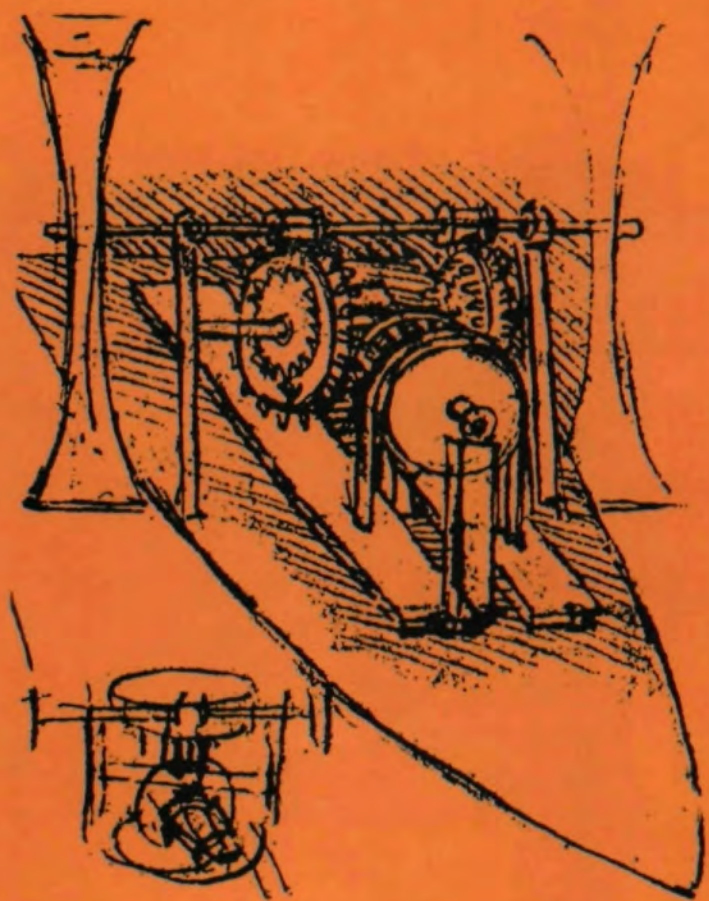
Gear Expo '87
GEAR EXPO '87

GEAR TECHNOLOGY

The Journal of Gear Manufacturing

SEPTEMBER/OCTOBER 1987

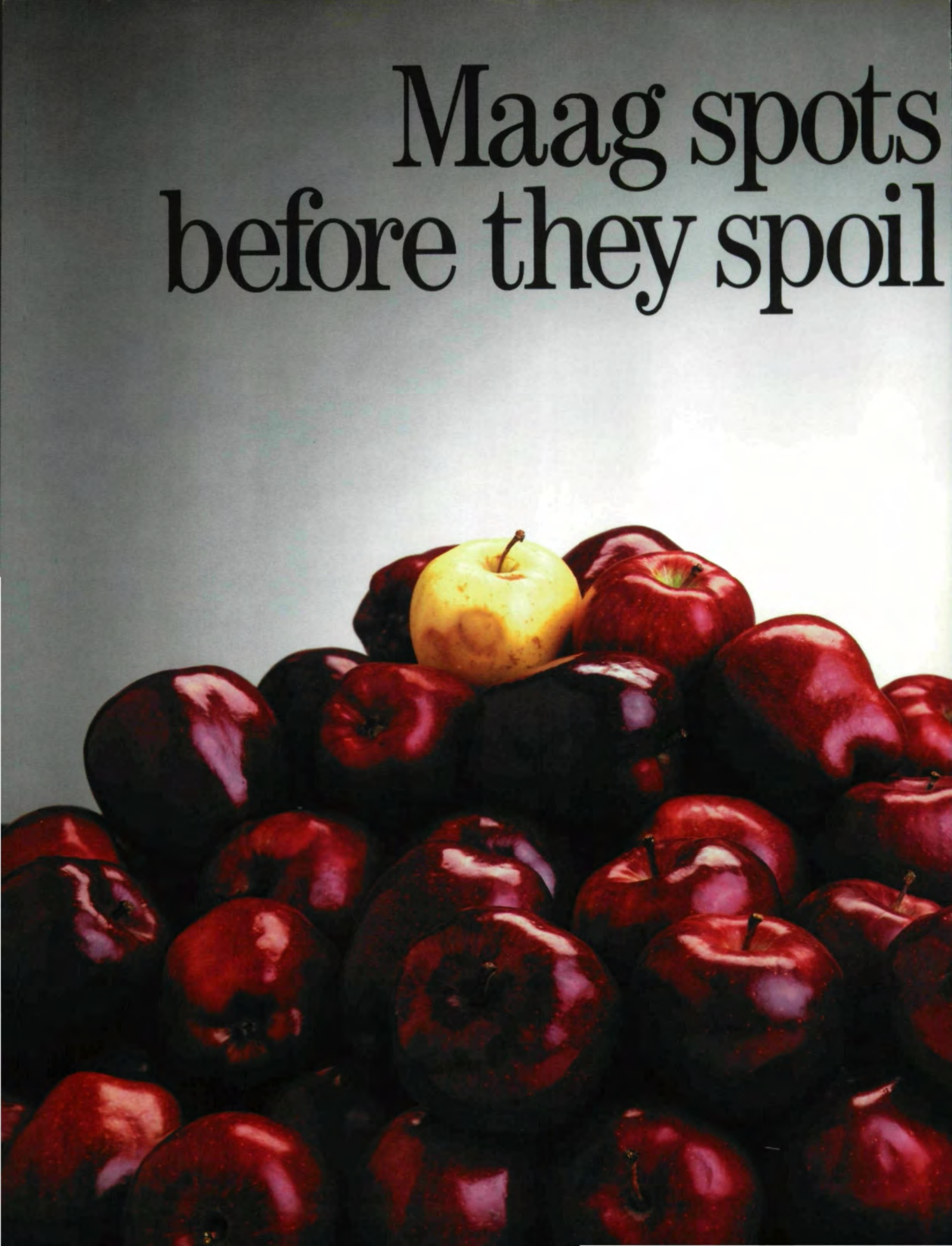


Tooth Thickness Measurements

Selection of Proper Ball Size to Check an Involute or Helical Gear

Effects of Quality & Resharpening Errors on Generating Accuracy

SI Units — Measurements & Equivalencies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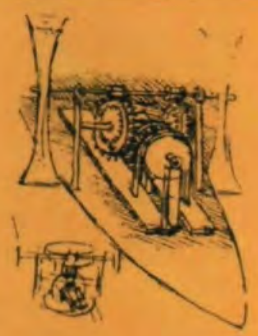

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COVER

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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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September/October, 1987 Vol. 4, No. 5

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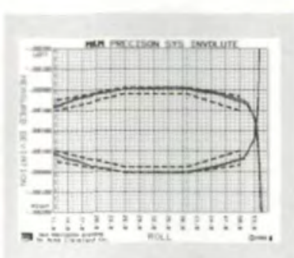

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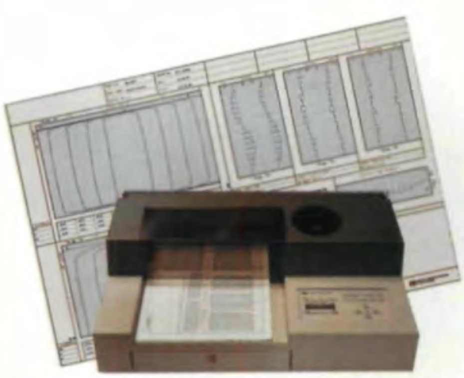

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EDITORIAL

INVEST IN THE FUTURE — NOW!

It is with great anticipation that we move closer to AGMA's Fall Technical Conference and Gear Expo '87, which is being held on Oct. 4-6 in Cincinnati, OH. This bold undertaking by both AGMA and the exhibitors in the Expo's 160 booths is an attempt to make a major change in the industry's approach to the exposition of gear manufacturing equipment. By combining the Expo with the Fall Technical Conference, those involved in gear manufacturing will have the opportunity to review the latest equipment, trends, and most innovative ideas, while keeping up with the newest technology in the industry.

AGMA and the exhibitors have put months of discussion and planning into getting Expo '87 off the ground. Their investment of time and resources represents the exhibitors' belief that if we are to stay competitive, we must be as well-informed as possible about developments within our industry. They are investing in the future, but their commitment and faith in our indusry is not enough. The most important ingredient in the success of this undertaking is YOU. . . the gear manufacturer and gear machinery and equipment buyer. Without your support, all the pre-planning in the world will not make the show succeed. You must match their vision.

There is a feeling in the air that a show devoted exclusively to the gearing industry is an idea whose time is definitely here. After some lean years and tough times, business in the gear industry is beginning to improve again. From this point of view, the timing of the show is perfect for looking at ways of changing and upgrading your manufacturing capabilities and for purchasing and installing new technology and equipment to improve your product and lower its cost.

Holding this expo away from the colossus IMTS will allow exhibitors and attendees the opportunity to focus on this small, but important industry. Products and ideas that might get lost in the crowd at IMTS will get an opportunity for center stage at the Gear Expo '87. Cincinnati, with its lower costs and location in the heartland of gear manufacturing America, provides the added advantage of keeping the costs down for both the exhibitors and attendees.

For those in nearby areas, a one-day trip with some of your employees might be one good way to sample the exhibits. The return in knowledge and in employee morale will far outweigh the nominal cost. Courtesy tickets are readily available from Gear Expo exhibitors listed elsewhere in this issue.

An Expo and Conference attended by large numbers of engineers and management will help make the next biennial show even bigger and more fruitful for both the exhibitors and the attendees. Do something for yourself, your employees, your company and the welfare of your industry. Make plans to visit the Gear Expo '87 in early October. We'll see you there.

Michael Goldstein
Editor/Publisher

ENGINEERING CONSTANTS . . .

SI Units — Measurements and Equivalencies

Stan Jakuba
S. R. Jakub Associates,
West Hartford, CT

Throughout the history of civilization attempts have been made to limit the number of the measuring systems in use with the result that today only two systems, English and metric, are practiced in the industrial nations. Globally, the metric system has been gaining ground, and the English system has been losing it. As of 1986, only the United States, Burma and Brunei remain uncommitted to metric conversion in the sense that no government controlled deadlines for the conversion have been established. In the U.S., the lack of governmental leadership is surprising in light of the importance our founding fathers placed on the need to adopt a systematic and decimal set of measures in this country. Thanks to Thomas Jefferson, our nation was first to have a decimal coinage plan (ten cents to a dime, ten dimes to a dollar); he also proposed the division of a day into decimal increments — a measure which still awaits acceptance. Despite the present lack of governmental involvement, metrification has been progressing, particularly in the automotive and export oriented industries. The conversion is guided by various engineering and educational societies, and there is an agreement among them to establish the so-called SI version of the metric system here.

The abbreviation SI has been adopted by all languages of the world to denote the International System of Units. SI was created in 1960, and it is intended to serve the needs of professionals as well as the general public worldwide. SI is distinguised by its coherent set of units; there is only one unit for any physical quantity in such a system. If, for example, the English system were coherent, it would have only one unit of length, and that unit would replace all other length and length-related units such as angstrom, mil, foot, acre, pint, gallon, bushel and barrel; conversely, instead of ounce, which denotes two different volumes, two different masses and a force, it would have an individual unit for each of the three quantities. A coherent system has no conversion factors and is easy to learn.

AUTHOR:

MR. STAN JAKUBA *has over twenty years experience in the gear industry in the United States and overseas. President of S.R. Jakub Associates, Engineering and Training Consultants, Mr. Jakuba was educated in Czechoslovakia and holds a masters degree in mechanical engineering from MIT. He is the holder of several patents for engineering products and is a member of ASME and SAE. He is also secretary of the U.S. Metric Association.*

The two systems practiced today, English and metric, have been revised several times in their history and are concurrently used in more than one version. As we know, this country uses the U.S. version of the English system, a version which has most units defined on metric standards and which also employs metric units. We also know that other English speaking countries have used the Imperial version. It is not so well known that most metric countries have used a mixture of several versions of the metric system, such as cgs, MKS, gravitational and SI.

By now, most industrial nations have imposed mandates requiring the exclusive use of SI units. There is, however, a large body of metric literature, drawings and standards which contain old and obsolete units. Furthermore, not every company and every department within a company has implemented the mandate yet, and thus non-SI units are still appearing. A person trying to learn SI by studying documents which contain metric data faces a confusing ordeal.

The purpose of the following charts is to bring some order to this multiplicity of units and aid the engineer or technician in making necessary conversions.

Table I is a chart of U.S. customary and SI units arranged in the alphabetical order of their respective physical quantities. About fifty quantities are listed, selected to cover the common mechanical engineering disciplines.

Experience indicates that the resistance against SI data and SI calculations stems largely from the lack of the feel for the "ball park figures." To provide some feel, the table includes a column of Typical Values with approximately one hundred and fifty engineering constants and reference numbers.

