgart, Germany



	Intermediate Gear	Helical Gear
Standard Treatment	driving gear C15 BNT	driven gear C15 BNT
No. of Teeth	9	12
Max Speed	4300 rpm	3250 rpm
Sliding Velocity	approx 1.6 m/sec at 1000 rpm	
Theoretical Surface Pressure Load	3250 — 4600 N/mm^2 at 1000 rpm	
Power Transmitted	4.3 kW (5.9 PS) max.	

Fig. 1 — Investigated pair of gears.

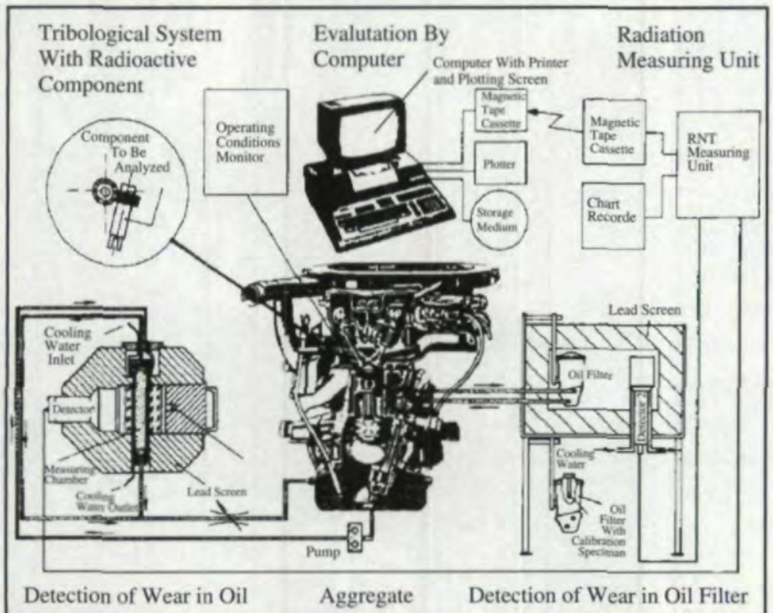


Fig. 2 — Wear measurement by means of radionuclide technology.

In this article we will characterize the nitride layers that are generated by different nitriding processes and compare their respective wear characteristics.

Test Apparatus

Fig. 1 shows a pair of gears on which the wear measurements were carried out: an intermediate gear and a helical gear, which drive the oil pump of a combustion engine. These gears were nitrided by different methods. Fig. 1 also gives some typical data on the gears in question. The components have a high sliding velocity and a very high theoretical surface load. The wear of the helical gear used here has been measured by means of radionuclei technology (RNT). In the process not less than two helical gears ready for installation were activated by neutron radiation. The wear particles of such a radioactive gear can be measured very accurately by measuring its gamma radiation and converting it into wear rates, following previous calibration.

Fig. 2 shows the working setup of the RNT measuring apparatus. In this test, a crankcase containing an oil pump and helical gear drive was set up. An individual oil pump was used in place of a complete engine. The intermediate gear shaft is driven directly by a D.C. motor. The load on the system is controlled by varying the diameter of the oil supply line. Speed and oil temperature can also be controlled. If the helical gear develops wear during the test run, the radioactive wear particles pass into the lubrication oil circuit. Some particles are deposited in the oil filter and some are retained in the oil. The amount of wear is measured by two detectors; one in the oil and one in the filter. The total wear is the sum of the two amounts so measured.

Test Program

Fig. 3 shows the test program. The top of the illustration shows the speed of the test, the load of the gears and the temperature. The lower part of the chart shows the test program and the typical wear curves. The test program is composed of three main parts:

a. A "run-in," that is, the first step at the beginning of the curve;

b. The "main run," going up to 94 hours; and

c. The "overload run" of over 21 hours.

The three parameters controlled were speed, load and oil temperature. The test program was selected so that a standard helical gear would only just survive, and that a distinct separation of the good from the bad helical gear sets could be expected. The main parts of the test program are generally reflected in a typical wear curve. Following a rapid rise during the "run-in" at the beginning of the test, only gradual increases will result during the "main run" and the curve will flatten. However, during the overload run, wear will sometimes rise catastrophically. Even though the wear particles in the oil decrease because of the filtering system, the sum of the amounts of wear particles increases. A number of different gears, 6 nitrided by different processes, were subjected to the test program.

Table 1 lists the treatment methods that have been analyzed. These include six bath nitride processes (BNT), five short cycle gas nitriding processes (KNT) and three plasma nitriding processes (PNT) of different manufacturers. Steel C15, identical to SAE 1015, was exclusively used as a basic material. In addition, the wear characteristics of gear sets made of case hardened steel (E) with carbon content of 0.16 and chromium content and manganese content of approximately 1% and lower were measured for comparison. Metallographic examinations were performed on all variants.

(Figs. 4, 4A) The thickness of the compound layer was 18 ± 4 micrometers. The porosity, the diffusion zone, the structure, the microhardness gradient and the measurement of surface roughness were recorded as criteria of the test or service condition. No correlation was established between hardness, surface roughness or wear characteristics. A specific amount of wear can be correlated to each part of the test; that is "run-in," "main run" and "overload run." Total wear was determined by checking residue in the oil and in the filter. However, the total wear is not the only decisive factor; the wear rate must also be considered. It is possible to compare different treatment methods by comparing the amount of wear and wear rates at the same point in time. To make this comparison possible, a representative total wear curve is given for each variant.

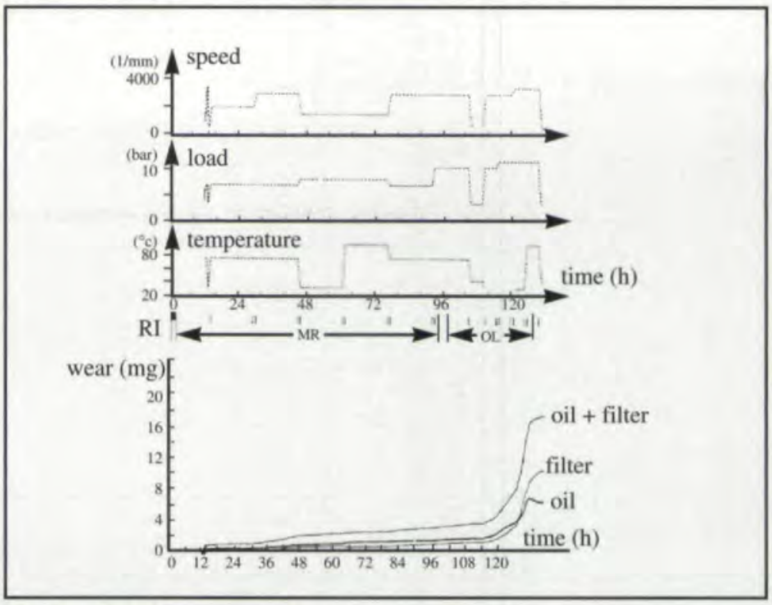


Fig. 3 — Test program and wear curve.

Table 1 — Treatment Variants

No.	Treatment		Material
1	BNT	Salt bath high in cyanide/water	C 15
2	BNT	Salt bath low in cyanide /water	C 15
3	BNT	Salt bath low in cyanide /cooling bath 310° C	C 15
4	BNT	Salt bath low in cyanide /nitriding temperature 610° C	C 15
5	KNT	1.NH3 + exogas 2.NH3 + endogas	C 15
6	KNT	NH3 + methylamine	C 15
7	KNT	NH3 + exogas	C 15
8	KNT	NH3 + endogas	C 15
11	KNT	NH3 + exogas	C 15
12	BNT	Salt bath low in cyanide/water	C 15
9	PNT	Mixed phase	C 15
10	PNT	Monophase γ'	C 15
2/3	PNT	Mixed phase	C 15
2/5	BNT	Salt bath low in cyanide with some ppm S addition	C 15
2/4	E	Case-hardened	16MnCr5

Prof. Dr.-Eng. Claus Razim

is Senior Vice President of Technology and Environment and Chief Enviornment Officer for Mercedes-Benz AG in Stuttgart.

- Determination of Thickness of
  - Compound Layer CL
  - Porosity P
  - Diffusion Zone DZ

  aim CL 18 ± 4 μm
  with PNT variants CL 7-12 μm only
- Structure
- Progression of hardness HV.5 in DZ
- Measurement of surface roughness
  - R_a, R_z, R_{3z}

Fig. 4 — Metallographic examinations.

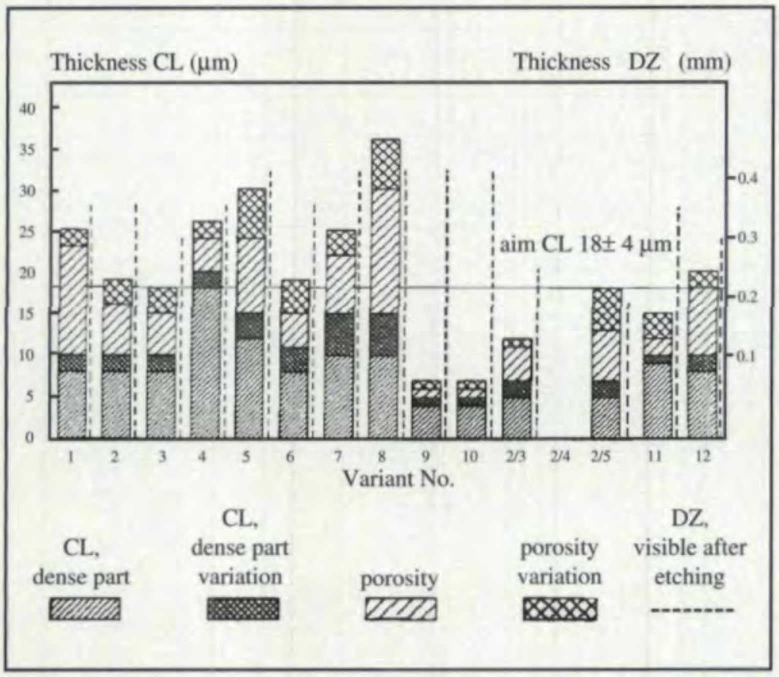


Fig. 4a — Comparison of variants' compound layer, porosity and diffusion zone.

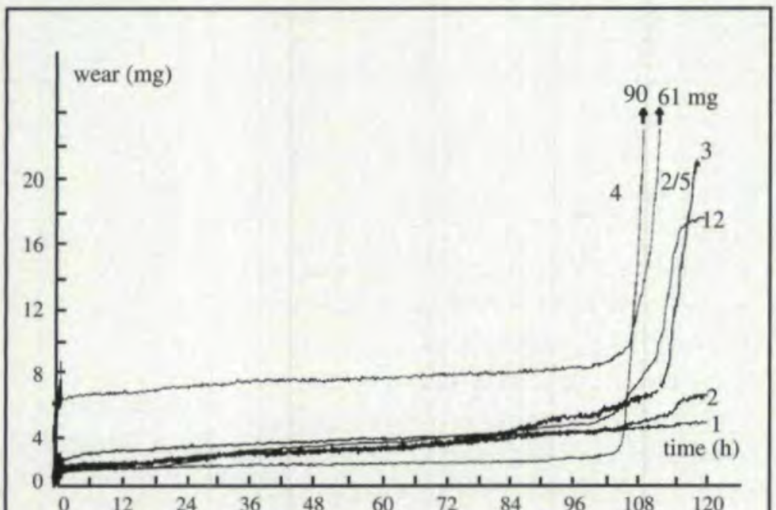


Fig. 5 — Various bath nitriding processes' (BNT) wear characteristics.

Test Results

Fig. 5 shows the curves for all the BNT processes. Up to the end of the main run the wear rates of all variants, with the exception of 2/5 (which has some added sulphur), are within a very narrow range. Only the overload reveals a distinct difference. Variant 1 is the only variant that shows no increased wear during the overload; all other BNT variants have a higher wear rate during this period. The layer developed with the addition of sulphur (2/5) has a high run-in wear and a steep rise in wear during the overload run, but shows an acceptable wear rate during the main run. If we classify the loading to the end of the main run as being medium, all BNT layers over this range are approximately on the same level. For high loads, as in the overload run, the layer developed in the salt bath that was high in cyanide/water (shows the best results).

Fig. 6 shows the least favorable curves for short cycle gas nitrided (KNT) parts. The total oil and filter wear has again been plotted. It is evident that the differences between the various short cycle gas nitride (KNT) processes are much greater than those between the BNT processes. Variant 5 shows the best results, and is very similar in its wear characteristic to Variant 1, the salt bath high in cyanide/water. With the other KNT variants, wear increases catastrophically sooner or later. With Variant 11 this increase in wear appears in the early stage. Variants 7 and 11 are actually the same process.