The column of the SI units shows their symbols in the form best suited for typing. Prefixes are included where a particular prefix is always encountered in engineering practice, such as dimension (mm) and kinematic viscosity (mm^2/s), and also in the case of the kg. The numbers in the Typical Values column provide a guide for the selection of suitable prefixes for the other quantities.

SI Equivalents

The numbers are rounded off to satisfy common engineering accuracy. The use of table is illustrated at the end.

Acceleration: longitudinal

1 ft/sec^2	is	0.305	m/s^2
1 in/sec^2	is	0.0254	m/s^2
1g	is	9.81	m/s^2

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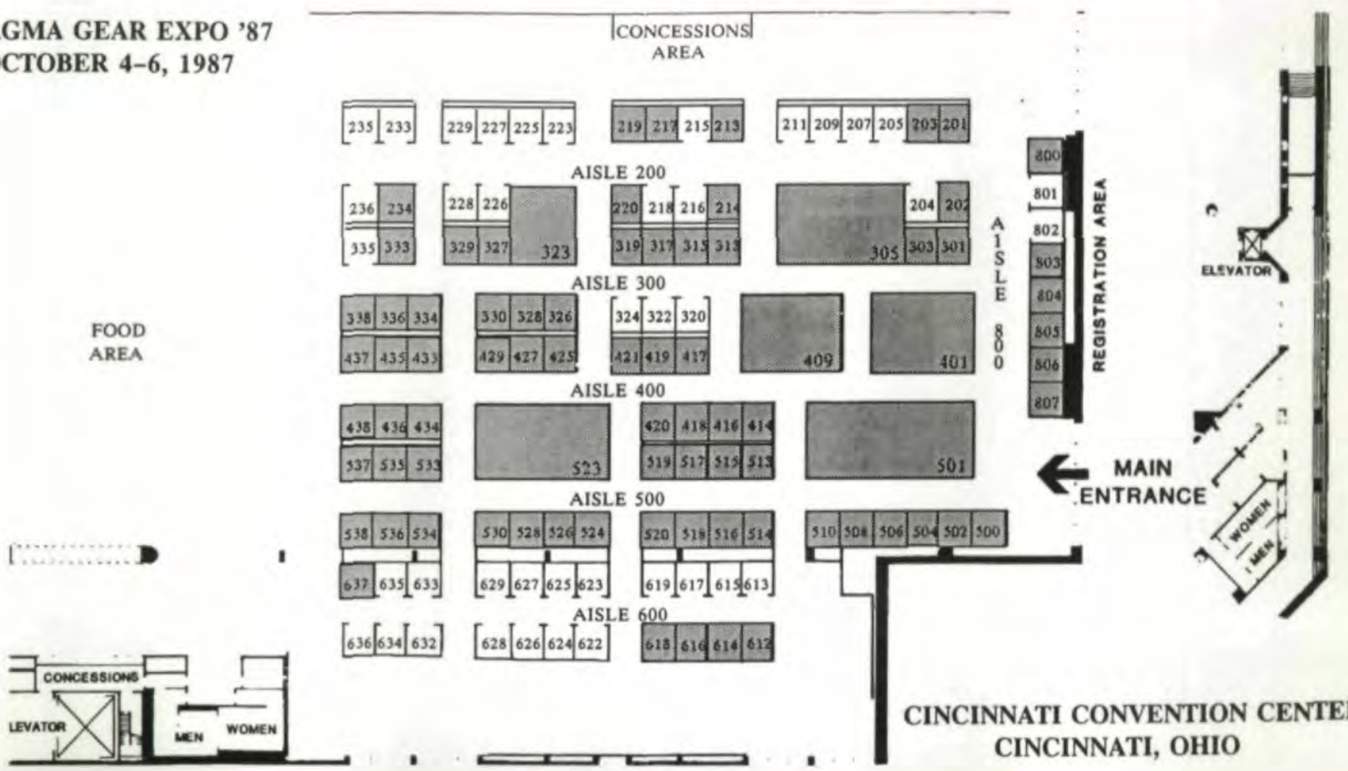
AGMA GEAR EXPO '87
OCTOBER 4–6, 1987



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GUEST EDITORIAL

KEEPING AHEAD BY KEEPING UP

A commitment to boost United States' industrial competitiveness in future years must strike beyond legislative action and economic debates.

The commitment must start with education—education from elementary schools to research laboratories, right through to professional development courses. This is not meant to suggest that education alone can reduce the trade deficit, resolve the issues of protectionism and return the U.S. to manufacturing superiority; education can establish trends, however, and lay a foundation on which to build our industrial competitiveness through the years.

The Council on Public Affairs of the American Society of Mechanical Engineers has addressed this issue in a policy statement called "Restoring America's International Competitiveness." Its recommendations and conclusions focus on a number of ways to lay this crucial educational foundation.

To lead the world through the technological jungle, we require superior engineers and scientists. Improvements in our educational system and a rise in the level of funding for research in engineering and science would point us in the right direction.

Steps must be taken to encourage young students to pursue mathematics and science because future engineers and scientists will grow from the children that enjoy those subjects.

Local and state boards of education should increase emphasis on math and science with expanded course requirements for high school curricula. Introductory courses in technology should also be pursued. Japan has established priorities by requiring the study of science and technology during elementary school. In the U.S., one-half of high school graduates take no math or science beyond the 10th grade.

This leads to another major issue: let's get the most qualified science and mathematics teachers back into the classrooms. Obviously, public schools cannot approach the salaries of the private sector; however, many working professionals could teach on a part-time basis and recently retired engineers and scientists could provide a strong impact on educating the young.

At the university level, the federal and state governments should increase investments in engineering and science programs while stressing modernization of research facilities. A recent survey by the National Science Foundation revealed that only 18 percent of equipment used in university engineering laboratories is state-of-the-art. Once again, part-time instructors and retirees from industry could share their knowledge as well as help to provide an important link between industries and universities.

A key to international economic competition is a deep-rooted commitment to research. The federal government should target funds for high-risk, long-term engineering research, and industries should make research in manufacturing processes a priority.

Government, industry and universities must sharpen their focus on research and development. Federal and state governments need specific entities to develop, coordinate and implement policies that affect research. In the long term, a cabinet level Department of Science and Technology would coordinate programs and create a single voice for the fields of science and engineering. Companies and universities also need to re-evaluate their programs in order to pursue interests with a long-term benefit.

Beyond the labs and classrooms, education must continue for professionals in the field. The knowledge, motivation and quality of the work force will influence heavily the nation's ability to compete in the international marketplace. The ultimate responsibility for personal educational improvement lies with the individual, but companies can benefit the individual and organization by encouraging on-going educational programs.

The necessity of lifelong education grows in saliency when thinking about the rapid technological changes and increasingly competitive world market. Some economists project that people now entering the workforce may need to be retrained at least six times during their careers. For engineers, this point is particularly potent.

A well-educated work force fuels technology and spurs the ability of companies to compete.

Richard Rosenberg

Richard Rosenberg
President, ASME

AUTHOR: RICHARD ROSENBERG *is president of the American Society of Mechanical Engineers. He spent 25 years with GA Technologies in San Diego, CA, working on the design and development of various components for nuclear power reactors. He retired in 1986 as manager-systems and components, directing all GA activities for the Fort St. Vrain nuclear power plant, Platteville, CO. A registered professional engineer in Pennsylvania and California, Mr. Rosenberg holds several patents related to mechanical equipment for nuclear reactors. He is a Fellow of ASME and a Fellow of the Institution of Mechanical Engineers, United Kingdom. Mr. Rosenberg earned a B.S. in Mechanical Engineering from the University of Tennessee, Knoxville.*

The Interrelationship of Tooth Thickness Measurements as Evaluated by Various Measuring Techniques

Paul M. Dean, Jr., Consultant
Schenectady, NY

Abstract

Measured tooth thickness as established by measurements made by conventional gear measuring techniques: over pins, the span measurement, or with a gear tooth vernier caliper, do not always agree with the "effective tooth thickness," (the value "seen" by the mating gear). Methods of adjusting the specified value of measured tooth thickness to assure that the required value of effective tooth thickness will not be exceeded are discussed.

Introduction

The first commandment for gears reads "Gears must have backlash!" When gear teeth are operated without adequate backlash, any of several problems may occur, some of which may lead to disaster. As the teeth try to force their way through mesh, excessive separating forces are created which may cause bearing failures. These same forces also produce a wedging action between the teeth with resulting high loads on the teeth. Such loads often lead to pitting and to other failures related to surface fatigue, and in some cases, bending failures.

If, however, the mesh contains excessive backlash, certain applications, particularly those in which the direction of loading reverses, will exhibit rough running and poor performance. It is, therefore, very important that the tooth thickness and the center distance, both of which govern backlash, be correctly designed, properly specified and accurately controlled.

In most cases, when problems relating to backlash do occur, it is not because the basic design values used by the gear designer were too small; rather, the problem is that the maximum values for the tooth thickness measurements specified on the drawings could actually yield effective tooth thicknesses greater than the designer anticipated as "seen" by the mating gear.

The objective of this article is to show how to determine drawing specifications for conventional tooth thickness measuring techniques that will yield gears with the desired effective tooth thicknesses. The relationships between the various types of tooth thickness are considred. The methods used by the AGMA to specify the allowable variations for each given quality number of accuracy, spacing, profile, runout and lead are examined. The way in which each of these allowable variations enter into each tooth thickness measuring method are considered. Finally, a simplified method of relating the measured value of tooth thickness to the effective thickness is shown.

Types of Tooth Thickness

In order to establish proper tooth thickness specifications, four types of tooth thicknesses should be recognized:

- Design tooth thickness.
- Effective tooth thickness.
- Functional tooth thickness.
- Measured tooth thickness.

Design tooth thickness is the arc distance measured along a specific circle, usually the standard pitch circle, (See Equation 2.) between the involute curves defining each side of a gear tooth.

It is a theoretical value, usually established by engineering considerations. AGMA Standards 218, 219, 360 and 370 offer guidance in establishing design tooth thickness.

The maximum limits on design tooth thickness for each member of a pair is typically established on the basis of minimum operating center distance, considerations of thermal differential expansion due to temperature extremes in the gears and mountings, the internal runouts and clearances within the bearings supporting the gears and the minimum allowable backlash.

The maximum design tooth thickness may be interpreted as a maximum metal condition on all of the active surfaces

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of the teeth in a gear or pinion. (See Fig. 1.)

Effective tooth thickness is the envelope of tooth thickness as "seen" by the mating gear in an operating set of gearing. In most cases, the numerical value of the maximum effective tooth thickness, T", is established equal to the maximum design tooth thickness, T'.

Functional tooth thickness is a measured value of the effective thickness of a gear as "seen" by a master gear. It is determined by means of a properly designed and calibrated master gear operating on a gear rolling fixture.

The maximum functional tooth thickness of a work gear may be obtained from Equation 7. The value of C_{max} to use in the equation is the largest value of instantaneous center distance that was observed when all teeth of the work gear had passed through mesh.

Measured tooth thickness of a gear is the arc distance from a point on one side of a tooth to a similar point on the other side at a specific diameter. It may be determined by means of a gear tooth vernier, a measurement over 1, 2 or 3 wires

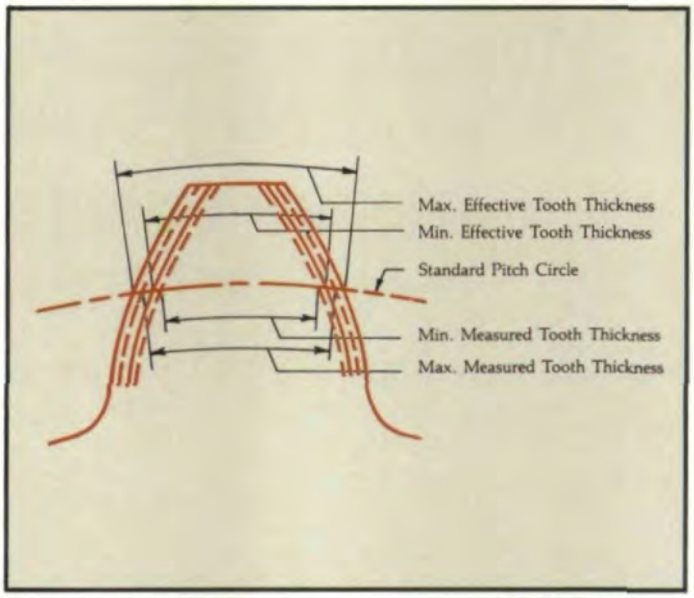

Fig. 1 – The relationship of effective and measured tooth thickness.

NOMENCLATURE

a	Radial distance from measuring circle (chordal tooth thickness measurement) to outside circle
a_c	Chordal addendum setting on a tooth vernier caliper
C	Standard center distance
C'	Operating center distance
C_M	Center distance, work gear and master gear (use $C_M = C_{max}$ for largest value)
C_{max}	Largest observed center distance master and work on a gear rolling fixture
ΔC	Change in center distance, (runout)
D	Standard pitch diameter
D_b	Base circle diameter
D_M	Pitch diameter, master gear
d_w	Measuring wire diameter
d_M	Diameter of measuring circle
M_N	Testing center distance specification for gear rolling fixture
M_1	Measurement over 1 wire
M_2	Measurement over 2 wires
N_P	Number of teeth in pinion
N, N_G	Number of teeth in gear
N_M	Number of teeth in master gear
n	Number of teeth in span
P_{nd}	Diametral pitch, normal
P_d	Diametral pitch, transverse
R_W	Radius of circle through center of measuring wire
s	Space width on standard pitch circle
T_E	Tooth thickness, effective
T_M	Tooth thickness, master gear
T_{Mc}	Tooth thickness, chordal tooth thickness measurement
T_{Ms}	Tooth thickness, span measurement
T_{M1}	Tooth thickness, over wires measurement, (1 wire)
T_{M2}	Tooth thickness, over wires measurement, (2 wires)
T_W	Tooth thickness, functional, maximum
T'	Tooth thickness, maximum design
T"	Tooth thickness, maximum effective
t	Tooth thickness on the standard pitch circle
t_b	Base tooth thickness
tc	Chordal tooth thickness specification
Δt	Change in arc tooth thickness
Δt_1	Tooth thickness adjustment factor (measurement over 1 wire)
Δt_2	Tooth thickness adjustment factor (measurement over 2 wires)
Δt_c	Tooth thickness adjustment factor (Chordal tooth thickness measurement)
Δt_s	Tooth thickenss adjustment factor (Span measurement)
v_L	Lead variation, allowable
v_P	Profile variation, allowable
v_S	Spacing variation, allowable
v_R	Runout variation, allowable
v_{TR}	Runout variation, in pitch plane, allowable
Φ	Standard pressure angle, profile
Φ'	Operating pressure angle
Φ''	Operating pressure angle, work gear with master gear
Φ_M	Pressure angle at center of wire
Ψ	Helix angle

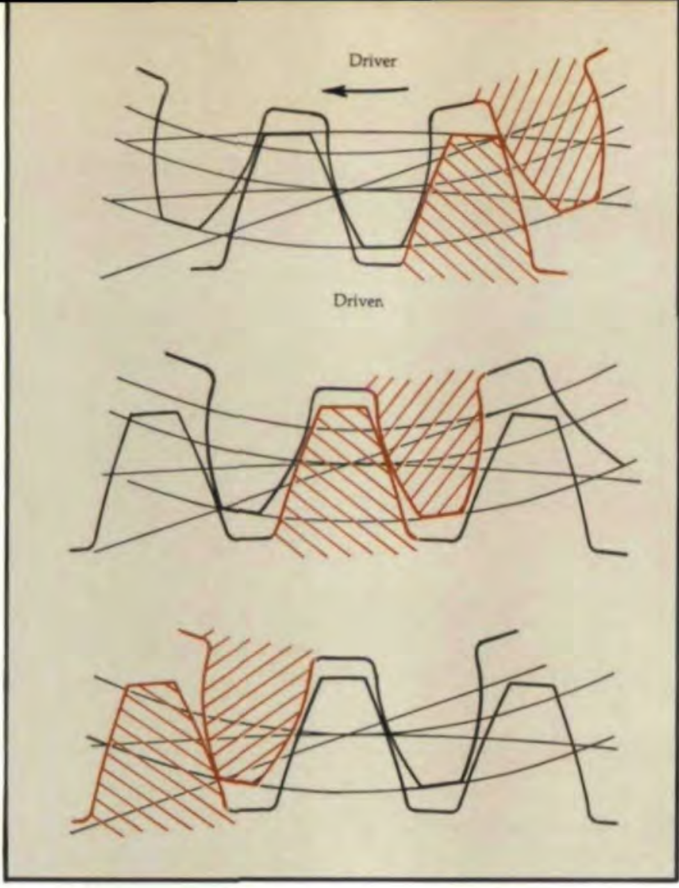

Fig. 2 – Meshing sequence of a pair of theoretically perfect gear teeth viewed from the transverse plane.

(pins), a span measurement or by means of a master gear and a gear rolling fixture.

Geometric Considerations

Fig. 2 shows the sequence of events that takes place in a transverse plane as a pair of theoretically perfect gear teeth go through mesh. All contact between involute teeth takes place along a line of action defined by the base circles of the meshing gears. The line of action is actually an edge view of the plane of action. The plane of action is a surface tangent to each of the base cylinders of the gears. In the plane of action, illustrated in Fig. 3, the lines of contact are shown. These lines show the actual contact across the length of the teeth. At any instant, in perfect gears, contact occurs throughout the entire length of all of the lines bounded by the zone of contact. The zone of contact is defined by the sides of the gears (ends of the teeth) and the outside diameters of each member.

Contact between meshing gears can also be studied in a meshing plane, which is the developed surface of either of the pitch cylinders of the pair. (See Fig. 4.) The cross hatched sections indicate the arc thickness of the teeth of each member. The allowable variations in spacing, the allowable lead and the radial component of runout, as shown in the *AGMA Handbook* are specified in this plane.

Actual gear teeth are not perfect. They have variations in spacing, profile and lead. Also there are low and high areas relative to the theoretical surfaces of the teeth. A "high" area will be "seen" by a meshing tooth as a thicker part of the tooth. A "low" area, however, is not so likely to be "seen" by the mating tooth, since it may be bridged over by the

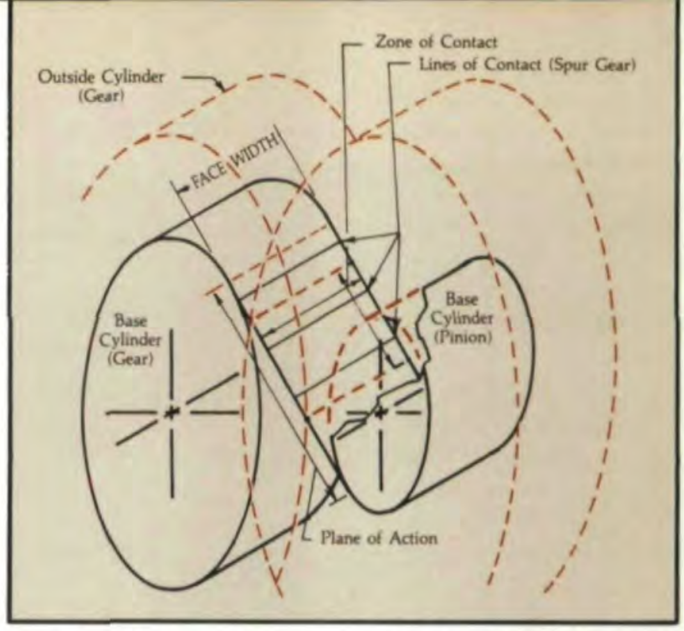

Fig. 3 – Lines of contact in the plane of action for the same set of theoretically perfect gear teeth.

overall length of the line of contact. Thus, a mating gear "sees" only a maximum metal condition.

Measurement of Tooth Thickness

Functional tooth thickness is measured by means of a calibrated master gear, a gear rolling fixture and the *AGMA Handbook, 390.03.* (See Fig. 5.) This method of measurement determines the maximum metal conditions on the work gear, since the master gear has the potential of contacting all parts of the active profiles of each tooth. For those gears that can be checked by the functional check, it is generally satisfactory to specify the value of maximum allowable testing center distance, $C_{T\,max}$, which is based on the value of maximum design tooth thickness, T′. (See Equation 9.)

From both the gear user's and the gear manufacturer's standpoint, it would be ideal if a single method of measurement that would determine if gears would be suitable for their intended service could be applied prior to shipment. The functional check comes close to this ideal, but its application is limited to a somewhat restricted range of gear sizes. It is the only method of gear inspection in common use that directly evaluates the effective tooth thickness of a gear.

The *AGMA Handbook* describes the functional check and the methods of calibrating the gear rolling fixture and the master gear when tooth thickness measurements are to be made. For gears that cannot be conveniently measured by the functional check, it is necessary to use a more indirect method to determine the effective tooth thickness of a gear. Three steps are involved in this process. First, the measured tooth thickness is obtained by means of a measurement over wires, a chordal tooth thickness measurement or by a span measurement. From a practical standpoint, each of these measurements actually evaluate only a small local area of the tooth.

Next, the gear is evaluated to determine its quality by means of measurements of its individual tooth element variations. The *AGMA Handbook* recognizes four specific types of tooth element variations. These are lead, pitch, profile and runout. Allowable tolerances are given for each AGMA Quality Number based on the pitch diameter and diametral pitch of the specific gear.

Finally, the effective tooth thickness is determined by adding to the measured tooth thickness the amount that each of the individual elements, lead, pitch, profile and runout, contribute to the effective tooth thickness of the gear.

When specifying gear tooth dimensions, the gear designer should specify a value of measured tooth thickness which is

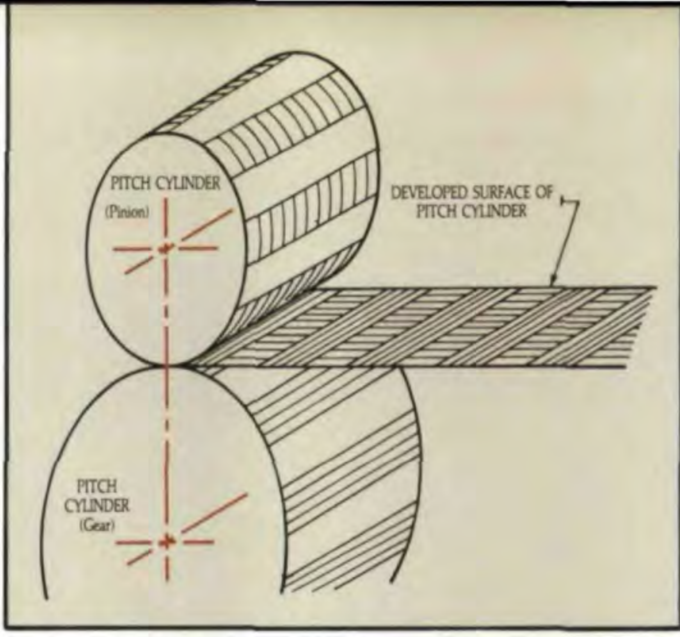

Fig. 4 – Developed surface of pitch cylinders containing cross sections of the meshing teeth.

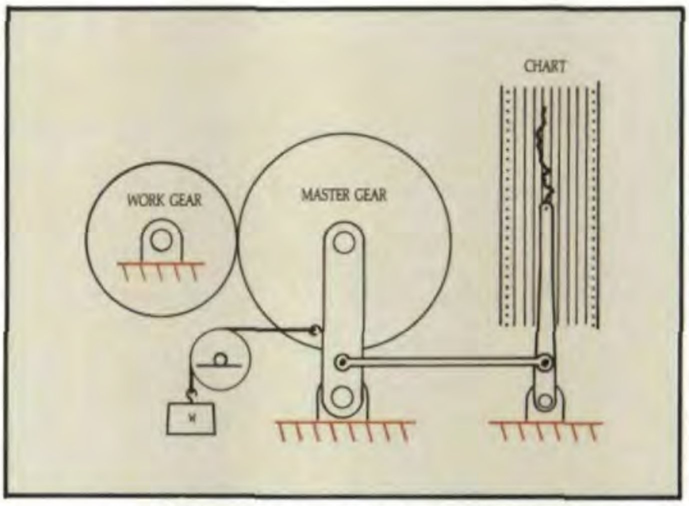

Fig. 5 – Schematic diagram of a gear rolling fixture.

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less than the value of effective tooth thickness. The amount by which the effective tooth thickness should be reduced is a function of the combined statistical effects of each of the individual allowable tooth element variations.

Effects of Tooth Element Variations on Effective Tooth Thickness

The effects that the various individual element variations have on effective tooth thickness can be illustrated by means of the meshing plane, which is a developed surface of the pitch cylinders of the meshing gears. (See Fig. 4.)

Lead. Fig. 6 shows a cross section of a tooth of Gear B lying in the space between two teeth of Gear A in a meshing plane. To the left is shown the situation of perfect teeth; neither member has a lead variation. The tooth of Gear B lies parallel to the space of Gear A. In this case, the measured tooth thickness is the same as the effective tooth thickness. This produces the backlash shown. The backlash is the difference between the effective space width and the measured tooth thickness. To the right is shown an example in which the teeth of Gear A have no lead variation. (The effective space width is the same as in the left illustration.) Gear Tooth B, however, shown in the right illustration, has the same measured tooth thickness as Gear Tooth B in the left illustration. It also has a lead variation which produces an effective tooth thickness that is larger than the measured tooth thickness. The resulting backlash is a smaller value.

The method of measuring lead and the way in which tolerance on lead is specified shows why the effective tooth thickness will generally exceed measured tooth thickness.