Fig. 7 shows the wear characteristics of the plasma nitriding process (PNT), the ion-nitriding variant. In this process the thickness of the compound layer is only 7-12 μm – that is, one-half the thickness of the BNT or KNT layers. According to the manufacturer the thinner layer should guarantee a more porous free layer. With the BNT layers, wear increases at early stages during the main run. Level and rise are similar to those of the least favorable PNT variant. The curves of the different PNT variants show rather similar progressions. The variant with the highest proportion of epsilon phase, that is the variant 2/3 shows the least wear at the end of the main run. The wearing characteristics of the analyzed variant can be compiled in a working sketch and classified with the scope of the test program used here. From the experience gained during vehicle test runs only two variants, that is the BNT Variant 1 and the KNT Variant 5 can be judged as "good." Variants surviving the main run with low wear, and of medium quality, were found to be the BNT variants 2, 3, 4 and 12 and 2/5, as well as the KNT variants 7 and 8. The case-hardened variant is not on the same level, since it showed a very low wear rate during the main run, despite its generally overall high wear level.

Variants that show a rapid rise in wear during the main run must be classified as poor (see Fig. 8). The KNT variants 6 and 11, as well as all PNT variants with the analyzed layer thickness, fall into this category. The wear characteristics of the case-hardened helical gears are basically different from those of the nitrided gears. During the run-in, when the nitrided layers show a wear of approximately 4 milligrams, the case-hardened versions show a wear as high as 80 milligrams, so in the subsequent main run, the case-hardened gears show hardly any wear. At less than 5 milligrams, the main run wear is comparable with that of the good nitrided variant. However, during the overload run one of the two gears shows a catastrophic increase in wear. Case-hardened parts are very sensitive to the addition of sulphur components, for example, zinc thiophosphate.

Damage Analysis

Let us assess the analysis of the damage. All the nitrided helical gears show a similar damage pattern at the end of the test run. There is a pattern of different widths with pittings and more or less large breakouts. The large breakouts frequently go down to the basic material as has been shown by means of spot analysis with copper ammonium chloride. Breakouts may even exist after the run-in. To obtain information on the defect mechanism, test runs were carried out with no activated gear sets of the variants 1, 11 and 12 followed by scanning electron-microscopic examinations and metallographic sections made through the cracks.

Fig. 9 shows a general view of one tooth of each of the three variants. The top illustration records Variant 1, the salt bath high in cyanide/water (BNT), the center represents Variant 12, the salt bath low in cyanide/water (BNT) and the bottom a short cycle gas

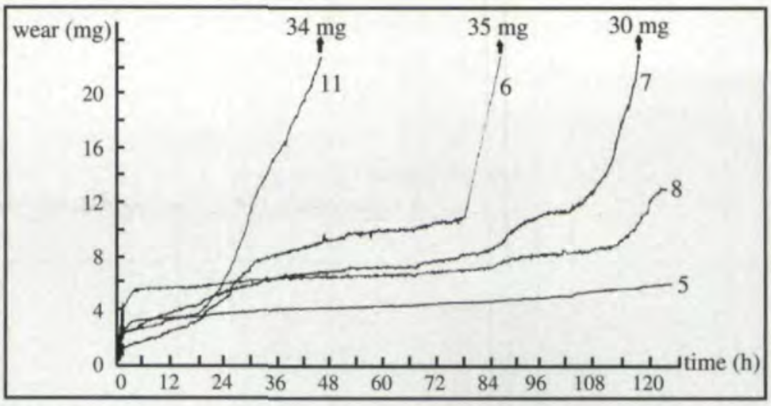


Fig. 6 — Various gas nitriding processes' (KNT) wear characteristics.

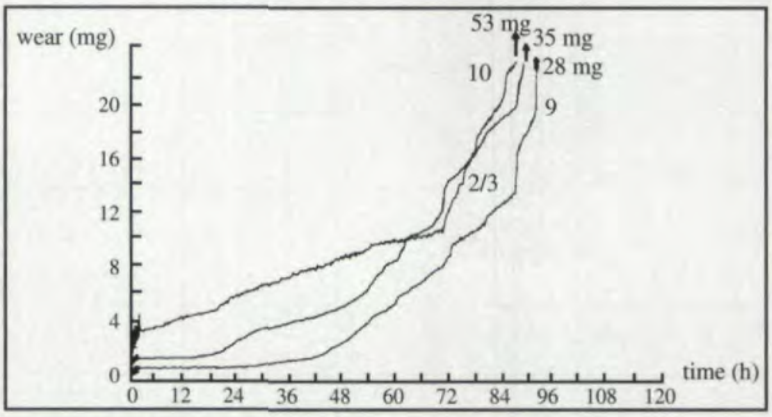


Fig. 7 — Various plasma nitriding processes' (PNT) wear characteristics.

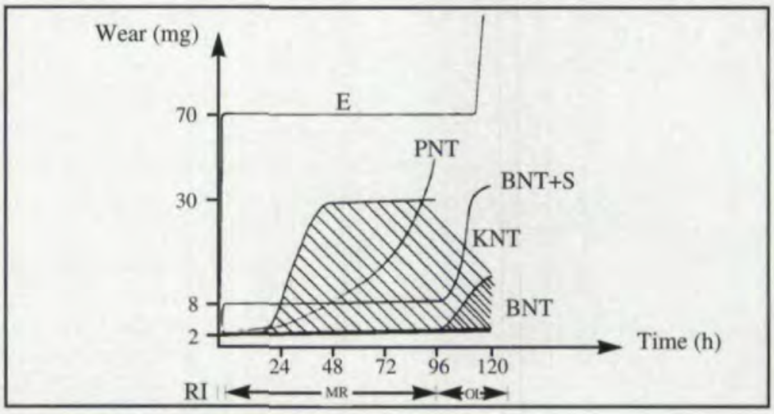


Fig. 8 — Wear characteristics, composite.

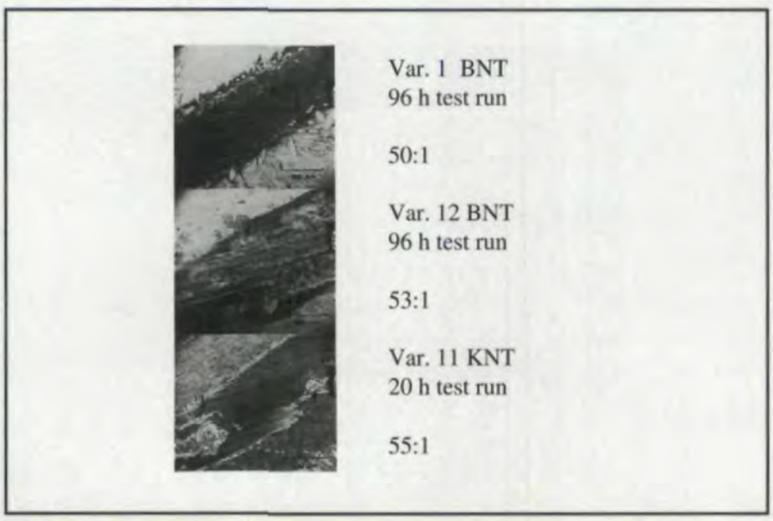


Fig. 9 — General view of wearing zones.

nitrided part. The run-in time was shorter for the gas nitrided gear (KNT) than that for the salt bath nitrided gears (BNT).

Fig. 10 shows this area with greater magnification. On Variant 1 the pittings are completely within the porous zone and only light fracture lines are visible in this area of breakouts. On Variant 12 (Fig. 11) the porous zone has broken out. The destruction extends into the dense portion of the compound layer. The photo of Variant 11 has a magnification of only half that of the other variants, and yet the surface breakouts are of greater magnitude. The damage here extends down to the diffusion zone. The same situation exists on the metallographic section perpendicular to the wear face.

Var. 1 BNT
96 h test run

200:1

Var. 12 BNT
96 h test run

210.1

Var. 11 KNT
20 h test run

220:1

Fig. 10 — Details of wearing zones.

Var. 1 BNT
96 h test run

2000:1

Var. 12 BNT
96 h test run

2100:1

Var. 11 KNT
20 h test run

1100:1

Fig. 11 - Details of wearing zones.

Var. 1 BNT
96 h test run

1000:1

Fig. 12 — Helical gear after test run. Cross section of wearing zone.

In Fig. 12, photographs of Variant 1 show cracks within the porosity zone. Breakouts occur within this zone. In Fig. 13, photographs of Variant 12 show large areas of subsurface cracking and crack diversification. Similar photographs of Variant 11 (Fig. 14) show pronounced crack diversification, cobblestone-like break outs and cracks in the base material. The cracks generated through a pitting formation are stopped by the extensive porosity zone of Variant 1, in contrast to Variants 11 and 12. The dense layer beneath the porous zone of Variant 1 generates greater resistance to crack propagation than that of Variants 11 and 12.

X-ray diffraction tests were carried out to determine the face composition. The results are shown in Table 2. The readings were taken with molybdenum radiation, allowing for analysis to a depth of approximately 50 μm. Iron-nitride, gamma prime and epsilon iron nitride, as well as amorphous carbo-nitrides, are the major phases occurring over this range. In addition, lines of iron oxides, and, depending upon the thickness of the compound layer, lines of the basic material ferrite, may appear. In practice, the compound layer generally consists of a mixture of epsilon and gamma prime phase in proportions that may vary greatly. However, quantifying this phase mixture is difficult because the phases are not homogeneously distributed in the compound layer, and vary as the depth increases. For a proper interpretation, the compound layer would have to be successively removed.

Since there were two phase examinations performed on the original component, there were two measuring points available for the qualitative analysis. The first was on the tooth of the helical gear, the original layer, and second on the shaft of the helical gear with case removed. The readings taken on the tooth show very similar diffraction diagrams in practically all cases. Over the analyzed area the layer essentially consists of epsilon phase. Variant 10 is an exception, since it consists of gamma prime mono-phase. Only Variants 2, 3, 9 and 2/3 show distinct gamma prime lines. Variants 4, 7 and 11 show only slight traces. Variants 3 and 11 show distinct iron oxide lines because of the thickness of the compound layer. The alpha iron lines of the basic material are distinctly visible on the PNT layers of Variants 9, 10 and 2/3 only. The readings of the shaft will show more distinct differences between the layers of the different variants. The proportion of the gamma prime phase increases as expected. and is greater in the deeper layers of the compound zone. According to the readings taken, Variants 1, 5, 11 and 12 have the lowest gamma prime proportion.

Variants 1, 11 and 12 were subjected to Auger examinations. They provide the depth profile of the concentration of elements in the compound layer. The Auger examinations were carried out by continuous sputtering of the compound layer using argon ions, rather than on metallographic section perpendicular to the compound layer.

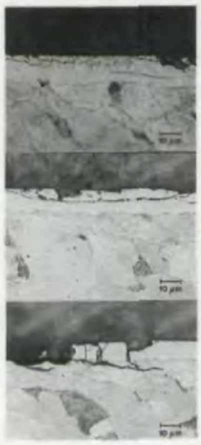
Fig. 15 shows the concentration profiles of the three variants. The nitrogen concentration profiles are similar for all specimens. Major differences are found in the progression of the oxygen and carbon concentration. Variant 11 has a pure iron oxide layer to a depth of 0.6 µm. This is magnetite. However, with a depth of greater than 0.6 µm these specimens have a very low oxygen content, which is distinctly below that of the two BNT variants. Since, as is well known, magnetite forms a brittle layer, this layer may affect the wear characteristics despite the fact that it is relatively thin.

The carbon content of variant 11 is approximately 6-12% and decreases with decreasing nitrogen content towards the matrix material. By contrast, the BNT variants have an increasing carbon content. This effect is particularly striking on the Variant 1, where the carbon content comes up to the level of nitrogen concentration at a depth of approximately 10 µm. Properly speaking, all three variants have, in fact, a carbo-nitride/oxy carbo-nitride layer. The total sum of the oxygen plus carbon plus nitrogen concentrations on Variant 11 is the most stable as

Var. 12 BNT
96 h test run

1000:1

Fig. 13 — Helical gear after test run. Cross section of wearing zone.



Var. 11 KNT
20 h test run

1000:1

Fig. 14 — Helical gear after test run. Cross section of wearing zone.

Table 2 — X-ray diffraction measurements of compound layer.

Examinations at 2 points on original component

1. on tooth = original surface

2. on shaft = ground surface.

Results:

1. on tooth:
- All variants highly similar
- Mainly ε-phase with the exception of Variant 10
- Var. 2, 3, 9 and 2/3 distinctly γ'
- Var. 3 and 11 distinctly Fe_3O_4

2. on shaft:
- Difference between different variants, γ' proportion increasing
- Variants 1, 5, 11 and 12 have low proportion of γ'-phase

γ'-proportion increases with depth

a function of depth.

Because of the low carbon solubility of the gamma prime phase, the high carbon content of Variant 1 obstructs the formation of the gamma prime. This would also be consistent with the results of the x-ray diffraction tests. The mechanical properties of the epsilon phase, in particular, depend on the level of carbon content in the layer. According to the available literature, the hardness of the epsilon phase decreases rapidly as the proportion of carbon increases. The hardness values of the epsilon phase are generally below those of the gamma prime phase. Hence, it is readily conceivable that an increasing carbon content at depth will cause a continuous transition of the mechanical properties of the compound layer and the mechanical properties of the diffusion layer.