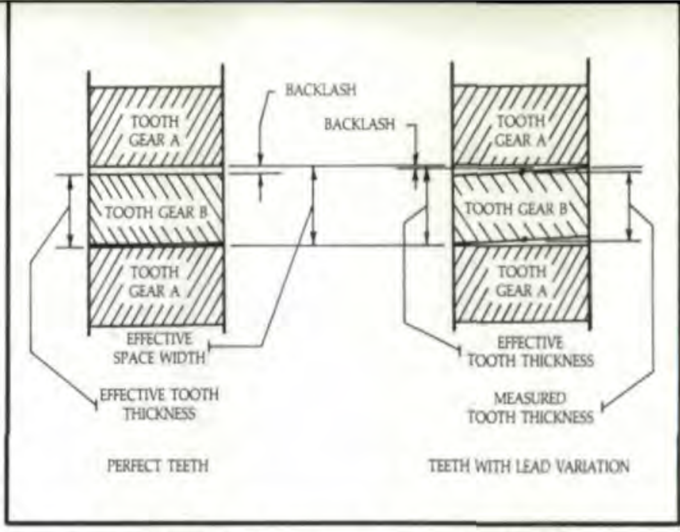

Fig. 6 – Effect of lead variation on measured tooth thickness.

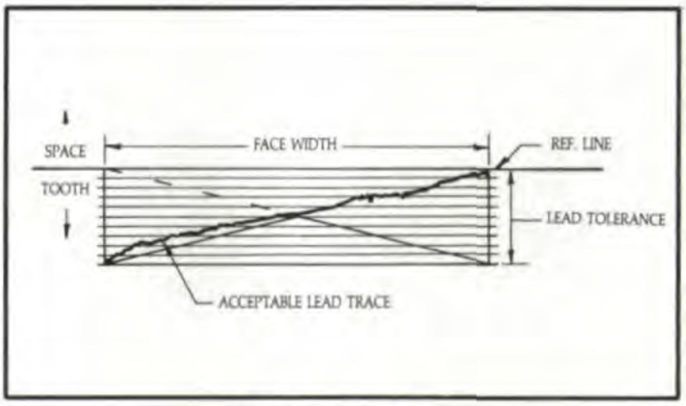

Fig. 7 – Lead "K" Chart.

Lead may be considered as the amount that a point on an active profile at one end of a tooth is ahead or behind the position of a similar point on the same active profile at the other end of the tooth. Lead is generally measured on a gear measuring machine called a lead checker. A stylus is moved along the length of the tooth surface to determine the amount that the surface departs from a theoretical helix at that given diameter.

A typical lead checking machine produces a trace on a recording chart which is a record of the lead variations found in the tooth being measured. The allowable magnitude of lead variations is usually specified by means of a lead "K" chart. (See Fig. 7.)

For a given AGMA quality class, any lead trace recorded by the lead measuring machine that will fit within the shaded area of the lead chart is acceptable. A point at one end of a given helix can be ahead or behind a similar point at the other end of the same helix by the amount of the lead tolerance shown by the "K" chart. This also applies to the helix on the other side of the tooth. Thus, a condition recognized as taper is permitted. The "K" chart indicates that the allowable deviation from the theoretical lead at the middle of the tooth is 1/2 of the allowable lead variation for the given AGMA quality level.

As shown previously, the effective tooth thickness is an envelope value of tooth thickness. It is the maximum metal condition for each specific gear. The measured tooth thickness, as obtained by means of a measurement over wires, a chordal tooth measurement or a span measurement, is only a local value. It is a linear dimension between two similar points on a given tooth. These points represent only a very small part of the total surface of the tooth. Also, these points are evaluated near the mid-height of the tooth and usually at the middle of the length of the tooth. Each of the measuring methods involve only one or two of the tooth element tolerances, lead, pitch, profile or runout, in various combinations.

Spacing. Fig. 8, left illustration, shows a cross section of two teeth of a perfect pair of gears in mesh in their meshing plane. The teeth are shown at the phase of the meshing cycle when there are two pair of teeth in contact. (See Fig. 2, top and middle illustrations.) Fig. 8, left illustration, shows the teeth of a perfect pair of gears in mesh. The right illustration shows the effect of a spacing variation. When one tooth of Gear A is in contact with Gear B, the next tooth is displaced by the magnitude of the spacing variation, thus, increasing the effective tooth thickness. Spacing (pitch) may be viewed as the amount that a given tooth is ahead or behind its correct position along its pitch circle relative to its adjacent tooth; thus, spacing variations over a group of teeth can be cumulative.

Profile. Profile is generally measured by means of a profile checking machine. The machine traverses a stylus along

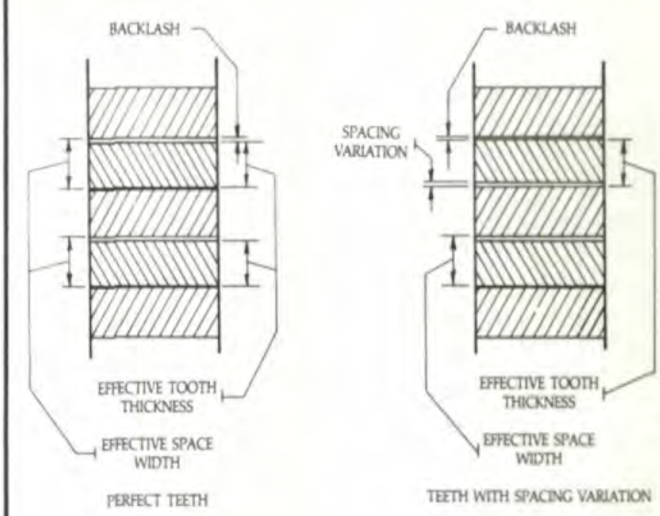

Fig. 8 – Effect of spacing variation on measured tooth thickness.

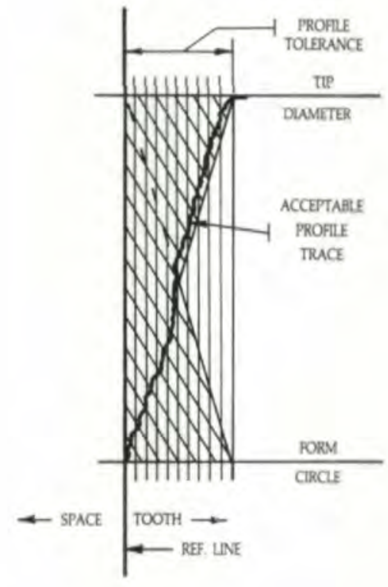

Fig. 9 – Profile "K" chart.

the active face of the tooth from root to tip. A chart is produced which shows the departure of the actual profile from a theoretical involute profile for the tooth being evaluated.

The allowable magnitude of profile variation is usually specified by means of profile "K" chart. (See Fig. 9.) For any given AGMA quality class, any profile trace that will fit within the shaded area of the profile chart is acceptable. The chart indicates that the allowable deviation from the theoretical profile at mid-height is 1/2 of the allowable variation for each quality level. For example, for a gear of a size and AGMA quality number permitting a .002" profile tolerance, the maximum metal condition could exceed the mid-height value by .001". In such a case, the effective tooth thickness would exceed the measured tooth thickness by .001" on each side of the tooth, a total of .002".

Runout. Runout may be measured by placing a measuring wire in each successive tooth space as the gear is rotated about its axis under an indicator. A radial reading is taken at each position. Runout is the maximum variation from high to low readings of all of the tooth spaces in the gear. It is due to an eccentric condition between the circle on which the teeth were cut and the axis of rotation, or due to an out-of-roundness of the gear. Runout in a gear as "seen" by its mating gear appears as variations in tooth thickness. Equation 18 shows the relationship among space width, tooth thickness and circular pitch on the standard pitch circle. The relationship between radial measurement of out-of-roundness variation (runout), ΔC, and the tooth thickness variation is given in Equation 19.

Tooth Thickness Adjustment Factor

In order to determine the drawing specification value of tooth thickness measurement for each of the measuring methods, over wires, chordal tooth thickness or span measurement, it is necessary to determine the value of tooth thickness adjustment factor for the method to be specified. This value is subtracted from the calculated value of effective tooth thickness to achieve a value of measured tooth thickness. The value of measured tooth thickness is then used to calculate the specific drawing dimensions required by the chosen measuring technique.

The magnitude of the tooth thickness adjustment factor, Δt, depends on the specific method of tooth thickness measurement to be specified. In each of the tooth thickness measuring methods there is a unique combination of tooth element variations that enter into the calculations, providing a stack-up of tolerances. Since it is reasonable to assume that these tolerances will exhibit a normal distribution, the root mean square of all of the individual element tolerances that enter into each specific measurement is used. This gives better than a 95% assurance that the effective tooth thickness will not be exceeded.

Measurement Over A Single Wire. The measured tooth thickness of a gear is obtained most directly by a measurement over a single wire. This measurement provides the chordal distance between similar points on the profiles on each side of a tooth space at a specific distance from the axis of rotation. A gear measuring wire, made to precise tolerances on roundness and diameter, is placed in a tooth space, and a radial measurement is made from the axis of rotation of the gear to the top of the wire. The wire typically contacts the tooth at about mid-height. The quality variations that enter into this measurement are lead (each side), profile (each side) and runout. According to the profile "K" chart, the tip or the root of a tooth can be plus metal by up to 1/2 of the profile tolerance. This is also true of lead. It is also reasonable to assume that with two measurements taken at random, the mid-point of runout will be measured.

Tooth Thickness Adjustment Factor

$$\Delta t_1 = [2\,(\tfrac{1}{2}v_P)^2 + 2(\tfrac{1}{2}v_L)^2 + \tfrac{1}{4}v_{TR}^2 + v_S^2]^{1/2}$$

where

$$v_{TR} = 2 \tan \Phi\, v_R$$

The measured tooth thickness, T_{M1}, on which the drawing values of measurement over one pin are based, may be calculated as follows:

$$T_{M1} = T_E - \Delta t_1$$

The value of measured tooth thickness is used to calculate the specified measurement over one wire. (See Equation 12.)

Gear Tooth Vernier Measurement. The gear tooth vernier caliper measures the normal chordal distance between the profiles of a tooth at a specified distance from the top land (outside diameter) of the tooth. A gear tooth vernier is a type of vernier caliper which includes a special blade that can be pre-set to a drawing value of chordal addendum to establish the distance from the top land to the point on the teeth where the measurement is to be taken. Equation 13 may be used to calculate the chordal addendum setting. The distance between the jaws is read as they make contact with the sides of the tooth. This is the chordal tooth thickness. This value can be converted to measured arc tooth thickness by Equation 14. Contact between the sides of the tooth and the jaws of the tooth vernier caliper occurs over a relatively small area of the tooth in most cases. Thus, the measurement does not include either the profile or the lead variations. Since only a single tooth is evaluated at one measurement, spacing is not included. Also, the diameter at which the jaws contact the tooth is fixed by the outside diameter, which is usually machined independantly of the teeth; thus,the full effect of


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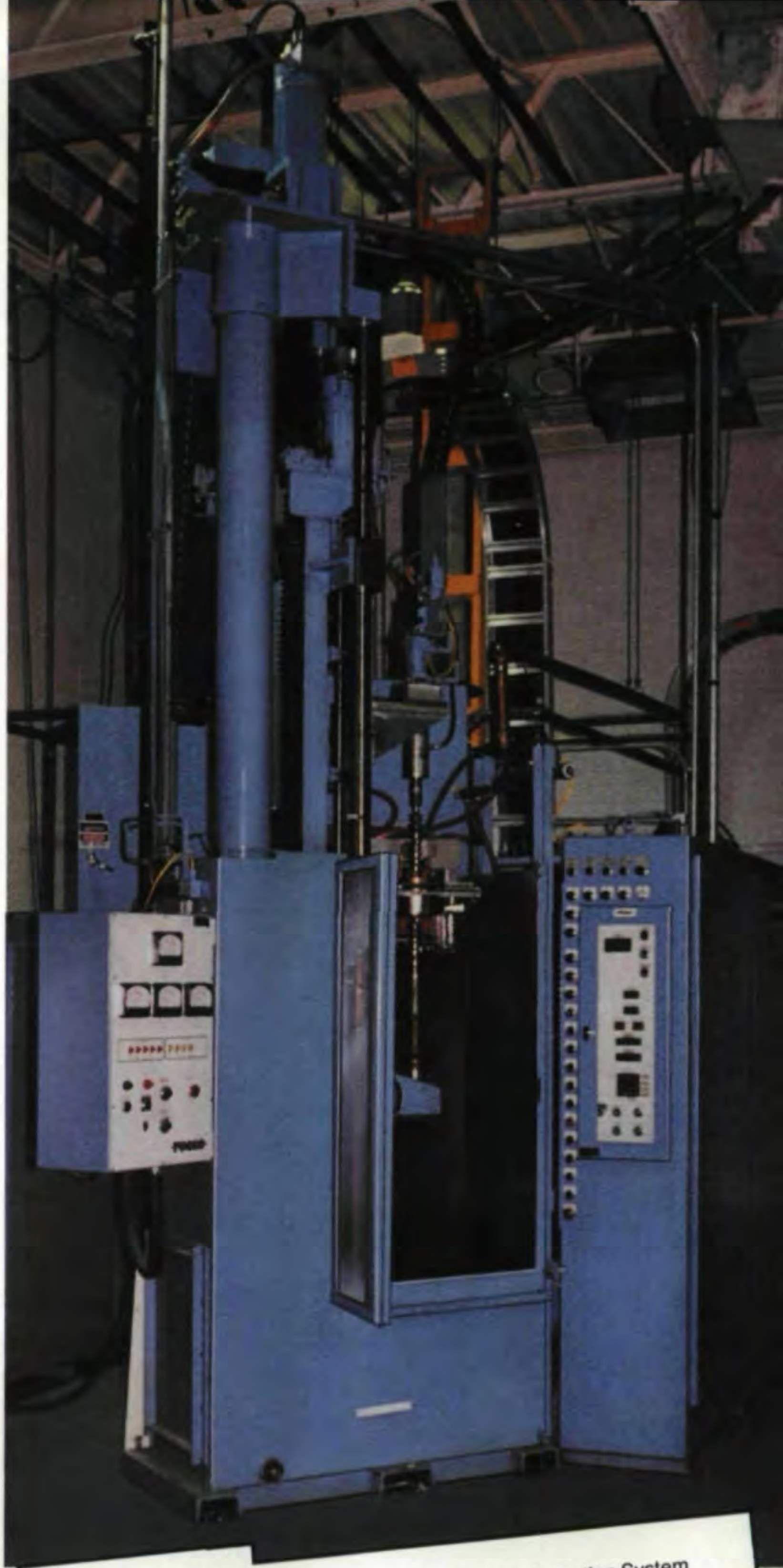

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runout may be present.