Now, assuming that the ductility (the deforming capability of the layers) is an inverse function of the hardness of the compound zone, then the carbon-containing epsilon compound layer would be the most ductile of all possible layers, that is, the layer with the lowest tendency to brittle crack formation. However, there are also contradictory statements concerning the ductility of the epsilon/gamma prime phase.

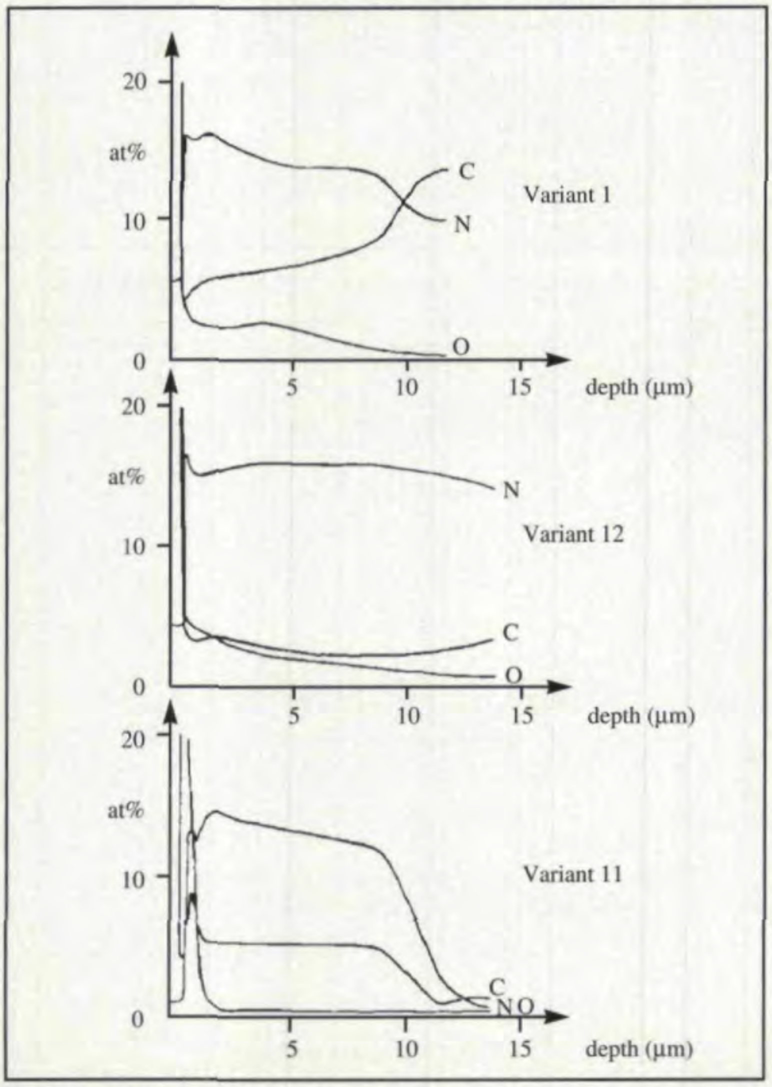


Fig. 15 — Helical gear after test run. Cross section of wearing zone.

Conclusion

In conclusion, we have analyzed the wearing characteristics of nitrided layers made by different processes and have achieved a good classification of the various methods involved. To arrive at a relationship between layer structure and wear characteristics, we have looked more closely into one nitriding variant, each with good, medium and poor wearing characteristics. The nitriding variant with good wear characteristics has the following features: A high proportion of porosity, a very low portion of gamma prime phase (that is, high homogeneity of the compound layer), a rapidly increasing carbon content of the compound layer with increasing depths and a relatively constant oxygen plus nitrogen plus carbon total content over the compound layer. If all these features are present, we believe that pitting is avoided or hindered. The porosity helps to promote good "running-in" characteristics. The homogeneous monoface layer structure and the increasing ductility toward the basic material helps to reduce pitting in the compound layer.

Considering the results of the wear analysis, it would appear that a nitride layer with the properties just mentioned would be easier to obtain by a bath type nitriding than by short cycle gas nitriding. The variants subjected to short cycle gas nitriding revealed greater variation, suggesting that the direction of the treatment parameters plays a more important role. Nevertheless, as has been proved by Variant 5, gaseous short cycle nitriding is quite capable of producing the nitride layer, and has properties equivalent to those obtained in a salt bath. However, our experience tell us that currently the salt bath treatment still has a distinct lead in regard to accuracy of control. ■

Author's note: This research was carried out in cooperation with the companies Robert Bosch and Degussa, within the scope of two projects sponsored by the German Federal Ministry of Research and Technology.

ADVERTISERS INDEX

Stress of Planet Gears with Thin Rims

Takeshi Ishida & Teruaki Hidaka
Yamaguchi University, Yamaguchi, Japan

Abstract

This article discusses the relationships among the fillet stress σ on a thin rim planet gear, the radial clearance between the gear rim and the gear shaft Δr, the tooth load P_n, the rim thickness h, the radius of curvature of the center line of the rim r, the face width b and the module m. It is shown theoretically that $P_n/(mb)$, $(h/r)(\Delta r/r)$ and $(h/m)(h/r)$ can express the tooth load, the radial clearance and the rim thickness, respectively. Furthermore, when the tooth loads are applied at the tips of a thin rim planet gear which is cut by a rack-type cutter with a pressure angle of 20° and a tip radius of 0.375 m, the maximum tensile and compressive stresses, by which both the amplitude and the mean value of the alte-rnating fillet stress can be estimated, are analyzed by the finite element method. They are expressed by equations using the expressions $P_n/(mb)$, $(h/r)(\Delta r/r)$, $(h/m)(h/r)$ and the number of teeth z.

Introduction

The fillet stress σ on a gear with a thick rim is proportional to the ratio of the tooth load to both the module and the face width, $P_n/(mb)$. Therefore, the fillet stress σ on the gear with a thick rim is frequently discussed when the ratio $P_n/(mb)$ is constant. The fillet stress on the gear with a thin rim is much different because of the elastic deformation of the rim (Refs. 1-6). The stress on the planet gear with a thin rim is influenced by the rim thickness and the radial clearance between the rim and the gear shaft (Refs. 3-6). Some dimensionless expressions have been used to show the relative rim thickness and the relative radial clearance. For the rim thickness, either the ratio of the rim thickness to the module *(h/m)* or the ratio of the rim thickness to the radius of curvature of the center line of the rim *(h/r)* is often used (Refs. 3-6). For the radial clearance, either the ratio of the radial clearance to the module *(Δr/m)* or the ratio of the radial clearance to the radius of curvature of the center line of the rim *(Δr/r)* is often used (Refs. 3-6). However, the effects of these ratios, *h/m, h/r, Δr/m* and *Δr/r*, on the fillet stress have not been discussed sufficiently.

This article discusses the relationships among the fillet stress σ, the radial clearance Δr, the tooth load P_n, the rim thickness h, the radius of curvature of the center line of the rim r, the face width b and the module m. It is shown in theory that $P_n/(mb)$ can express the tooth load, $(h/r)(\Delta r/r)$ can express the radial clearance, and $(h/m)(h/r)$ can express the rim thickness when the fillet stress is expressed by σ. In addition, the maximum stresses on the tension and compression side fillets of the loaded tooth, and the maximum stress on the fillet near the position where the rim receives the reaction force from the gear shaft are analyzed by the finite element method. Then the maximum stresses on those fillets, σ, (by which both the amplitude and the mean value

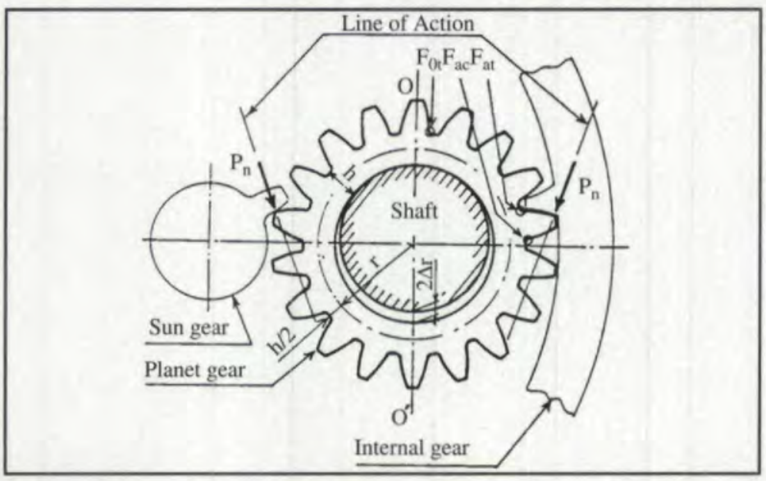


Fig. 1 — Schematic view of the planet gear.

of the alternating fillet stress can be estimated) are expressed with equations using the expressions $P_n/(mb)$, $(h/r)(\Delta r/r)$, $(h/m)(h/r)$ and the number of teeth z. This analysis is made in a particular case when tooth loads are applied at the tips of a thin rim planet gear which is cut by a rack-type cutter with a pressure angle of 20° and a tip radius of 0.375 m.

Planet Gears Discussed

Fig. 1 shows the schematic view of a planet gear with a radial clearance between the gear rim and the gear shaft. In the case of the planet gear, it has been shown that both the amplitude and the mean value of the alternating fillet stress can be estimated by using the maximum stresses on the tension and compression side fillets, shown by F_{at} and F_{ac} in Fig. 1, and the maximum stress on the fillet near the position where the reaction force is applied from the gear shaft, shown by F_{0t} in Fig. 1 (Ref. 4). Therefore, these three maximum stresses on F_{0t}, F_{at} and F_{ac} were analyzed in this study.

The pressure angle and the whole depth of the planet gears discussed were 20° and 2.25 m (m:module) respectively. The gears were assumed to be cut by a rack-type cutter with a tip radius of 0.375 m. Both the number of teeth and the rim thickness of the gears analyzed are shown in Table 1.

An example of the mesh pattern of the finite element model of the planet gear is shown in Fig. 2. The mesh pattern per one pitch of any planet gear model was almost the same as that shown in Fig. 2.

The longitudinal elastic modulus and the Poisson's ratio of the gears were assumed to be 206 GPa and 0.3, respectively. The gear shaft was assumed to be rigid because the contact stiffness between the planet gear rim and the gear shaft was considered to be much larger than the bending stiffness of the thin rim of the planet gear. The ratio of the tooth load to both the module and the face width, $P_n/(mb)$, was chosen as 124 N/mm^2.

Theoretical Discussion of Expressions of Fillet Stress, Tooth Load, Rim Thickness and Radial Clearance

Fillet stress on the loaded tooth of a thin rim planet gear with a large radial clearance must be approximately expressed as follows:

$$\sigma = \sigma_T + (\sigma_{Rb} + \sigma_{Ra}), \qquad (1)$$

Table 1 — Specifications of the planet gears discussed.

Number of teeth z	$(h/m)(h/r)$		
18	1.4 (0.2)	2.0 (0.3)	3.1 (0.5)
38	2.2 (0.13)	2.6 (0.16)	3.2 (0.19)
	4.0 (0.25)	4.9 (0.32)	
62	2.8 (0.1)	4.1 (0.15)	6.4 (0.24)
90	3.4 (0.08)	5.0 (0.12)	7.9 (0.2)

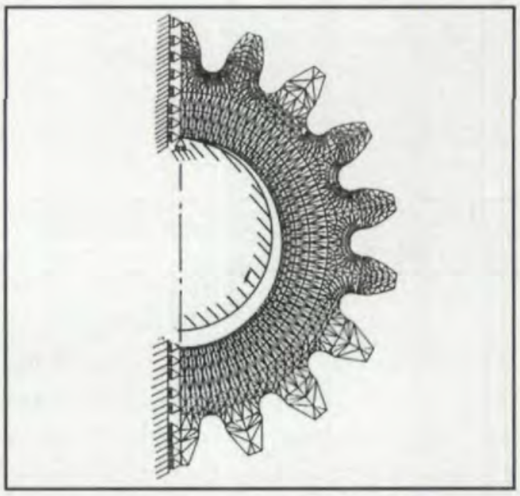


Fig. 2 — A mesh pattern of the planet gear model (z = 18, $(m/h)(r/h)$ = 0.645).

where σ_T shows the fillet stress of the loaded tooth when the effect of the rim stress is very small. σ_{Rb} and σ_{Ra} show the stresses caused by the bending moment of the rim and by the circumferential force of the rim, respectively. σ_T can be put as zero on the fillet of the tooth to which the tooth load is not applied. Since σ_{Rb} must be greater than σ_{Ra}, σ_{Ra} can be neglected in the case of the thin rim gear.

The distance to the center line of the rim from the intersection between the line of action and the center line of the tooth form is η. σ_{Rb} can be expressed as follows by using coefficients a_1 and a_2:

$$\sigma_{Rb} = a_1\{P_n r/(bh^2)\} + a_2\{P_n \eta/(bh^2)\}. \qquad (2)$$

In the case of a thin rim gear, the radius of curvature of the center line of the rim r is generally much larger than the distance η, i.e., $r >> \eta$. Eq. 2 can be rewritten as follows using $[P_n/(mb)]$, which is used in the case of the gear with a thin rim:

$$\sigma_{Rb} = a_1\{P_n/(mb)\}(m/h)(r/h). \qquad (3)$$

From Eq. 3, σ_{Rb} can be supposed to be approximately proportional to $(m/h)(r/h)$ when

Dr. Takeshi Ishida
has been an Associate Professor in the Department of Mechanical Engineering of Yamaguchi University since 1987.

Dr. Teruaki Hidaka
has been a Professor of Mechanical Engineering at Yamaguchi University since 1977.

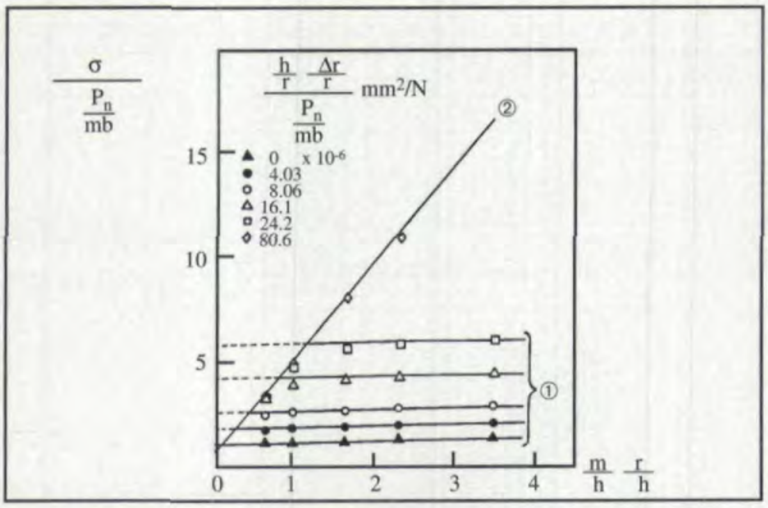


Fig. 3 — Relation between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$ $(z = 38)$.

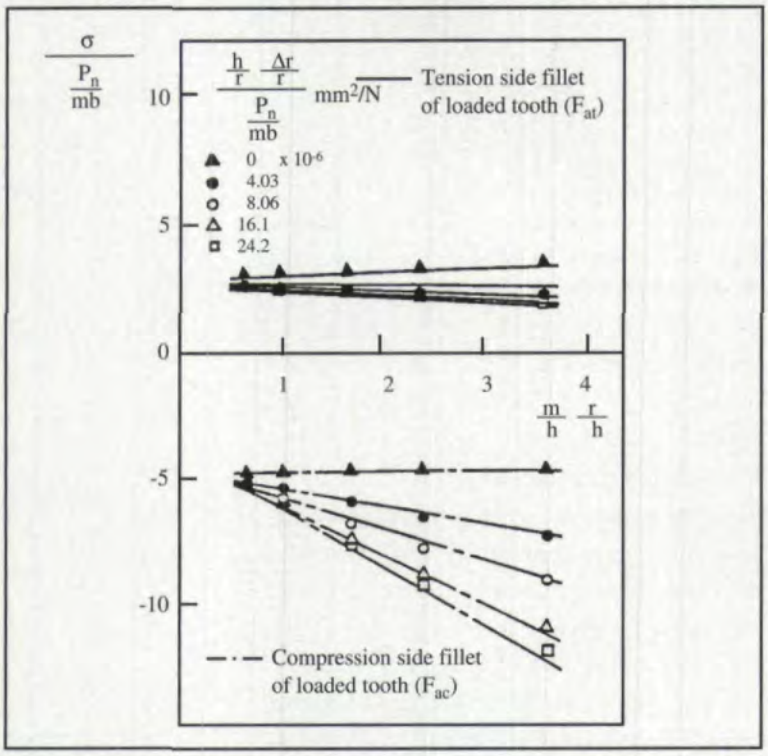


Fig. 4 — Relations between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$, and between $\sigma_{ac}/\{P_n/(mb)\}$ on F_{ac} and $(m/h)(r/h)$ $(z = 38)$.

$[P_n/(mb)]$ is constant.

On the other hand, σ_T can be expressed by using the coefficient b_1 as follows:

$$\sigma_T = b_1(P_n/mb). \quad (4)$$

Therefore, Eq. 1 can be expressed approximately as follows:

$$\sigma = \{P_n/(mb)\}\{b_1 + a_1(m/h)(r/h)\} \quad (5)$$

When the radial clearance between the gear rim and the gear shaft decreases, the contact area between the gear rim and the gear shaft changes from line contact to face contact. Therefore, when discussing the fillet stresses on the planet gear with both a thin rim and a small radial clearance, the displacement of the gear rim must be considered.

The case where the radial clearance is very large and only line contact occurs is considered before the case of face contact. The displacement of the gear rim δ can be approximately expressed as follows because of $r >> \eta$:

$$\delta = c_1 P_n r^3/(bh^3) = c_1\{P_n/(mb)\}(m/h)(r/h)^2 r. \quad (6)$$

Then the following equation is reduced:

$$(h/r)(\delta/r) = c_1\{P_n/(mb)\}(m/h)(r/h). \quad (7)$$

Eq. 7 shows that when tooth loads are applied to any planet gear whose $(m/h)(r/h)$ are equal to one another under the condition where $P_n/(mb)$ is equal, $(h/r)(\delta/r)$ are the same, and the rim stress reaches σ_{Rb} as shown in Eq. 3. Therefore, when $(h/r)(\Delta r/r)$ is larger than $(h/r)(\delta/r)$, line contact between the rim and the gear shaft occurs, and the rim stress is proportional to $P_n/(mb)$.

Next, the case where $(h/r)(\Delta r/r)$ is less than $(h/r)(\delta/r)$ will be considered. On F_{0t} (shown in Fig. 1), as the radius of curvature of the rim changes from r to $(r - \Delta r)$, the following equation can be obtained:

$$1/(r - \Delta r) - 1/r = M/(EI), \quad (8)$$

where M, E and I denote the bending moment, the longitudinal elastic modulus and the moment of inertia of the cross section of the rim, respectively. Because $1/(r - \Delta r) - 1/r \approx \Delta r/r^2$ and M is proportional to σh^2, Eq. 8 becomes:

$$\sigma \text{ is proportional to } (h/r)(\Delta r/r). \quad (9)$$

It is clear from Eq. 9 that the fillet stress on F_{0t} is not a function of $(m/h)(r/h)$, but a function of $(h/r)(\Delta r/r)$. After the face contact between the rim and the gear shaft at the position near F_{0t}, face contact on other regions does not always occur. However, even in such cases, since $(h/r)(\delta/r)$ of the rim must be proportional to $(h/r)(\Delta r/r)$, the fillet stress in the region near which the face contact does not occur seems to be expressed by an equation similar to Eq. 5.

Approximate Expression of Maximum Fillet Stresses

Relation between the stress σ and $P_n/(mb)$. For planet gears whose number of teeth z and $(m/h)(r/h)$ are equal to each other, when $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ is constant, the stress σ is proportional to $P_n/(mb)$ (Ref. 3). Therefore, in the case of the finite element analysis under $P_n/(mb)=124\ N/mm^2$, and denoting by $(h/r)(\Delta r/r)_{124}$ and σ_{124} the dimensionless radial clearance and the stress, respectively, when

$$(h/r)(\Delta r/r)/\{P_n/(mb)\} = (h/r)(\Delta r/r)_{124}/124 \quad (10)$$

is satisfied, the following equation must be satisfied.

$$\sigma/\{P_n/(mb)\} = \sigma_{124}/124 \quad (11)$$

Considering Eqs. 10 and 11, $(h/r)(\Delta r/r)_{124}/124$ and $\sigma_{124}/124$ are translated into $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ and $\sigma/\{P_n/(mb)\}$ to express the approximate equations.

Relation between $\sigma/\{P_n/(mb)\}$ ***and*** $(m/h)(r/h)$**.** The relations between the maximum stresses on the fillets F_{0t}, F_{at} and F_{ac}, and $(m/h)(r/h)$ with a parameter $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ were analyzed by the finite element method as shown in Figs. 3 and 4 where the number of teeth is $z = 38$. In Figs. 3 and 4, the stresses are translated into $\sigma/\{P_n/(mb)\}$. The relationship between $\sigma/\{P_n/(mb)\}$ and $(m/h)(r/h)$ with a parameter $(h/r)(\Delta r/r)/\{P_n/(mb)\}$, where the number of teeth are both $z = 18$ and 90, was also analyzed and shown in Figs. 5-8. Although the stresses on the planet gear with the number of teeth $z = 62$ were also analyzed, these results are not shown here. Symbols in Figs. 3-8 show the values analyzed by the finite element method, and the lines in Figs. 3-8 show the values obtained by the approximate equations mentioned below. The value of $80.6 \times 10^{-6}\ mm^2/N$ for $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ in Figs. 3, 5 and 7 corresponds to an extremely large radial clearance. It seems from these figures that the stress on F_{0t} when $(h/r)(\Delta r/r)/\{P_n/(mb)\} = 80.8 \times 10^{-6}\ mm^2/N$ is proportional to $(m/h)(r/h)$. This phenomenon must correspond to Eq. 3. Moreover, although Eq. 9 shows that the stress on F_{0t} when $(h/r)(\Delta r/r)/\{P_n/(mb)\} = 80.8 \times 10^{-6}\ mm^2/N$ does not change with $(m/h)(r/h)$, the stress changes slightly with $(m/h)(r/h)$ in Figs. 3, 5 and 7 owing to the circumferential force of the rim. Because it is clear from Figs. 3-8 that the relations between $\sigma/\{P_n/(mb)\}$ and $(m/h)(r/h)$ on each fillet can be approximately expressed by straight lines, the relation between $\sigma/\{P_n/(mb)\}$ and $(m/h)(r/h)$ is expressed as follows:

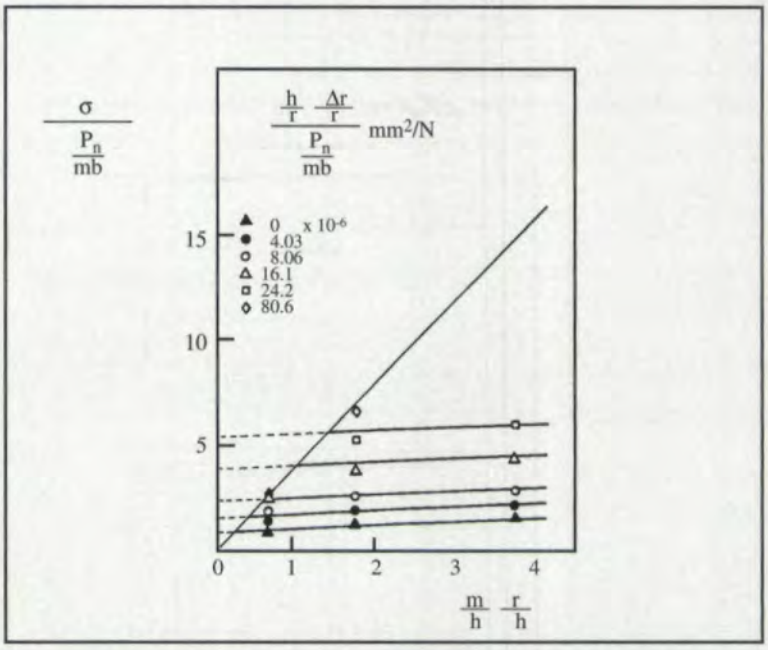


Fig. 5 — Relation between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$ ($z = 18$).

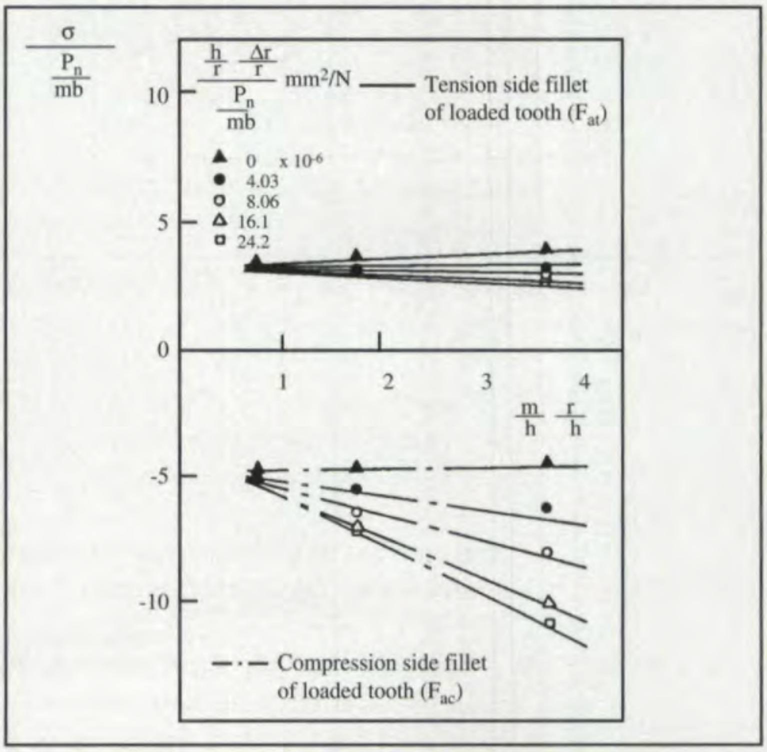


Fig. 6 — Relations between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$, and between $\sigma_{ac}/\{P_n/(mb)\}$ on F_{ac} and $(m/h)(r/h)$ ($z = 18$).