Tooth Thickness Adjustment Factor

$$\Delta t_c = [2(\tfrac{1}{2}v_P)^2 + 2(\tfrac{1}{2}v_L)^2 + \tfrac{1}{2}v_{TR}^2 + v_s^2]^{1/2}$$

The measured tooth thickness, T_{Mc}, on which the drawing values of chordal tooth thickness measurement are based, may be calculated:

$$T_{Mc} = T_E - \Delta t_c$$

The vernier caliper setting for chordal tooth thickness may be obtained from Equation 14.

Measurement Over Two Wires. In this technique, a gear measuring wire is placed in each of two tooth spaces near or at opposite ends of a diameter, and a measurement made across the tops of the wires. The wire diameters are chosen such that the wires will contact the tooth surfaces near their mid-height. Contact conditions are similar to the over one wire measurement, except that the measurement is not related to the gear axis.

Tooth Thickness Adjustment Factor

$$\Delta t_2 = [2(\tfrac{1}{2}v_P)^2 + 2(\tfrac{1}{2}v_L)^2 + v_S^2 + V_{TR}^2]^{1/2}$$

The measured tooth thickness, T_{M2}, on which drawing values of measurement over two pins are based, may be calculated from:

$$T_{M2} = T_E - \Delta t_2$$

The value of measured tooth thickness is used to calculate the specified measurement over two wires. (See Equations 15 or 16.)

Span Measurement. In this technique, a measurement is made over a group (span or block) of teeth using a conventional vernier (dial) caliper. The number of teeth included within the span will determine where the contact will take place on the teeth. In most cases, a number of teeth to be included in the span is selected which will provide contact near the mid-height of the teeth. In addition to the effects of profile and lead on the sides of two teeth, there is the effect of the pitch accumulation produced by the number of teeth within the span. The full effect of spacing is included in the measurement.

Tooth Thickness Adjustment Factor

$$\Delta t_s = [2(\tfrac{1}{2}v_P)^2 + 2(\tfrac{1}{2}v_L)^2 + v_{TR}^2]^{1/2}$$

The measured tooth thickness, T_{MS}, on which the drawing values of span measurement are based may be calculated as follows:

$$T_{MS} = T_E - \Delta t_s.$$

Appendix A
General Equations for Tooth Thickness
(Parallel Shaft Gearing)

Standard Center Distance

$$C = (N_P + N_G) / (2\ P_d) \qquad (1)$$

Standard Pitch Diameter

$$D = N/P_d \qquad (2)$$

Operating Pressure Angle

$$\Phi' = \cos^{-1}(C \cos\Phi / C') \qquad (3)$$

Transverse Diametral Pitch

$$P_d = P_{nd} \cos\Psi \qquad (4)$$

Pitch Diameter of Master Gear

$$D_M = N_M / P_d \qquad (5)$$

Operating Pressure Angle
(On a Gear Rolling Fixture)

$$\Phi'' = \cos^{-1}(C \cos\Phi / C_M) \qquad (6)$$

Tooth Thickness of Work Gear
(From a Gear Rolling Fixture Measurement)

$$T_W = [(\mathrm{inv}\Phi'' - \mathrm{inv}\Phi)(D_M(N + N_M) + \pi D_M) / N_M] - T_M \qquad (7)$$

Operating Pressure Angle
(Gear Rolling Fixture)

$$\Phi'' = \mathrm{inv}^{-1}\ \mathrm{inv}\Phi + [N_M(T_M + T_W) - \pi D_M] / [D_M(N_M + N)] \qquad (8)$$

Testing Center Distance Specification
(From Value of Functional Tooth Thickness)

$$M_M = C \cos\Phi / \cos\Phi'' \qquad (9)$$

Pressure Angle To Center of Wire
(Measurement over 1 wire)

$$\Phi_2 = \mathrm{inv}^{-1}[(T_{M1}/D_w) + \mathrm{inv}\Phi + (d_w/D_b) - (\pi N)] \qquad (10)$$

Radius to Center of Wire

$$R_W = D_b / \cos\Phi_2 \qquad (11)$$

(continued on page 36)

Selection of a Proper Ball Size to Check an Involute Spur or Helical Gear Tooth

Carroll K. Reece
Harlan Van Gerpen
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Abstract:

The selection of a properly sized ball or pin to use in the check of dimension over or between balls or pins is not an easy task. If the ball size is taken from standard tables of Van Keuren Pins for standard pitches, and if the proportion of the tooth being checked is not standard, interference with some feature of the gear tooth may result, giving an erroneous reading for the tooth thickness. This article gives a procedure for selecting a properly sized ball or pin to check an involute tooth of any proportion. A set of standard sizes for both inch and metric balls or pins is suggested.

Introduction

A much-used method for checking the tooth thickness of an involute gear tooth is to measure the dimension over two balls placed in most nearly opposite spaces in the case of external gears, and the dimension between the balls in the case of internal gears. This measurement is then checked against a pre-calculated dimension to denote an acceptable part.

From this point, references will be to external gears only. It is confusing if the text is correct for both external and internal gears because the words "over" and "between" are continually interchanged when addressing the dimension over or between balls. Confusing, also, are references to major diameter of the external and to minor diameter of the internal, and to the the root of the external and the major circle of the internal.

However, calculations for external and internal gears are the same if a minus one factor is entered at certain points in the equations for internal gears. The formulas shown will properly calculate either external or internal gears and splines.

The dimension over balls is a measure of the thickness of a theoretically true involute tooth generated from the base circle used in the equations at the ball contact point. The ball contact point is the point on the tooth where the stated measuring ball contacts the tooth on both sides of the tooth space. For standard gears the size of the measuring ball is usually chosen so the ball contact point is near the gear pitch circle.

Tables of sizes over balls have been published for standard gears using standard increments based on constants for external gears and internal gears, the ball sizes being determined by dividing the constant by the diametral pitch of the gear set. These ball sizes are satisfactory for purely standard gears. But for gears with teeth which are of non-standard proportions, the pitch circle is sometimes not near the middle of the tooth height and is sometimes off the tip of the gear tooth or below the gear tooth root. In the case of non-standard gears, a measuring ball size which contacts the gear tooth somewhere near the midpoint between the form circle and the outside circle must be chosen.

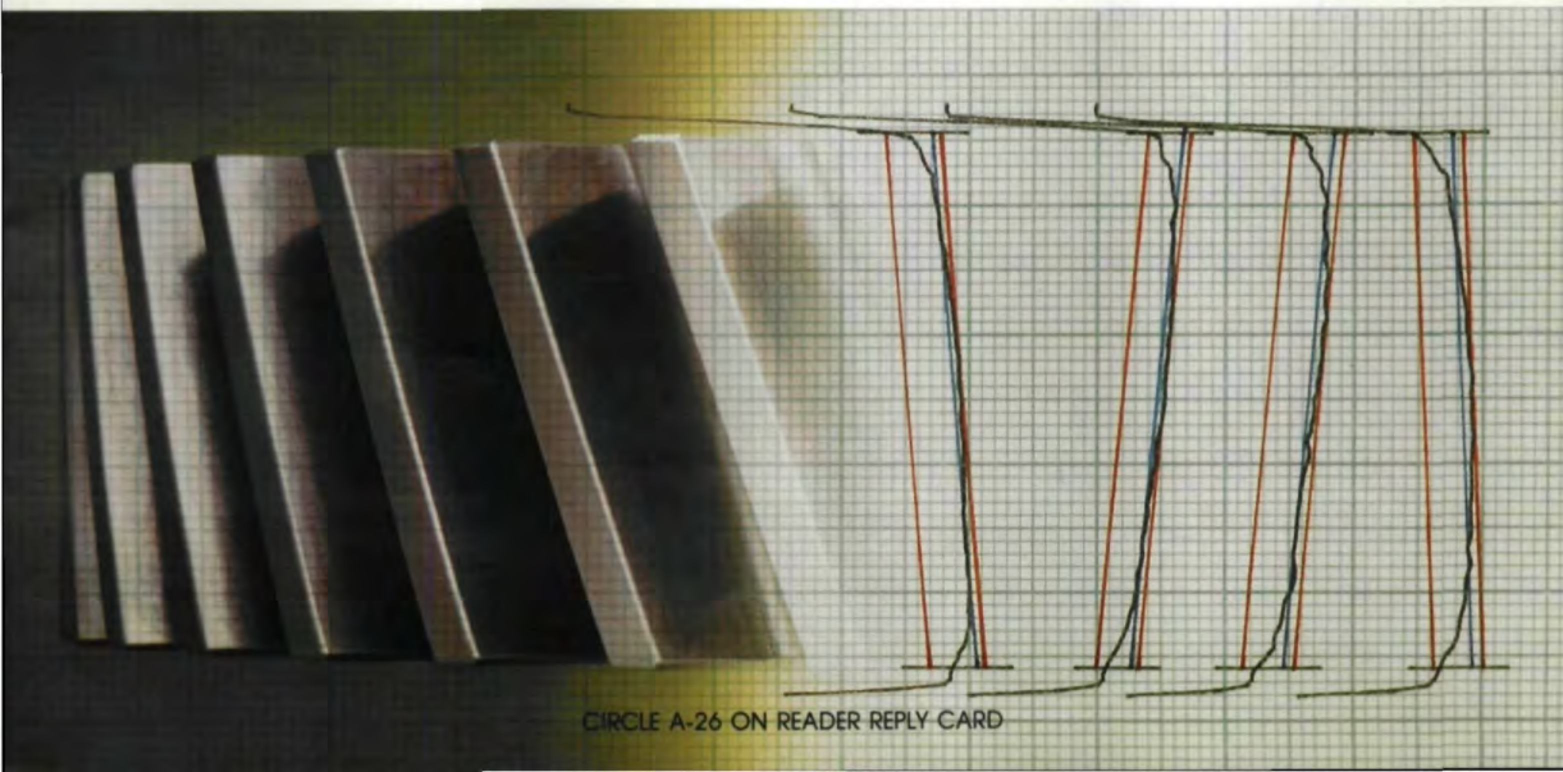
Exact Ball Size

The exact measuring ball size needed to contact this point is calculated by using Equations (2), (3) and (4).

The size of the measuring ball determined by Equation (4) would be different for every gear tooth designed with nonstandard proportions and if allowed to stand, would create a difficult, if not unmanageable, system of measurement. Keeping the ball contact point at the mid-height of the tooth is not a critical factor so long as the ball contacts the involute surface, and the problem has been resolved by changing the measuring ball size as determined by Equation (4) to a size found in a predetermined standard set of measuring ball sizes.

Standard Ball Sizes

Any set of standard ball sizes may be used to determine the final ball size. The size increments must be small enough so that the ball contact point does not move too far radially on the gear tooth when the ball size is changed from the exact size determined by Equation (4) to one of the standard increments. Experience has shown that results will be somewhat better if the exact ball size is changed to the next higher standard ball increment, thereby moving the ball con-



tact point toward the gear outside circle. However, rounding to the nearest standard size or moving to the next larger or the next smaller size is a matter of choice. Any method will yield satisfactory results if the proper checks are made for interferences with important tooth features.

Experience has also shown that within the range of balls normally used for measuring gears, a 1/64" increment for the sizes of balls is satisfactory and actually gives more selections than are needed. Since the balls are not outrageously expensive, this selection is satisfactory for inch sizes.