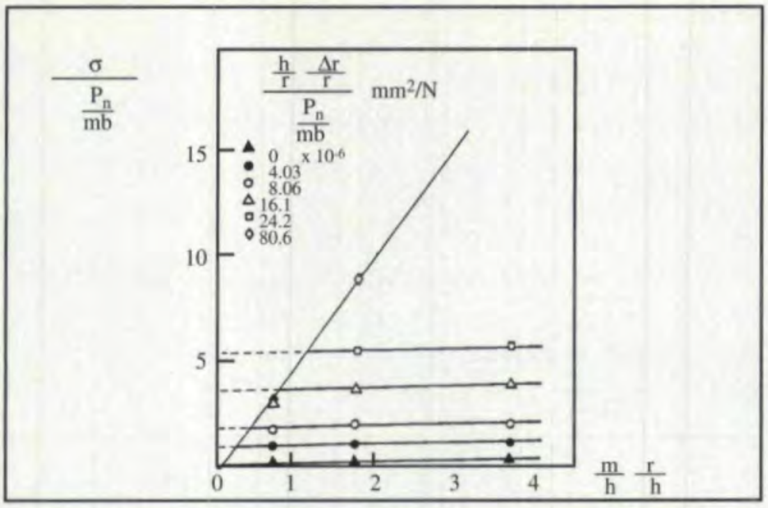


Fig. 7 — Relation between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$ $(z = 90)$.

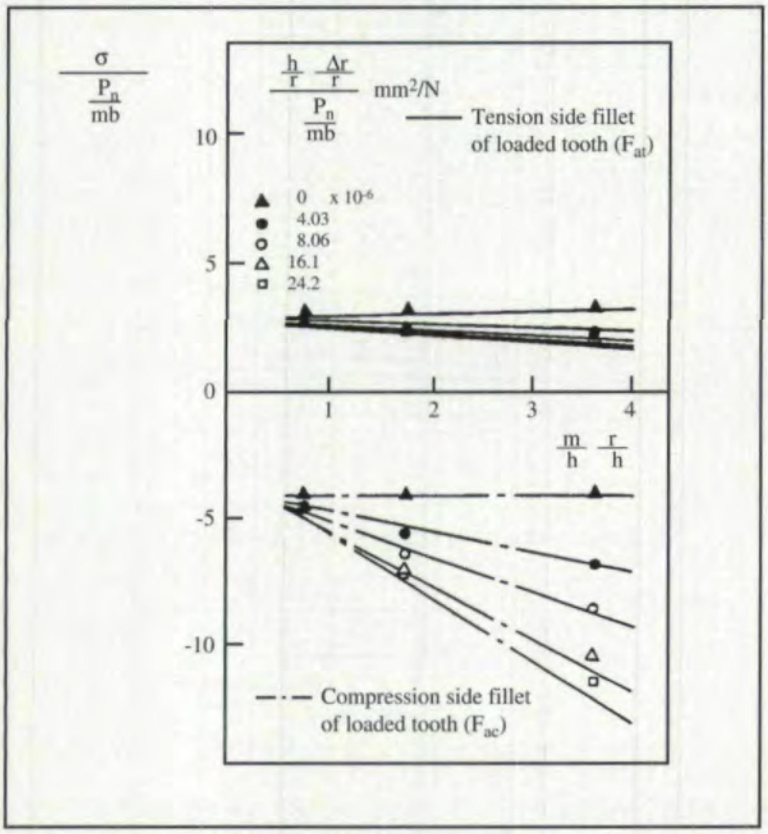


Fig. 8 — Relations between $\sigma_{at}/\{P_n/(mb)\}$ on F_{at} and $(m/h)(r/h)$, and between $\sigma_{ac}/\{P_n/(mb)\}$ on F_{ac} and $(m/h)(r/h)$ $(z = 90)$.

$$\sigma_{0t}/\{P_n/(mb)\} = A_{0t}(m/h)(r/h) + B_{0t}, \quad (12)$$
$$\sigma_{at}/\{P_n/(mb)\} = A_{at}(m/h)(r/h) + B_{at},$$
$$\sigma_{ac}/\{P_n/(mb)\} = A_{ac}(m/h)(r/h) + B_{ac},$$

where A and B denote the coefficients influenced by the number of teeth z and $(h/r)(\Delta r/r)/\{P_n/(mb)\}$.

$\sigma_{0t}/\{P_n/(mb)\}$ on fillet F_{0t}. Considering the result shown in Fig. 3, the relation between $\sigma_{0t}/\{P_n/(mb)\}$ and $(m/h)(r/h)$ can be expressed approximately by two kinds of lines as shown by **1** and **2** in Fig. 3. The values due to **1** and **2** are shown by σ_{0t1} and σ_{0t2}, respectively. Then comparing σ_{0t1} and σ_{0t2}, the values shown by the solid lines are chosen as σ_{0t}. Fig. 9 shows the relationship between $\sigma/\{P_n/(mb)\}$ and $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ when the number of teeth $z = 18$ and $(m/h)(r/h) = 1.69$. As shown in Fig. 9, when $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ is small, $\sigma_{0t1}/\{P_n/(mb)\}$ increases linearly before reaching the constant value of $\sigma_{0t2}/\{P_n/(mb)\}$. This corresponds to Eq. 9. Therefore, when $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ is small (before $\sigma_{0t1}/\{P_n/(mb)\}$ reaches the constant value), the coefficients A_{0t1} and B_{0t1} in Eq. 10 can be approximated as follows:

$$A_{0t1} = a_{0t11}(z)(h/r)(\Delta r/r)/\{P_n/(mb)\} + a_{0t12}(z),$$
$$B_{0t1} = b_{0t11}(z)(h/r)(\Delta r/r)/\{P_n/(mb)\} + b_{0t12}(z), \quad (13)$$

where the coefficients a and b are the function of the number of teeth z. Because the coefficient A_{0t1} hardly changes with $(h/r)(\Delta r/r)/\{P_n/(mb)\}$, and is almost constant, $a_{0t11} \approx 0$ is reduced under the condition when z is constant. When $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ is large (*after* $\sigma_{0t1}/\{P_n/(mb)\}$ reaches the constant value in Fig. 3), because $\sigma_{0t}/\{P_n/(mb)\}$ is approximately proportional to $(m/h)(r/h)$ and is not a function of $(h/r)(\Delta r/r)/\{P_n/(mb)\}$, coefficients A_{0t2} and B_{0t2} are reduced as follows:

$$A_{0t2} = a_{0t22}(z)$$
$$B_{0t2} = 0 \quad (14)$$

Analyzing the results shown in Figs. 3-8, the relations between coefficients A_{0t1}, B_{0t1}, A_{0t2} and B_{0t2} and $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ respectively, coefficients a and b are determined as shown in Table 2.

$\sigma_{at}/\{P_n/(mb)\}$ and $\sigma_{ac}/\{P_n/(mb)\}$ on fillets F_{at} and F_{ac}. It seems from Fig. 9 that coefficients A_{at}, B_{at}, A_{ac} and B_{ac} can be expressed by exponential functions of $(h/r)(\Delta r/r)/\{P_n/(mb)\}$. However, discussing the effects of $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ on the coefficients A_{at}, B_{at}, A_{ac} and B_{ac}, it was concluded that the first degree equations of $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ were enough to express both B_{at} and B_{ac}. Therefore, the approximate equations to express the coefficients A_{at}, B_{at}, A_{ac} and B_{ac} were chosen as follows:

$$A_{at} = a_{at1} + a_{at2}\exp\{a_{at3}(h/r)(\Delta r/r)/\{P_n/(mb)\}$$

$$B_{at} = b_{at1}(h/r)(r/r)/\{P_n/(mb)\} + b_{at2}$$
$$A_{ac} = a_{ac1} + a_{ac2}\exp\{a_{ac3}(h/r)(\Delta r/r)/\{P_n/(mb)\}$$
$$B_{ac} = b_{ac1}(h/r)(\Delta r/r)/\{P_n/(mb)\} + b_{ac2} \quad (15)$$

The coefficients *a* and *b* are shown in Table 2.

Comparison between approximated value and value by finite element method. The values calculated by the approximate Eqs. 12-15 are shown by the solid lines and the dotted lines in Figs. 3-8. It is clear from Figs. 3-8 that *the approximate values on the fillets are almost* the same as the values obtained by the finite element method. Although the comparison in the case where the number of teeth is $z = 62$ is not shown here, the approximate values were again almost the same as the values obtained by the finite element method.

Approximation of Mean Value and Amplitude of Alternating Fillet Stress

Comparing σ_{0t} calculated by Eqs. 12-14 with σ_{at} calculated by Eqs 12 and 15, the larger value is chosen as σ_t. Using σ_t and $\sigma_c = \sigma_{ac}$, both the mean value σ_m and the amplitude σ_r of the alternating fillet stress on the weakest section of the planet gear can be calculated as follows (Ref. 4):

$$\sigma_m = (\sigma_t + \sigma_c)/2$$
$$\sigma_r = (\sigma_t - \sigma_c)/2 \quad (16)$$

Summary

In order to estimate the bending strength of a spur planet gear with a thin rim whose pressure angle is 20° and whole depth is 2.25 m, the maximum stresses on both the tension side and compression side fillets of the loaded tooth, and the maximum stress on the fillet near the position where the reaction force from the gear shaft is applied, were analyzed by the finite element method, and were expressed by approximate equations. ■

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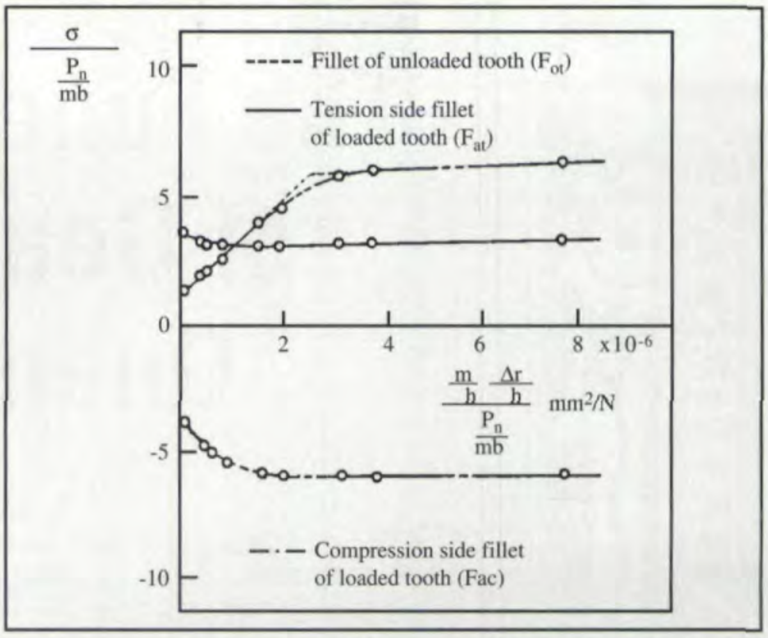

Fig. 9 — Relation between $\sigma/\{P_n/(mb)\}$ and $(h/r)(\Delta r/r)/\{P_n/(mb)\}$ on each fillet [$z = 18$, $(m/h)(r/h) = 1.69$].

Table 2 — Coefficients for the approximate Equations 12 - 15.

A_{0t1}	a_{0t11}	0
	a_{0t12}	2.72/z + 0.0677
B_{0t1}	b_{0t11}	1000(-777/z + 231)
	b_{0t12}	11.3/z + 0.119
A_{0t2}	a_{0t11}	0
	a_{0t12}	-26.3/z + 5.63
B_{0t2}	0	
A_{at}	a_{at1}	-0.645/z - 0.242
	a_{at2}	2.48/z + 0.297
	a_{at3}	10000(377/z - 30.3)
B_{at}	b_{at1}	1000(317/z - 15.1)
	b_{at2}	9.68/z + 2.97
A_{ac}	a_{ac1}	11.8/z - 3.06
	a_{ac2}	-11.2/z + 3.03
	a_{ac3}	-78200
B_{ac}	b_{ac1}	36000
	b_{ac2}	-14.8/z - 3.89

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The Fundamentals of Gear Press Quenching

L. E. "Skip" Jones
Lindberg Technical & Management Services Group
Charlotte, NC

Abstract: Most steel gear applications require appreciable loads to be applied that will result in high bending and compressive stresses. For the material (steel) to meet these performance criteria, the gear must be heat treated. Associated with this thermal processing is distortion. To control the distortion and achieve repeatable dimensional tolerances, the gear will be constrained during the quenching cycle of the heat treatment process. This type of fixture quenching is the function of gear quench pressing equipment.