The R40 series from ISO recommendation R-3 has been selected as the standard set of metric pins for measuring ISO 4156 and ANSI B92-1980 standard metric splines. A good feature of this selection of sizes is that the size increment increases as the size of the pin (ball) increases. The R40 series gives very satisfactory results for metric ball sizes for measuring gears.

Interference

When a standard size ball is substituted for an exact ball size, the contact point moves from the exact mid-point of the tooth. This change must not cause the ball to interfere with any feature of the gear tooth. To insure this, calculate the radius to the standard ball contact point, the radius over the balls and the radius under the balls.

The following features of the gear tooth must be checked. **The root surface of the tooth.** To get a good measurement of the gear tooth thickness, the ball must rest on the involute tooth side surfaces. It is very difficult to tell in some cases whether the ball is resting on the root surface or the involute sides of the tooth when measuring a gear. It is much better to check this by calculation when determining the dimension over balls so as not to leave this responsibility to the shop personnel when the gear is being cut. The check here is to compare the radius under the balls with the gear root radius. If interference occurs, the ball size must be moved to the next higher standard increment and the calculations for interference repeated.

The ball contact point must not be below the gear involute form point. Equations (5), (6) and (7) assume that the involute form extends from an infinitely large diameter to the base circle; whereas, it actually only extends from the tooth tip to the form point. Therefore, a separate check must be made to be sure the ball contact point is not below the form point. Again, if this interference occurs, the solution is to move the ball size to the next higher increment.

The ball contact point also must not be outside the gear outside circle; otherwise the ball will rest on the corners of the teeth at the gear outside circle. This will result in an erroneous reading. If the balls are found to be resting on the corners of the teeth, change the ball size to the next lower standard increment and repeat the checks for interference.

The radius over the balls should not be less than the gear outside circle radius because the anvils of the measuring micrometer may rest on the corners of the gear teeth. This problem is solved by increasing the ball size to the next higher increment.

A situation can occur where the ball size is being moved up and down in an effort to avoid an interference only to have interference occur at some other point. When that happens, the ball size should be moved back to the original exact size found by Equation (1). If that ball is resting on the root surface, a calculation is made to determine how much

(Continued on page 31)

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HARLAN VAN GERPEN *has 28 years of experience at the John Deere Product Engineering Center in Waterloo, Iowa, where he served as manager of technical services and a principal engineering specialist. He is a licensed professional engineer in the State of Iowa and a member of the ASAE where he is a member of the Research Committee and the Instrumentation and Controls Committee. He is also a member of the SAE Education and Human Factors Committees. Mr. Van Gerpen holds a master's degree in electrical engineering from the University of Illinois.*

CARROLL K. REECE *worked for the John Deere Product Engineering Center, Waterloo, Iowa, as an engineering design analyst and supervisor in the mechanical elements department. He has served for 25 years on the ANSI-B92 National Spline Standards Committee and for 18 years on both the ISO TC 32 International Spline Standards Committee and the ANSI-B6 National Gear Standards Committee. He holds a master's degree in agricultural engineering from Kansas State University and is a licensed professional engineer in the State of Iowa.*

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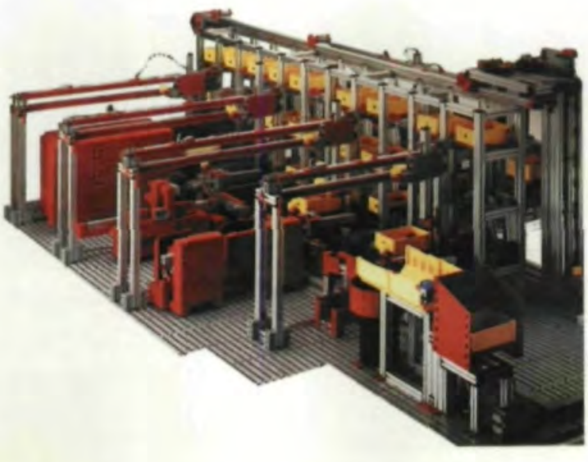

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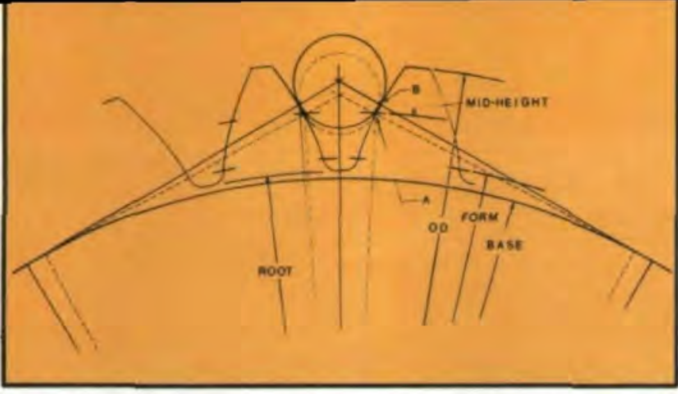

Fig. 1 – In selecting the measuring ball size for a full depth gear tooth, the next standard increment larger than the exact size is usually satisfactory.

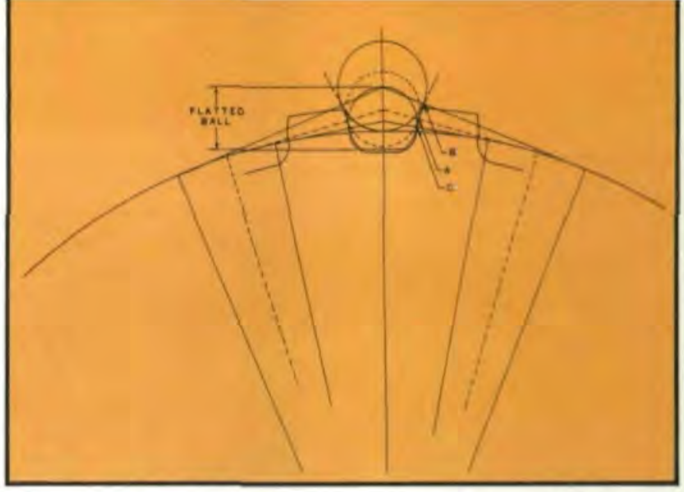

Fig. 2 – In this example, the first standard increment larger than the exact ball contacts outside the outside circle limit; the next size smaller contacts the root circle. The ball must be flattened to get a good measurement. This happens with stub tooth gears with low pressure angle.

the ball must be flattened to clear the root surface. From Equation (2) it can be noted that the exact ball size contact point will not be below the form point, nor will it be outside the gear outside circle, but the ball may be resting on the gear root surface. Even a standard ball may contact the root of the tooth. This condition must always be checked, whether using a standard ball or an exact ball size.

Fig. 1 illustrates the process of selecting the ball size for measuring a full depth gear. The dashed lines are the exact ball size as determined by Equation (4). Point A is the ball contact point for this ball and is chosen midway between the outside circle and the form circle by Equation (2). The solid lines represent the standard ball, and Point B is the ball contact point for the standard ball. Note that the standard ball clears the root of the gear tooth, and the radius over the ball is outside the outside circle of the gear. This ball would be satisfactory to check this gear. Finally, the dimension over the balls is calculated from Equation (10) or (11).

It can become a burdensome task to make the calculations outlined here only to discover interference at some point and have to do everything over again. The occurrance is unlikely in the case of full depth gears of the most often used pressure angles, but choosing a satisfactory standard ball size for gears and splines of unusual proportions can be a difficult task. Fig. 2 illustrates the type of gear or spline tooth which might pose a problem. The involute in Fig. 2 is very short. When the ball size is changed from the exact size represented by the dashed lines at Point A, the contact point moves off the tips of the teeth at Point B. The ball size is moved to the next lower standard increment, which is below the exact size. The contact Point C is below the center of the tooth, but is in the satisfactory range.

Sometimes when the involute surface is very short, Point B is outside the outside circle, and Point C is below the form point. In this case, the exact ball size is used. In Fig. 2 the standard ball at Point C contacts the root of the gear so a flattened dimension is calculated.

A computer program can be written which can accomplish this task very quickly and make all the checks necessary to guarantee good results.

One might question why a gear or spline tooth like the one shown in Fig. 2 would ever be designed, but the task of the computer program is not to decide what is reasonable. So long as the input data is valid, the computer program should determine a proper ball size and the dimension over those balls.

Fig. 3 shows a real master gear for which the data was presented to a computer program written according to the system outlined above for choice of a measuring ball. The program picked a standard inch ball and calculated a checking dimension over balls without difficulty.

Summary

Following is a summary of the steps to follow in picking

Fig. 3 – This photograph shows an actual master gear which has proportions similar to those shown in Fig. 2. The method outlined chose a measuring ball size without difficulty.

a measuring ball size and finding the dimension over balls.

1. Find the normal base tooth thickness from Equation (1).
2. Find the midpoint of the tooth between the form point and the outside circle from Equation (2).
3. Calculate an exact ball size to contact this point from Equation (3) and (4).
4. Change the ball size from the exact size to a standard size either larger or smaller.
5. Check for interferences with important features of the gear tooth using Equation (5), (6), (7), (8) and (9).
6. Move the ball size in the appropriate direction if there are interferences. Repeat the checks.
7. Change the ball size back to the exact ball size and check for interferences if a satisfactory standard ball cannot be found. Calculate the flattened size if the ball contacts the root.
8. Finally, calculate the dimension over balls using Equation (10) or (11).

Equations

BTN is the normal tooth thickness at the base circle. If the tooth thickness is given at any other point on the tooth, BTN must be found. Use Equation (1).

$$BTN = BD*Cos\ (BHA)*\frac{(TTDD}{(Cos\ (HADD)*DD)} + K*Inv(Acos\ (BD/DD))) \quad (1)$$

Equation (2), (3) and (4) are for choosing the exact ball size.

$$PACP = Acos\ \frac{((2*BD)}{(OD + FD))} \quad (2)$$

$$\left. K*Inv(PACP) \right] + PACP$$

$$D = K*((Inv\ (PACB) + \frac{K*\pi}{Z}) *Cos\ (BHA)*BD - K*BTN) \quad (4)$$

After the standard ball size has been selected use Equations (5), (6), (7), (8) and (9) to determine if the standard ball size is satisfactory.

$$PACB = Ainv\left[\frac{K*(BTN + DS)}{(Cos\ (BHA)*BD)} - \frac{K*\pi)}{Z}\right] \quad (5)$$

$$RCB = \frac{BD}{2*Cos\ (PACB)} \quad (6)$$

$$RCP = \sqrt{\left[(RCB*Sin\ (PACB)) - \frac{K*DS}{2*Cos\ (BHA)}\right]^2 + \frac{(BD)^2}{2}} \quad (7)$$

$$ROB = RCB + \frac{DS}{2} \quad (8)$$

$$RUB = RCB - \frac{DS}{2} \quad (9)$$

When the final selection of ball size is satisfactory, use Equation (10) or (11) to find the dimension over balls, depending on whether the number of teeth is even or odd.

If Z is odd:

$$DBALL = 2 * RCB * Cos(90/Z) + K * DS \quad (10)$$

(continued on page 47)

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Effects of Hob Quality and Resharpening Errors on Generating Accuracy

Brian W. Cluff
American Pfauter Limited
Elk Grove Village, Illinois

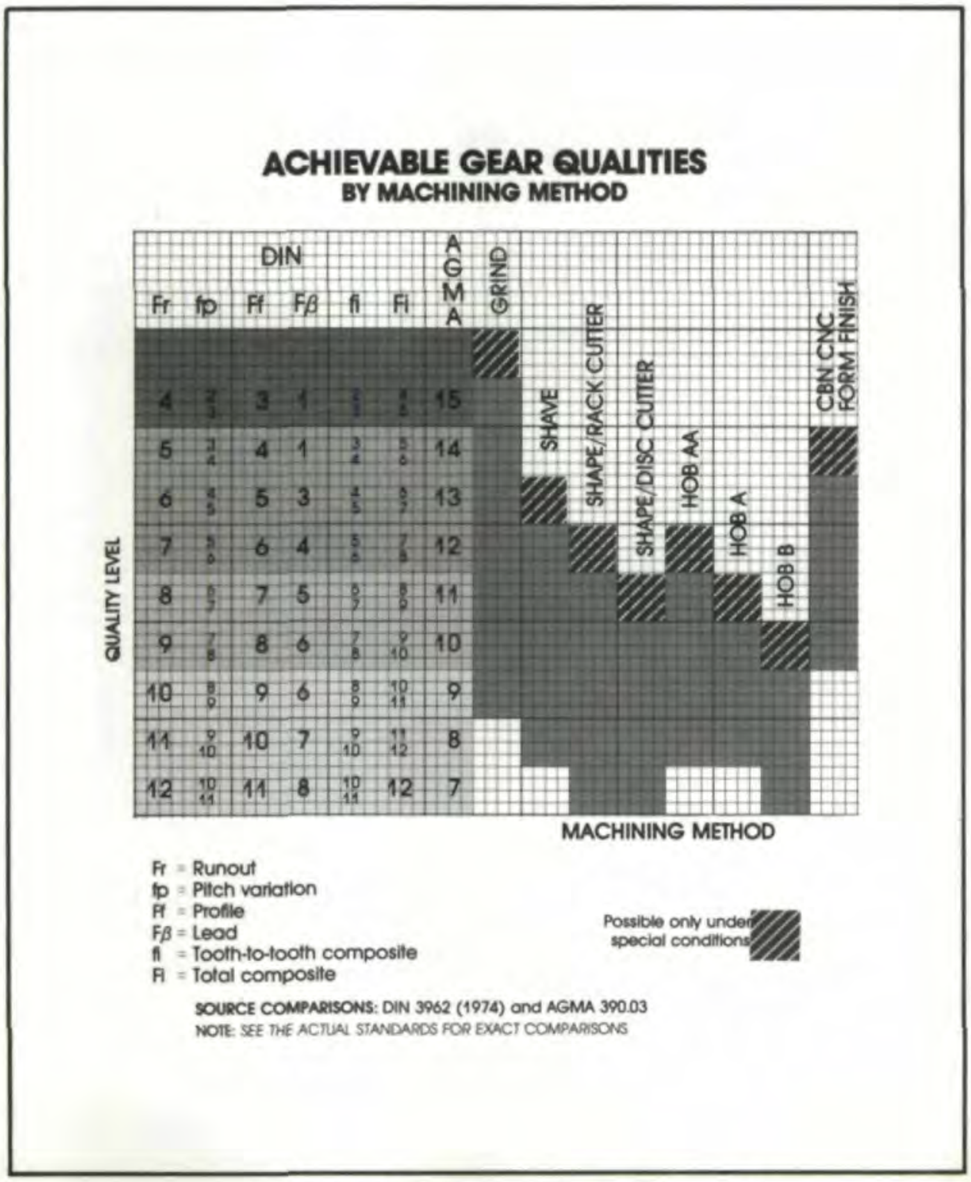

Fig. 1 – Achievable accuracies by machining method.