Introduction

To understand press quenching, we must first understand the phenomenon of quenching and the resultant stresses within the gears that translate into dimensional changes or distortion. When we refer to the basic definitions of "press" and "quench," we see that:

Quench "is the immersion of metal into a liquid to extract heat"; and that

Press "is to exert a steady force or pressure on a component to achieve a desired shape."

Therefore, gear press quenching can be defined as "*heating a gear to the austenitic condition and then immersing the gear into a liquid to achieve metallurgical transformation, while at the same time constraining the gear to hold size and shape.*"

The most important thing to remember about gear press quenching is that its purpose is only to constrain the part and to use only enough force to counteract the parts' transformation stresses. Press quenching is not designed to reshape or forge gears into their final shape.

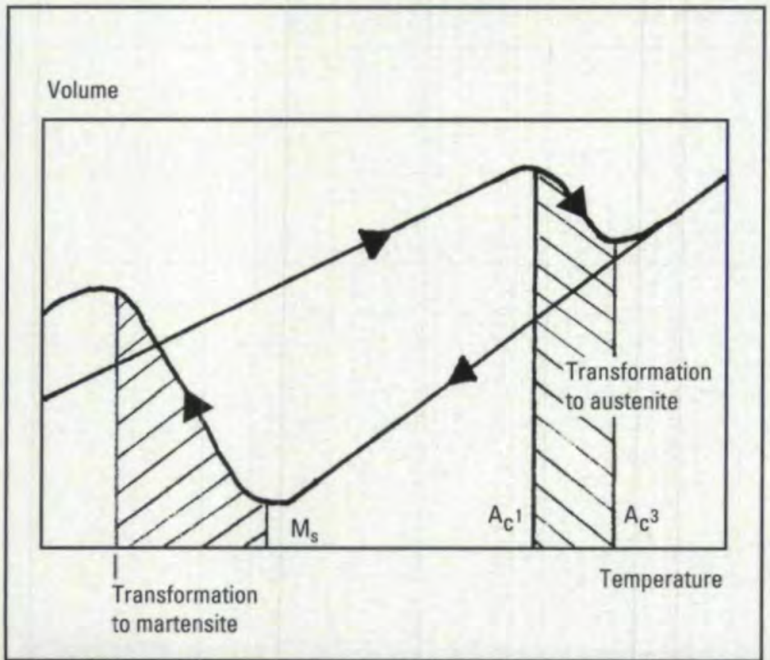

Fig. 1 – Volume changes due to structural transformation.

Causes of Distortions

Gear distortions can be caused by several factors and in any number of combinations. In the heat treating process, the gear is subjected to both thermal expansion and contraction. Non-uniformity in heating and cooling can cause stresses in a part, which, if large enough, can cause deformation. Consideration must also be given to the phase changes involved in hardening steel that result in expansions and contractions. Variations of this stress within the gear will cause deformation. Material composition variations resulting from the steel production process, grain flow variations caused in forging, and stresses due to pre-heat treat machining can all affect the dimensional stability of a part during hardening operations.

The magnitude of the distortion in heat treating is affected by the cooling rate, method of cooling, transformation temperature and part geometry. To minimize distortion, it is best to use a slow cooling rate and a method that will maintain a uniform temperature throughout the part.

It is critical that the transformation temperature be consistent throughout the piece, so that during cooling the entire piece transforms at one time. However, a carburized and hardened gear that has a composition gradient and, hence, a transformation temperature gradient, is more susceptible to distortion. (See Fig. 1).

Finally, the shape of a part strongly influences the degree of distortion it may undergo. Parts that are large and thin are more susceptible to distortion than parts that are compact and massive. Therefore, a complex gear also is a more viable candidate for press quenching than is a simple one.

Review of Quenching and Systems

Fundamentally, quench systems, regardless of the quench media – oil, water, polymer or brine – are similar. Agitation, temperature control and quenchant contamination control are of primary importance. Since the primary quench medium used in gear press quenching is oil, the emphasis will be focused here.

The quenching of steel provides the rapid cooling that is required to convert the austenite to hard martensite. The prime factors that determine the quench speed required are steel chemistry, part geometry and designed function of the part.

Too low a quench rate will result in soft parts. Too high a quench rate may result in excessive distortion and cracking.

Quenching has three distinctive stages:

1. *Vapor blanket stage* – Delta temperature creates an insulating vapor in slow cooling.

2. *Vapor transport stage* – Liquid breaks through the vapor barrier; rapid heat transfer takes place.

3. *Liquid stage* – Cooling is taking place by conduction and convection. At this stage, the part temperature is above the boiling point of the quenchant, but the rate of flow prevents the quenchant from boiling. (See Fig. 2).

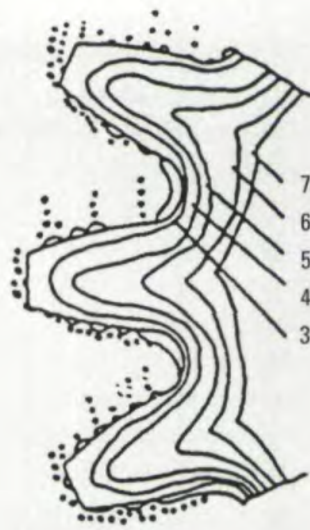

Fig. 2 – Temperatures during quenching.

The control of the vapor blanket stage is critical to the control of a consistent cooling rate for uniform heat extraction from the gear. Therefore, proper agitation or rapid movement of the quenchant is essential to prevent formation of vapor bubbles around the gear which insulate it and would result in irregular surface hardness and excessive distortion. As a rule of thumb, proper quenchant movement is defined as a volume flow of oil equal to two to three times the surface area of the gear in a one minute period.

To summarize, a proper quench system must have control of the temperature of the quench media, adequate quenchant movement, and consistent quenchant chemistry.

The Mechanics of the Press Quench

There are several manufacturers of quench presses, such as QPS, Inc., Gleason Works, Inc., Klingelnberg, Oerlikon and Jenny Presses, to name just a few. All quench press designs must incorporate a sufficient quantity of quench oil, variable quenchant flow rate, quenchant flow direction, die holding pressure, die contact points, and cyclic flow and pressures. The various press designs either use pneumatics or hydraulics for the pressure systems.

Our focus on a typical press quench operation will feature a press that uses hydraulics for mechanical movements and pressing force. (See Fig. 3).

L. E. Jones
is the Manager of the Technical & Management Services Group, Lindberg Heat Treating Company. He is a memeber of ASM, the Aerospace Council (SAE/AMC) and the International Federation of Heat Treaters and the author and editor of numerous books and papers on heat treating and metellurgy.

Basically, this type of quench press consists of a very rigid rectangular-box type construction. The base is a tank which acts as a reservoir for the quenching oil. It also serves as a support for the lower die. The upper portion of the machine contains the upper die ram assembly, hydraulic units, and the electrical panel. The opening at the front allows full access for changing the upper dies; the lower die table moves outward and inward for loading and unloading.

During operation, the component to be quenched is removed from a furnace (usually a pusher-type continuous or rotary-hearth-type), and placed onto the lower die in the "out" position. The automatic cycle moves the loaded lower die assembly into the center section of the machine and centers the die location. Next, the upper ram assembly descends, with an expander centering the part just prior to the inner and outer dies' location on their respective pressure points. The inner die, outer die and expander have completely independent pressure controls. (See Fig. 4). A circular guard completely encloses the upper dies to form a quench chamber which is affixed to the upper ram for movement. The chamber fills with oil flow from the lower die once the dies are in position. A variation in oil flow is controlled by three solenoid valves. Another unique feature is the ability to pulse the various die components (that is, maintain die contact on the component and cycle the pressure every two seconds or so). Also, the lower die is cam-adjustable, so the die rings can be raised or lowered to compensate for a predetermined amount of component dishing or camber. (See Fig. 5).

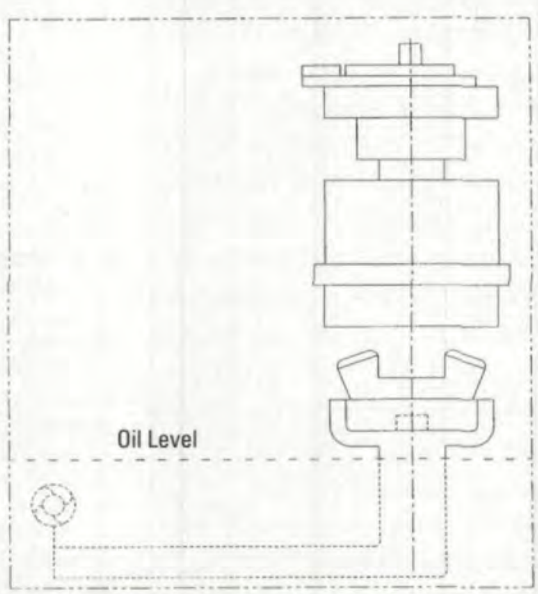

Fig. 3 – Schematic – Typical quench press.

The loading of the hot components can be either manual or robotic. The transfer mechanism is critical to assure a minimum of part heat loss from the furnace to the die. The temperature of the gear at the time of entering the quench is a determining factor of the final gear size.

The press quenchant system typically consists of the holding tank within the press, an external oil reservoir, circulation pump, heat exchanger, and internal heating source. The oil reservoir typically holds one and one-half to two times the holding capacity of the press itself. (See Fig. 6).

Within the quenchant system, oil temperature is maintained within 5°F plus or minus during operation. The operating oil temperature for most gear press quenching operations is between 130°F and 145°F.

Control of the Gear Press Quenching Process

This review of the control process will focus on carburized, case-hardened gears due to their high susceptibility to distortion. When examining a press quenching operation, consideration must be given to part geometry, chemistry of the component and the volumetric change resulting from the hardening process. Also, it must be remembered that the quench pressing operation will only round up and flatten the hot plastic gear, but will not change tooth shapes. With this in mind, it is imperative that the gear's prior manufacturing operations are closely controlled to assure consistent press quench results. The gear coming out of the press operation can only be as good as the gear going in. The variables that must be controlled before heat treatment are:

- Gear material composition,
- Grain direction and size,
- Prior thermal processing (normalizing),
- Machine stock removal,
- Cold work stresses,
- Tolerances and sizes of reference surfaces.

Once the gear enters the heat treat process, the procedures employed must be consistent, repeatable and closely controlled. The vari-

ables that must be controlled in carburizing are:

• Furnace temperature uniformity for heating rates and temperature distribution,

• Control surface carbon content and repeatable diffusion gradients,

• Gear fixturing during carburizing,

• Cooling rates after carburizing,

• Repeatability of temperature, uniformity, time at temperature and control cooling rates if annealing operations are used.

The reheating operation requires close control of:

• Temperature uniformity,

• Time at temperature,

• Fixturing and furnace loading,

• Atmosphere control or consistent stop-off procedures for surface integrity,

• Gear handling from the furnace to the press: Pickup contact and transfer time.

To successfully operate this type of press quench, the factors must be regulated:

• Quality of quench oil supplied,

• Duration of quenching,

• Quench oil temperature,

• Direction of quench oil flow,

• Pressure applied to hold the component,

• Location of component holding points.

In setting up this kind of press quench machine for operation, it is well recognized that the actual pressures and time settings must be developed for each component by trial and error. The following are general guidelines:

1. A fast quench or high oil flow is favored in the first stage to reduce the temperature of the part to the transformation range as quickly as possible. However, the flexibility available in the choice of duration and amount of oil flow permits a compromise between the need for a fast quench to ensure satisfactory hardening response and the desirability of a slower quench to minimize distortion.

2. The low flow rate in the second stage allows the temperature through the various cross sections to equalize. Thus, further cooling and martensitic transformation takes place with reduced internal stresses.

3. The third state returns to high oil flow when the component has nearly completed transformation. The high quenchant flow rate cools the part for operator handling.

4. In setting the pressures for the inner and outer dies, it is recommended that minimum levels be used. Use only the amount of pressure necessary to true up the component (high pressure settings can result in excessive distortions).