Introduction

The modern day requirement for precision finished hobbed gears, coupled with the high accuracy characteristics of modern CNC hobbing machines, demands high tool accuracy.

Modern CNC hobbing machines are capable of producing gears with lead and pitch accuracies of AGMA 14-15, but are still limited by the manufactured accuracy of the hob to a lower quality level on the involute profile (Fig. 1). For high accuracy hobbed profiles, high accuracy hobs are necessary.

The geometric peculiarities of the involute worm, from which the hob is derived, must be clearly understood to avoid loss of hobbed accuracy. Purchased tool accuracy and tool resharpening maintenance bear scrutiny in order to preserve hobbed accuracy.

Geometrical Peculiarities of Hobs

A hob is derived from the involute helicoid worm. A hob is a rotating cut-

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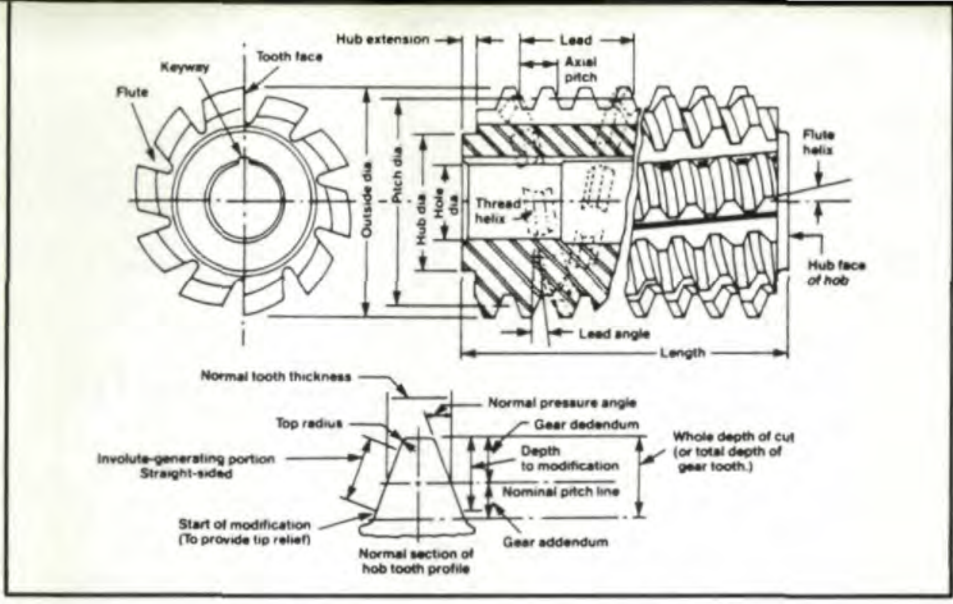


Fig. 2 – Geometrical elements of a typical cylindrical hob.

ting tool with its cutting edges arranged along a helix. It is used for generating gear teeth or other forms in a cylindrical workpiece (Fig. 2).

A hob is a reducing cylinder. Over its usable lifetime its diameter gets smaller due to repeated sharpenings. Each time it is resharpened it changes size relative to the amount of outside (tip) clearance and flank clearance (cam or backoff).

Every hob is designed with a basic (generating) rack profile which defines the pressure angle, the addendum and dedendum, the fillet radius, design modifications of the addendum profile, and design modifications for preshave, pregrind, preroll and prehard finish.

The Nature of the Enveloping Cut

Deviations from the theoretical or design generating helix of the hob (Figs. 3 and 4) effect the polygonal path of the enveloping cut along the gear tooth profile.

Figs. 3 and 4 show a single thread hob. In one revolution of the hob each of the 12 cutting edges removes metal from the tooth space enveloping the profile. The profile is made up of a series of individual cuts. The more cutting edges in a hob, the finer the network of enveloping cuts. The fewer the number of cutting edges in the hob, the rougher the involute profile.

If the hob is manufactured with deviations along its generating helix (thread error) or is resharpened so as to displace one or more cutting edges from the nominal pitch line cylinder of hob, the effect is a deviation in the network of enveloping cuts. This deviation manifests itself as profile error (Fig. 5).

Incorrect resharpening of the hob produces deviations in the design geometry which effect the basic rack tooth form of the hob, the position of one cutting edge to another, the rake of the hob cutting edge, and the lead of the gash (whether straight or spiral). These deviations are reproduced in varying magnitudes on the involute profile of the gear.

Mounting a theoretically perfect hob on an eccentrically running arbor causes the hob cutting edges to advance and retract in one revolution. This causes an advance and retreat of the network of enveloping cuts from the nominal, producing a "wandering" involute profile.

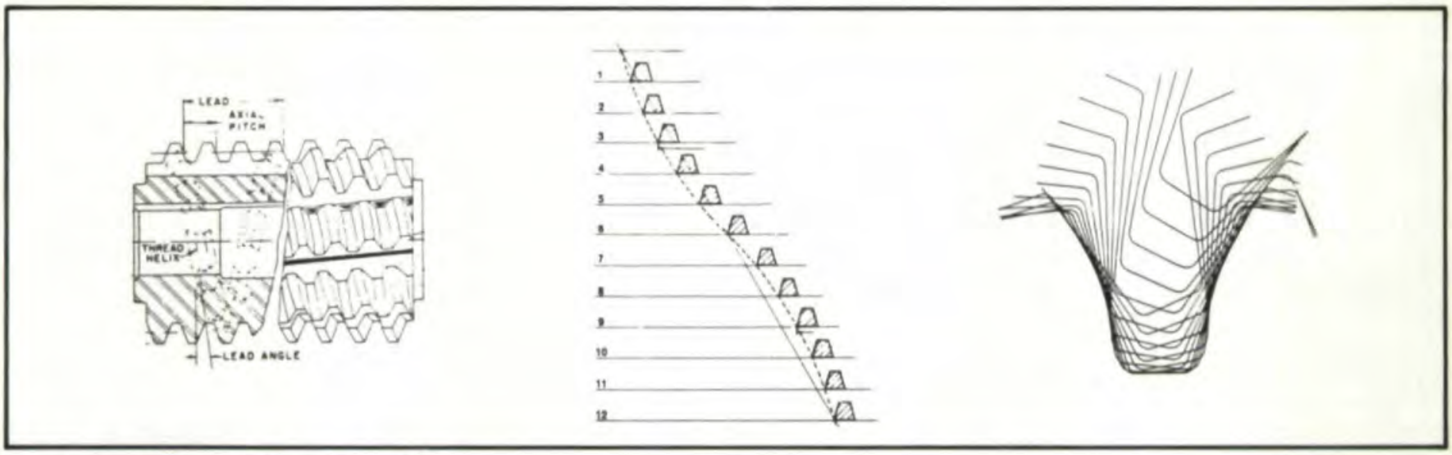


Fig. 3—A single thread hob in one revolution envelopes a tooth space with a series of polygonal cuts. (In this instance 12 gashes are shown.)

Achievable Profile Accuracies by Finish Hobbing

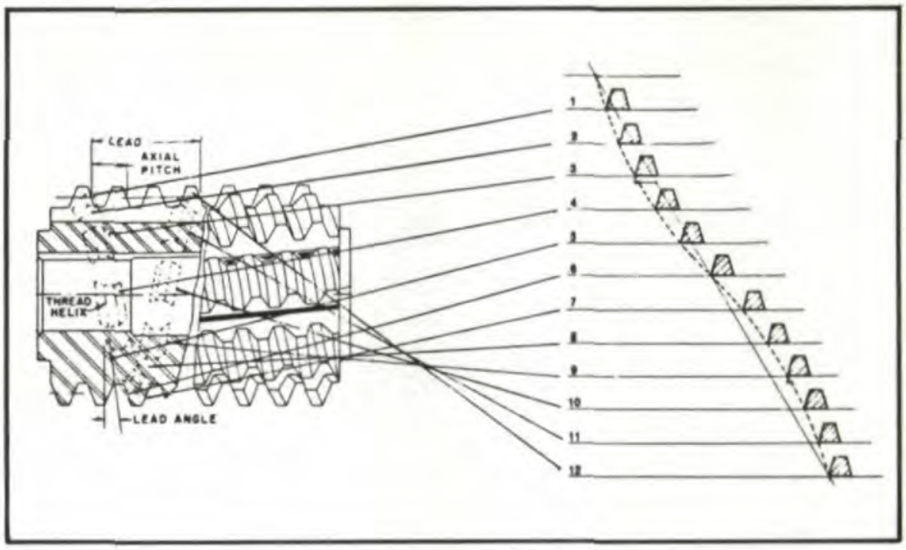
For most CNC hobbing machines the burden for involute accuracy rests with the hob. Pitch and lead accuracy are built into the machine kinematics and alignment characteristics as machine manufacturing tolerances.

Finish hobbed gear profile accuracies are directly related to manufactured hob class accuracy, mounting accuracy on the hobbing machine and resharpening accuracy.

Typically, a Class AA single thread hob can produce an AGMA Class 12 profile, a Class A hob can produce an AGMA Class 11 profile, and a Class B hob can produce an AGMA Class 10 profile in 10" to 20" PD gears, 3 DP to 20 DP (Fig. 6). This assumes a hob with adequate gashes, correctly resharpened to the tolerance requirements for its accuracy class (Figs. 7 and 8) and correctly mounted on the hob arbor in the hobbing machine within the runout value tolerance for its manufactured class accuracy.

To determine the profile accuracy to which a specific accuracy class hob can produce, read the tolerance value in tenths from the AGMA Hob Standard 120.01 for the characteristic "Lead. . . . In Any One Turn of Helix" (Fig. 7). The value "Lead. . . . In Any One Turn of Helix" refers to the accuracy to which the hob manuacturer produces the thread of the hob. It is the manufacturer's allowed deviation along the generating helix of the hob. It is the allowed wandering of the cutting flanks of the hob in one

Fig. 4 – The unwound generating helix of a 12 gash single thread hob, shown here, displays a deviation of the cutting edges (dotted line = thread lead error) from the nominal (solid straight line).