The variables within the actual press operation are:

• Die contact,

• Die oil flow,

• Oil flow rate and Delta temperature,

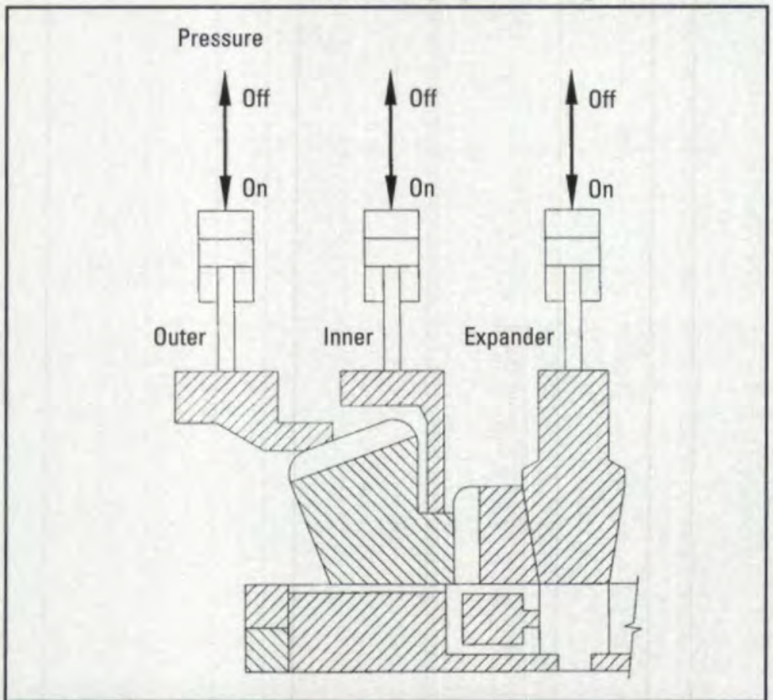

Fig. 4 – Die systems and ram pressure.

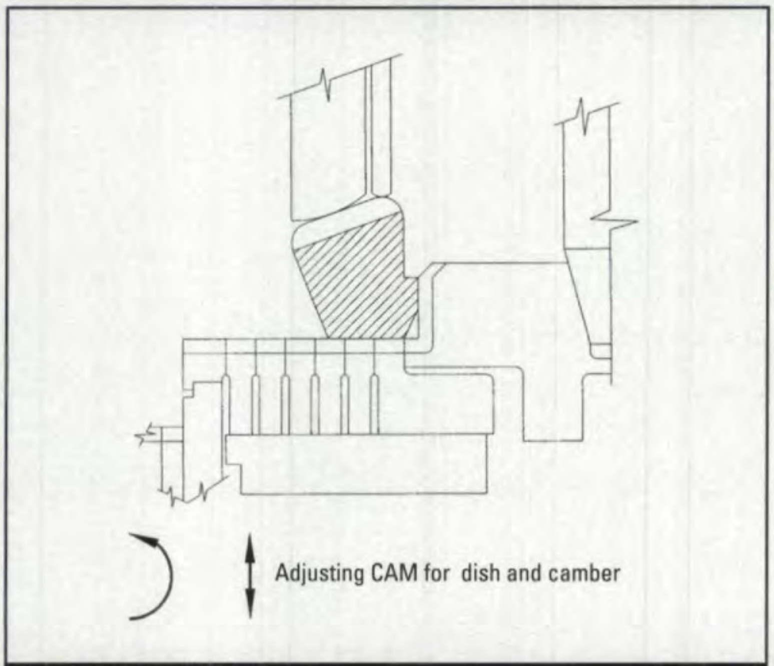

Fig. 5 – Lower quench die systems.

• Smooth movement of lower table,

• Die closure rate,

• Consistent ram pressure on all die rings,

• If pulse pressure is used, repeatable time for on and off cycles,

• Tooling care – nicks, burrs, scale or dirt *should be avoided so not to interfere with the* mechanical truing.

The control of gear press quenching operations cannot be isolated for control of gear distortion. The final results of the process are dependent on all prior operations. Therefore, if the gear manufacturing process is well documented, and the overall process is in control from the raw material stage to press quenching operation, repeatable results will be achieved with predictable size changes.

Gear Press Quench Instrumentation

As the demand on today's industrial quality systems and the need for system documentation and repeatability increase, more interest in press instrumentation systems is being generated. To date, development of instrumentation systems for press quenching has ranged from mechanical gauges to complex computer integrated systems. However simple or complex the instrumentation is, the information collected must be analyzed and correlated to the overall prior manufacturing process to gain control of the quench press operation.

A complete instrumentation system will document the following:

• The location of the lower table,

• Time measurement of the cycles indicating travel time, air lockup, cycle delay, stroke, cycle quench on, pulse on and off,

• Linear measurement of stroke movement,

• Quenchant flow rate,

• Quenchant oil temperature control,

• Quenchant Delta temperature through the die chamber,

• Hydraulic system pressure,

• Die position indicators,

• Non-contact temperature sensing of part temperature on transfer,

• Hydraulic sensitivity for dimensional measurement of the gear,

• Contact sensing device for measurement of part cooling rate.

With tabulation of the above data and immediate analysis, adjustments to the press quench operation can be made on-line, thus reducing rejected parts and increasing the predictability of the final as heat treat size. ■

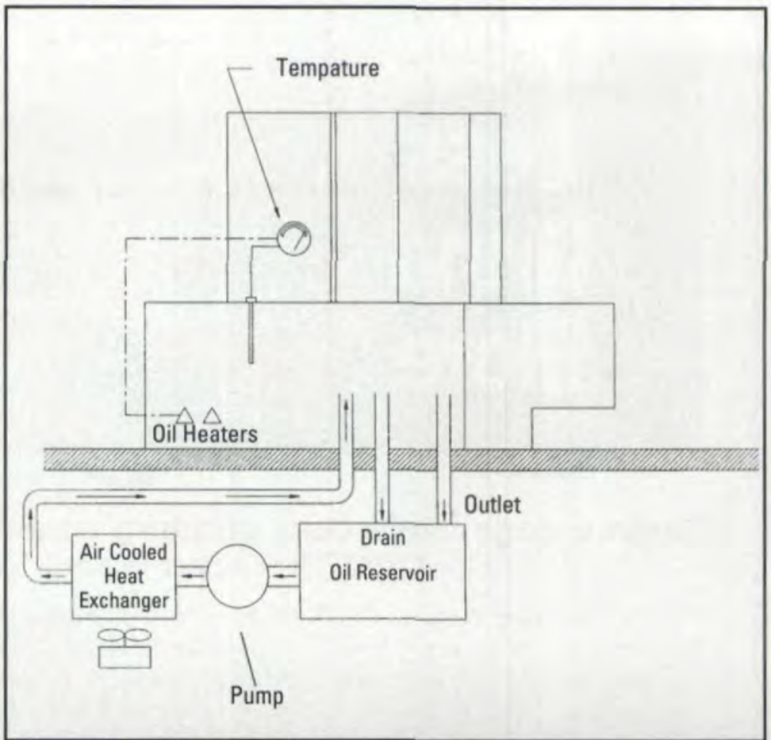

Fig. 6 – Quenching recirculation system.

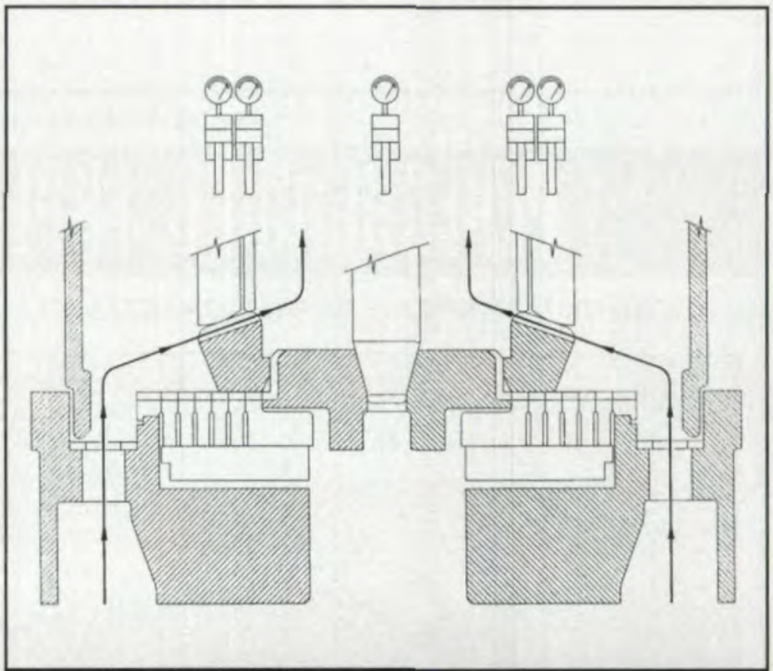

Fig. 7 – Typical die and quench chamber.

Acknowledgement: *Presented at the SME Conference, Heat Treating of Gears, August 26, 1993, Livonia, MI.*

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PEOPLE ON THE MOVE

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Norcross, GA. **Robert Wylie** has been named manager of market planning for ITW Fluid Products Group, a leading supplier of environmentally safe cutting systems, sump-maintenance fluids and associated products for the metal working and fabricating industries. Mr. Wylie's responsibilities include executing and overseeing marketing programs for the company's Accu-Lube, Cling Surface, Dykem, Rustlick and SafeTap product lines.

Janco Products, Inc.

Mishawaka, IN. Janco Products, a supplier of fiberglass reinforced plastic composites, has announced two promotions. **Chris Morin** has been promoted to production planner, and **Lenny Northam** will become the safety environmental director. Ms. Morin will plan production schedules for all departments, supervise print review and monitor engineering changes. Mr. Northam's new responsibilities will include the writing and implementation of the company's safety environmental programs and training procedures.

Do you have news of employee promotions, transfers or relocations you would like to share? Send your news release to Gear Technology, *People on the Move, P. O. Box 1426, Elk Grove Village, IL 60009, or fax it to our offices at (708) 437-6618. Note: Stories must be received by* ***the tenth of the month two months prior to the issue date*** *to appear in a particular issue. For example, if you want your story to appear in the May/June issue, it must be in our offices by March 10. Items reaching us past this deadline will appear in the next issue.*

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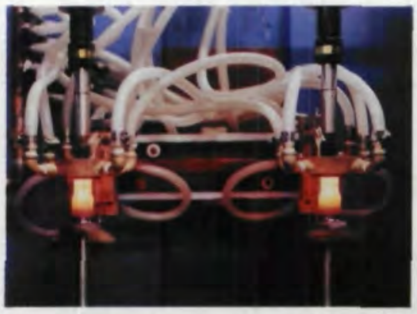

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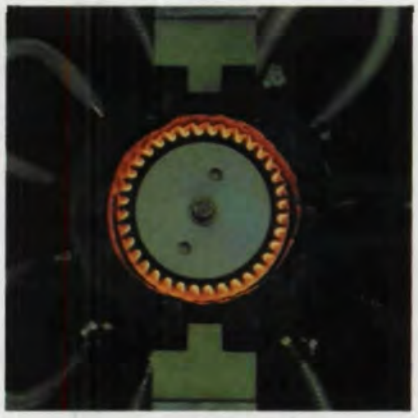

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Photography of Gear Failures

Robert Errichello

Photography is an essential part of gear failure analysis. It not only provides a fast, convenient way to accurately document the appearance of gear failures, but also is an effective diagnostic tool because the magnification obtained through photographic enlargement and slide projection often discloses evidence that may have been missed if the gears were not photographed.

I am not a professional photographer, however, I find it necessary to do my own photography. It has only been by personally composing the photographs through the camera viewfinder that the photographs show the particular detail I wish to capture.

While there are a multitude of ways to obtain good photographs and a huge array of photographic equipment to choose from, what follows is a description of equipment and techniques that I use for field and studio photography of gear failures. It is a proven system that has the following features:

- All equipment is lightweight and readily portable (easily carried on board aircraft);
- All equipment is robust enough to withstand the rigors of harsh industrial environments;
- The technique is easily learned and can be relied on to produce good photographs consistently.

The Basics

The equipment necessary for gear failure photography is not elaborate and is readily available at good camera stores. It consists of the camera, lenses, and a tripod. The subject of what kind of lighting to use and when will be discussed later in the article.

Camera. Most of the features of newer, computerized cameras are not necessary or do not work well for photographing gears. I have found that auto-focus cameras are not suited to the macrophotography of gears because the light reflected from shiny surfaces of gear teeth tends to confuse the computer. Almost any manual, 35 mm single lens reflex (SLR) camera body with quality lenses can be used for photographing gears. I use a manual Pentax SP 1000, SLR camera body that is an early version of the Pentax K-1000.

Tripod. For overall photographs I prefer to rely on ambient lighting, which usually requires relatively long exposures. With macrophotography, the lens needs to be stopped down in order to maintain the depth of field, which also requires long exposures. Therefore, a good tripod is a necessity to avoid camera-shake.

I use a Benbo Trekker 35 tripod with a pan-ballhead. It is compact and weighs less than five pounds. It is unique because the legs can be positioned at any angle, and it features a special monorail that allows you to raise, lower, extend, retract, twist or angle the camera without changing the tripod position. The flexibility is especially important for macrophotography because the working distances between the camera, umbrella, lamp

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

Robert Errichello
is the principal in GEARTECH, a gear consulting firm and founder of GEARTECH Software, Inc., in Albany, CA. He is also a writer and instructor on gear design, analysis and application subjects. He is a member of ASME, AGMA, STLE, and a Registered Professional Engineer in the State of California.

and subject are small, and the camera is usually the easiest item to adjust. The tripod adjusts easily to uneven terrain because each leg can be individually adjusted. This feature is especially important for photographing gears on-site where there may be limited space and no flat surfaces.

Lenses. I find that three lenses are sufficient for most of my failure analysis work. I use a 55 mm normal lens for most of the overall photographs of large gears and gearboxes. When there is not enough room to back away from the subject to include everything in the photograph, I use a 35 mm wide-angle lens. For close-up photographs of individual gear teeth, I use a 90 mm macro lens.

Macrolenses. Macrolenses are optically corrected for shooting at close range. Fixed-focal-length macrolenses have the best image quality. Macro/zoom lenses can focus closer than fixed-focal-length lenses, but they are not specifically designed for close-up work, and their image quality is not as good.

Fixed-focal-length lenses vary from about 50 mm to 200 mm. Longer focal lengths allow extra working distance between the lens and the subject. I use a Tamron SP 90 mm f/2.5 macrolens. The 90 mm focal length gives a minimum working distance of about 215 mm from the end of the lens to the subject (about 125 mm with extension tube).

The degree of magnification with macrolenses is specified as a magnification ratio, which is the ratio between the size of the subject's image on film and its real size. Most macrolenses have a magnification ratio of 1:2, which gives a film image that is 1/2 the real size. With an extension tube, the magnification ratio is increased to 1:1, giving a film image the same size as the subject.

Lighting The Subject

Gear teeth are hard to photograph properly, especially if they are highly polished, because light reflecting from shiny surfaces causes "hot spots" in the photograph that obliterate details. Camera-mounted flashguns are especially problematic because the light is reflected off shiny surfaces directly into the lens. Some improvement is obtained by using an extension cord and holding the flashgun to the side at an angle so that reflected light is directed away from the lens, but there is no sure way to know where to place the flashgun.

The basic problem with flashguns is that they emit a point source of very harsh light. Adding diffusers to the flash helps to soften the light, but the lighting is still too harsh because it originates from a single point. White cards can be used to bounce the light from the flashgun, which in effect increases the number of light sources, but the results are still unpredictable.

A ring flash is an improvement over a flashgun because it surrounds the lens with a ring of even light that is relatively soft. The ring flash eliminates hot spots in most cases, except for mirror-like subjects, such as bearing raceways.

The performance of a flash can be improved immensely by directing the light into an umbrella that reflects the light back to the subject. In effect, the source of the light is increased to the size of the umbrella, and the light is diffused and softened.

Tungsten Lighting. A major disadvantage of flash photography is that the lighting cannot be previewed by looking through the viewfinder. The solution is to eliminate the flash and substitute a tungsten lamp. With the light from the tungsten lamp reflected from an umbrella, the photograph can be composed through the viewfinder, and the photographer can see exactly what the photograph will show. Shadows are easily controlled by moving the umbrella or subject, and fill light can be added by using white cards. I find that the umbrella and tungsten lamp eliminate hot spots by providing soft, even illumination that results in photographs that are realistic and pleasing. The only disadvantage of the tungsten lamp is that it is hot and can be uncomfortable at close working distances.

Overall Photography

For overall photos of the entire gearbox or other large equipment, I use a 55 mm normal lens or a 35 mm wide angle lens. I prefer to use ambient lighting where possible because it

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Basic Equipment for Gear Failure Photography

What follows is a list of my "personal favorites" in terms of equipment. Other, comparable equipment is on the market, and readers are encouraged to look for that which works best for them.

- Camera — Pentax SP 1000, 35 mm SLR
- Tripod — Benbo Trekker 35 with pan-ballhead
- Normal lens — Takumar 55 mm
- Wide angle lens — Tamron 35 mm
- Macrolens — Tamron SP 90 mm f/2.5 with extension tube
- Tungsten lamp — Lowel Tota-Light, 1000 W
- Umbrella — Photek Goodliter Stowaway
- Light stand — Photoflex Litestand
- Color slide film — Kodak Ektachrome 320T Professional, ISO 320
- Color print film — Fujicolor Reala, ISO 100
- Gray card — Kodak 8 x 10 in. gray card
- Color card — Jobo
- Filters — No. 80A and skylight. (A skylight filter is a clear glass cover you may wish to put over your lenses to prevent scratches on them. A new lense may cost upwards of $100; a skylight filter costs less than $10.)
- Ring flash — Sunpak auto DX 8R
- Flashgun — Vivitar 283 with SB-4 power supply and flash cord
- Cable release

allows composition of the photograph through the lens. Ambient lighting avoids problems associated with flash photography, such as "hot spots" or overly dark backgrounds. Many industrial sites have fluorescent lights which create problems for most color films. However, Fujicolor Reala film gives excellent color reproduction, even with fluorescent lighting. The film speed is ISO 100, which is relatively slow, and it requires long exposures and a tripod.

Flash Photography

Flash photography is necessary for stop-action photography, such as when photographing a gear hanging from a swaying hoist. It also works well for photographing the interior of a gearbox through the inspection port because the light bounces off the interior surfaces of the housing, providing soft lighting. I use a Vivatar 283 flashgun for photographing components that are not shiny.

Nevertheless, to avoid hot spots, it is best to use an extension cord and hold the flash somewhat above and to the side, at about a 45° angle to the film plane. The flash can be mounted on a bracket attached to the camera or held in one hand. I find it convenient to hold the flash in my left hand and press the back of my wrist against my left temple, while holding the camera with my right hand. This maintains a consistent flash-to-camera distance.

When photographing at close range, this technique works well: Set the lens for the distance you want and the required aperture. Look through the viewfinder and rock back and forth slightly to focus precisely; then trip the shutter just as the subject becomes sharp. For components that are shiny, such as polished gear teeth, it is best to use the ring flash rather than the flashgun in order to avoid hot spots.

Ring Flash. It may not be practical to use a tungsten light in cramped quarters, for photography through small inspection ports, or in areas where electric power is not available. For these cases I use a Sunpak ring flash with a battery pack. The ring flash surrounds the lens with light and produces soft illumination that avoids most "hot spots." The results are not as good as those achieved with the tungsten lamp because ring lighting lacks gives "flat" photos.

Macrophotography

For most close-up photography, I use a macrolens and tungsten lighting. The advantages are:

• Tungsten lighting reflected from an umbrella gives soft, dispersed light that avoids "hot spots" or bright reflections from shiny gear teeth;

• Tungsten lighting allows viewing of the subject through the lens, enabling the photographer to see exactly what the photograph will show;

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Macrophotography is a very effective tool for failure analysis.

• Adjusting the angle of the light, camera or subject allows the photographer to get just the right amount of shadow or highlights to enhance the features he or she wants to show.

I use a Lowel Tota-Light, 1000 W tungsten lamp for a light source and a Photek Goodliter Stowaway umbrella. The light and umbrella are mounted on a Photoflex Litestand. All the lighting equipment was chosen for its compactness. Total weight for the lamp, umbrella and stand is only six pounds.

For most of my photographs of gear teeth, I use the macrolens without the extension tube. A 4 x 6 in. print is about twice the real size of the subject. There have been several instances where the magnification disclosed features that were not noted in visual inspection of the gears, proving that macrophotography is a very effective diagnostic tool for failure analysis.

For extreme close-ups, I add the extension tube to the macrolens. I reserve the extension tube for only those features of the subject that are especially interesting because it reduces the working distance and reduces the amount of light reaching the film. With less light, larger lens apertures are required, and there is less depth of field.

The major problems associated with macrophotography are:

• Working distances are small;

• Depth of field is shallow;

• Sharp focus is difficult to achieve.

Films for Macrophotography

For color slides, I use Kodak Ektachrome 320T professional film. It is balanced for tungsten light, eliminating the need for a lens filter. If a daylight-balanced film is used in tungsten light, a No. 80A filter must be added to the camera lens to keep the pictures from being too orange. The disadvantages of using a No. 80A filter are that it:

• Requires two stops of exposure compensation, giving less depth of field for the same shutter speed;

• Gives a dim, blue image in the viewfinder;

• Gives the photographer one more thing to remember when changing films or light sources;

• Causes loss of image quality;

• Does not reproduce colors as well as a film originally balanced for tungsten light.

Unfortunately, I have not found a tungsten-balanced film for color prints, so I normally use daylight-balanced Fujicolor Reala with a No. 80A filter.

Exposure

I use the camera's through-the-lens (TTL) meter and a gray card to get the proper exposure. For macrophotography with the tungsten light, a No. 80 A filter and Fujicolor Reala (ISO 100) film, the exposure is usually about f/11 with a shutter speed of one second. When using Kodak Ektachrome 320T slide film without the No. 80A filter, the exposure is usually about f/16 and 1/4 second.

I always bracket the exposures to ensure that I get at least one good photograph of each subject. For color print film, I bracket the base exposure by one stop. For example, if the base aperture is f/11, I also shoot the subject at f/16 and f/8. Because color slide film is more sensitive to the aperture setting, I bracket by 1/2 stop.

Documentation

I start every roll of film by photographing a color card. I write the film roll number and the date on the color card. This permits me to identify any exposed roll of film because the first frame is the color card with the roll number and date. When the film is processed, I ask the photo laboratory to include a print of the color card. By comparing the actual color card to the laboratory's print of the color card, the laboratory can correct the processing so that it produces the correct colors.

I record each photograph on a log sheet and include a brief description of the subject and the aperture setting, shutter speed, type of film and lighting. The log sheet helps to identify the best camera settings for future photographs and correlates the photographs with my separate, written descriptions of gear failures.

It is a good idea to type a list of instructions for each photographic process when first beginning to do failure analysis photography. Put the instructions on a laminated card and keep it in your camera bag for quick reference. It is also a good idea to make a list of all your camera equipment and put it on a laminated card. When packing up your equipment at the end of the day, checking the list will avoid leaving equipment behind. ■

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PUBLISHER'S PAGE

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rapidly becoming – at least in some industries – key to success in establishing global markets. If you want to play in that league, ISO 9000 is almost a must. At this point some 95 countries around the world have adopted the ISO 9000 standards in some form or other.

But more important, according to some who have been through the process, ISO 9000 certification can be good for your company. According to some participants in the program, its real benefit may come through the simple act of examining and documenting your particular business's processes and practices. One gear manufacturer I discussed ISO 9000 with said that for the first time since certification, people in his company felt they knew exactly what their responsibilities were, where to go to get problems resolved, and how their jobs fit into the overall process. According to a survey conducted by *Quality Systems Update* newsletter and the management consultants Deloitte & Touche, 42% of 620 companies surveyed reported that positive corporate change, greater company efficiency or enhanced internal communications resulted from their going through the ISO 9000 certification process. Better documentation was reported by 32% of those surveyed, and 20% cited greater quality awareness as a result.

Real dollar savings can result as well. According to the same survey, the *average* annual savings to companies who implemented ISO 9000 was $179,000. More than 50% of the respondents reported that they recouped their registratation costs in less than 40 months.

But there is a down-side to certification, and whatever your company's motive for implementing the ISO 9000 program, you should proceed with caution. The price alone ($35,000 start-up and in some cases well into six figures to get the process from start to finish) can be a deterrent, especially for small companies. So is the unfortunate politicalization of the process and the squabbling that has developed between ISO regulators in Europe and the U.S. A key question those considering certification must ask is whether or not the people they are using as registrars are going to be recognized by the people (usually their customers) who are demanding certification. A long hard look at your own markets and future plans, as well as your company's present quality assurance system are in order before buying into ISO 9000. Finally, no matter what ISO 9000 true believers say, ISO 9000 is not a panacea for all your problems.

So what's the answer on ISO 9000? It is the greatest thing since sliced bread or an expensive, over-sold marketing ploy? Probably something between those two extremes, but certainly a system important enough and useful enough to merit careful study by any company planning to operate in the current global economy.

Michael Goldstein, Editor-in-Chief

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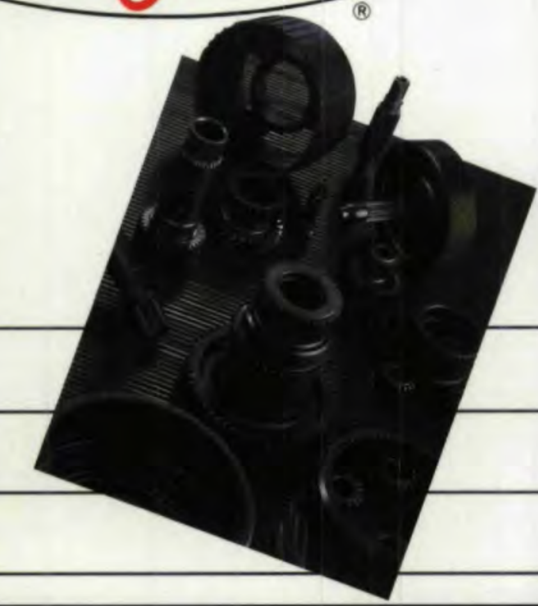

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