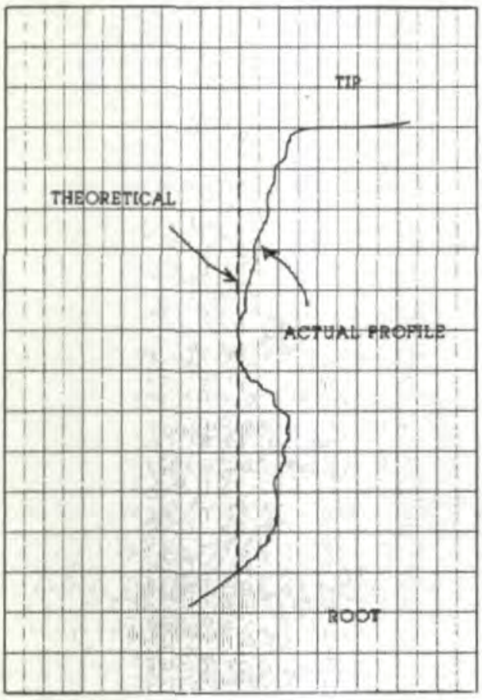
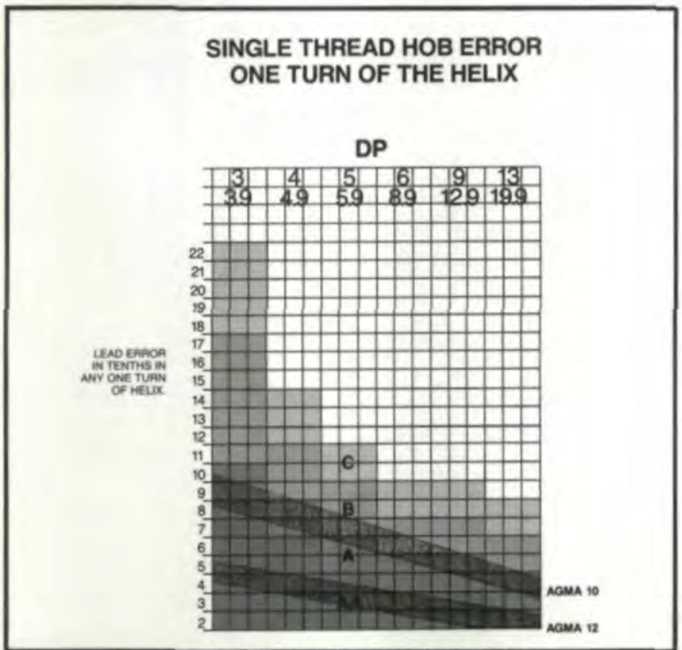


Fig. 5 – Profile error produced by manufactured deviations of the hob generating helix (thread).

Fig. 6 – Single thread hob error in one turn of the helix relative to hob class accuracy required to produce involute to AGMA 10 and AGMA 12 tolerances for 10" to 20" pitch diameter gears.



SINGLE THREAD HOB ERROR IN ONE TURN OF HELIX RELATIVE TO HOB CLASS ACCURACY REQUIRED TO PRODUCE INVOLUTE TO AGMA 10 & 12 TOLERANCES FOR 10" TO 20" DIAMETER GEARS.

Fig. 7 – Single-Thread Coarse-Pitch Gear Hob Tolerances[1] (In ten thousandths of an inch)

Hob Element	CLASS	NORMAL DIAMETRAL PITCH[2]							
		1 Thru 1.99	2 Thru 2.99	3 Thru 3.99	4 Thru 4.99	5 Thru 5.99	6 Thru 8.99	9 Thru 12.99	13 Thru 19.99
RUNOUT[3]									
Hub Face	AA	–	–	2	2	2	1	1	1
	A	8	5	2	2	2	2	2	2
	B	10	8	4	4	3	3	2	2
	C	10	8	4	4	3	3	2	2
	D	10	8	5	5	4	4	3	3
Hub Diameter	AA	–	–	2	2	2	1	1	1
	A	10	5	4	3	3	3	2	2
	B	12	8	6	5	4	4	3	2
	C	12	8	6	5	4	4	3	2
	D	15	10	8	8	6	6	5	5
Outside Diameter	AA	–	–	5	4	3	3	3	3
	A	30	20	15	15	10	10	10	10
	B	40	30	25	20	15	15	15	10
	C	50	45	40	25	20	17	17	12
	D	60	55	50	45	35	35	30	25
LEAD[3]									
Tooth to Tooth	AA	–	–	4	3	2	1.7	1.7	1.7
	A	7	5	4	3	2	2	2	2
	B	10	8	6	4	3	3	3	3
	C	15	12	8	6	5	4	4	4
	D	25	20	16	14	12	10	10	8
In Any One Turn of Helix	AA	–	–	8	6	4	3	3	2
	A	25	18	10	8	6	5	5	4
	B	35	25	17	11	9	7	7	6
	C	45	35	22	14	11	9	9	8
	D	60	50	40	30	25	20	20	18
In Any Three Turns of Helix	AA	–	–	12	9	6	5	5	4
	A	38	26	15	12	9	8	8	7
	B	53	38	22	16	12	11	10	9
	C	70	50	30	21	16	14	13	12
	D	120	100	80	60	50	40	35	25
TOOTH									
Pressure Angle[3,4]	AA	–	–	2	2	1.7	1.7	1.7	1.7
	A	10	5	3	3	2	2	2	2
	B	16	8	5	5	4	3	3	3
	C	25	15	10	5	4	3	3	3
	D	80	55	30	18	12	8	8	6
Thickness (minus only)	AA	–	–	15	15	10	10	10	10
	A	30	20	15	15	10	10	10	10
	B	30	20	15	15	10	10	10	10
	C	35	25	20	20	15	15	15	15
	D	40	35	30	25	20	20	20	20
Start of Tip Relief Modification (plus or minus)	AA	–	–	100	80	70	60	60	40
	A	200	180	160	140	120	100	80	60
	B	220	200	180	160	140	120	100	80
	C	220	200	180	160	140	120	100	80
	D	260	240	220	200	180	160	140	120
Symmetry in Start of Tip Relief Modification	AA	–	–	70	60	50	40	40	25
	A	150	130	120	100	90	80	60	50
	B	180	150	130	120	100	90	80	70
	C	180	150	130	120	100	90	80	70
	D	200	180	160	140	120	110	100	90

Fig. 8 – Single-Thread Coarse-Pitch Gear Hob Tolerances[1] (In ten thousandths of an inch)

Hob Elements	CLASS	NORMAL DIAMETRAL PITCH[2] 1 Thru 1.99	2 Thru 2.99	3 Thru 3.99	4 Thru 4.99	5 Thru 5.99	6 Thru 8.99	9 Thru 12.99	13 Thru 19.99
FLUTES									
Adjacent Flute Spacing[5]	AA	—	—	20	15	10	8	8	6
	A	40	30	25	20	15	10	10	10
	B	50	45	40	30	20	15	15	10
	C	50	45	40	30	20	15	15	10
	D	60	60	50	50	30	25	25	20
Non-Adjacent Flute Spacing[5]	AA	—	—	40	35	25	15	15	15
	A	80	60	50	40	30	30	30	25
	B	100	90	80	60	50	50	50	40
	C	100	90	80	60	50	50	50	40
	D	120	120	100	100	80	80	70	60
Rake To Cutting Depth[6]	AA	—	—	10	8	6	5	5	3
	A	30	15	10	8	6	5	5	3
	B	50	25	15	10	8	7	7	5
	C	50	25	15	10	8	7	7	5
	D	100	75	50	40	30	20	20	15

Flute Lead Over Cutting Face Width	CLASS	CUTTING FACE WIDTH in inches: Up to 1	1.001 to 2	2.001 to 4	4.001 to 7	7.001 & Up
	AA	8	10	15	20	20
	A	10	15	25	30	50
	B	10	15	25	30	50
	C	10	15	25	30	50
	D	15	23	38	45	75

HOLE	CLASS	HOLE DIAMETER in inches: 2.500	2.000	1.500	1.250	0.750	0.500 & Smaller
Hole Diameter (plus only)	AA	—	—	—	2	2	2
	A	8	8	5	2	2	2
	B	10	10	8	3	2	2
	C	10	10	8	3	2	2
	D	10	10	8	5	4	3

NOTE: 1. Tolerances apply only to standard hob sizes.
2. For combination pitch hobs, the coarser of the two pitches shall apply.
3. Total indicator variation.
4. Exclusive of Tip Relief Modification.
5. Compared against master index plate.
6. Radial (zero rake) tooth faces are standard.

Fig. 9 – AGMA 390.03 coarse pitch involute gear tolerance table for AGMA quality levels 8 through 12. Reprinted by permission of the AGMA, Arlington, VA.

AGMA QUALITY NUMBER	NORMAL DIAMETRAL PITCH	PROFILE TOLERANCE — PITCH DIAMETER (INCHES) 3/4	1½	3	6	12	25	50	100	200	400
8	1/2					42.6	47.7	53.1	59.1	65.7	73.1
	1				28.3	31.5	35.3	39.3	43.7	48.6	54.1
	2			18.8	21.0	23.3	26.1	29.0	32.3	36.0	40.0
	4		12.5	13.9	15.5	17.2	19.3	21.5	23.9	26.6	29.8
	8	8.3	9.3	10.3	11.5	12.8	14.3	15.9	17.7	19.7	21.9
	12	7.0	7.8	8.6	9.6	10.7	12.0	13.3	14.8	16.5	18.4
	20	5.6	6.2	6.9	7.7	8.6	9.6	10.7	11.9	13.2	14.7
9	1/2					30.4	34.1	37.9	42.2	46.9	52.2
	1				20.2	22.5	25.2	28.1	31.2	34.7	38.6
	2			13.5	15.0	16.7	18.6	20.7	23.1	25.7	28.6
	4		8.9	10.0	11.1	12.3	13.8	15.3	17.1	19.0	21.1
	8	5.9	6.6	7.4	8.2	9.1	10.2	11.4	12.6	14.1	15.6
	12	5.0	5.5	6.2	6.9	7.6	8.6	9.5	10.6	11.8	13.1
	20	4.0	4.4	4.9	5.5	6.1	6.8	7.6	8.5	9.4	10.5
10	1/2					21.7	24.3	27.1	30.1	33.5	37.3
	1				14.5	16.1	18.0	20.0	22.3	24.8	27.6
	2			9.6	10.7	11.9	13.3	14.8	16.5	18.3	20.4
	4		6.4	7.1	7.9	8.8	9.9	11.0	12.2	13.6	15.1
	8	4.2	4.7	5.3	5.9	6.5	7.3	8.1	9.0	10.0	11.2
	12	3.6	4.0	4.4	4.9	5.5	6.1	6.8	7.6	8.4	9.4
	20	2.9	3.2	3.5	3.9	4.4	4.9	5.4	6.1	6.7	7.5
11	1/2					15.5	17.4	19.3	21.5	24.0	26.7
	1				10.3	11.5	12.9	14.3	15.9	17.7	19.7
	2			6.9	7.6	8.5	9.5	10.6	11.8	13.1	14.6
	4		4.6	5.1	5.6	6.3	7.0	7.8	8.7	9.7	10.8
	8	3.0	3.4	3.8	4.2	4.6	5.2	5.8	6.4	7.2	8.0
	12	2.5	2.8	3.1	3.5	3.9	4.4	4.9	5.4	6.0	6.7
	20	2.0	2.3	2.5	2.8	3.1	3.5	3.9	4.3	4.8	5.4
12	1/2					11.1	12.4	13.8	15.4	17.1	19.0
	1				7.4	8.2	9.2	10.2	11.4	12.7	14.1
	2			4.9	5.5	6.1	6.8	7.6	8.4	9.4	10.4
	4		3.3	3.6	4.0	4.5	5.0	5.6	6.2	6.9	7.7
	8	2.2	2.4	2.7	3.0	3.3	3.7	4.1	4.6	5.1	5.7
	12	1.8	2.0	2.2	2.5	2.8	3.1	3.5	3.9	4.3	4.8
	20	1.5	1.6	1.8	2.0	2.2	2.5	2.8	3.1	3.4	3.8

enveloping revolution of the hob.

By comparing the lead tolerance in any one turn of the helix for a specific hob to the AGMA 390.03 profile tolerance table for gears (Fig. 9) it can be predetermined whether or not a particular profile tolerance can be finish hobbed.

The Effect of Hob Mounting Errors

Even if a hob is hypothetically perfect and manufactured error-free, it can produce profile errors if mounted eccentrically on the hobbing machine arbor.

Hob runout error due to either careless mounting or to improper sharpening is the greatest contributor to poor hobbed involute profiles. Figs. 10, 11 and 12 illustrate the effects three types of hob runout have upon the gear tooth form. These effects are created most often by:

1. Failure to true up the hob arbor
2. Failure to true up the hob on the hob arbor by indicating the hubs on the ends of the hob
3. Bent hob arbor
4. Oversize hob bore or undersize hob arbor
5. Non-parallel hob clamping spacers
6. Misaligned or worn outboard support bearing for hob arbor.

Often hob runout error is introduced at the first hob resharpening. If a hob is mounted carelessly – that is, without truing – on the sharpening arbor, runout can be sharpened into the hob by sharpening off progressively greater amounts of material from the hob gashes for half its rotation. The sources of this error in the sharpener are similar to those in the hobber.

In some precision gear manufacturing shops, the hob is sharpened on the hob arbor after careful alignment to insure optimum gear tooth profile accuracy.

The Effect of Hob Resharpening Errors

Fig. 13 illustrates the effects hob sharpening errors have on the basic rack profile of the hob and the resultant workpiece tooth profile. Figs. 14, 15, 16 and 17 illustrate diagrammatically typical resulting involute profiles. Figs. 9 through 11 illustrate the effects three types of hob runout have upon the involute profile due to careless mounting. Careless mounting of the hobs on the sharpening arbor can introduce the same error. A hob mounted on a bent resharpening arbor, for example, will be resharpened eccentrically, introducing the same error even if the hob is mounted concentrically on the hobbing machine arbor in the machine. Apart from runout errors four other errors can be introduced at the time of resharpening:

1. The hob cutting faces sharpened with incorrect lead (Fig. 14)

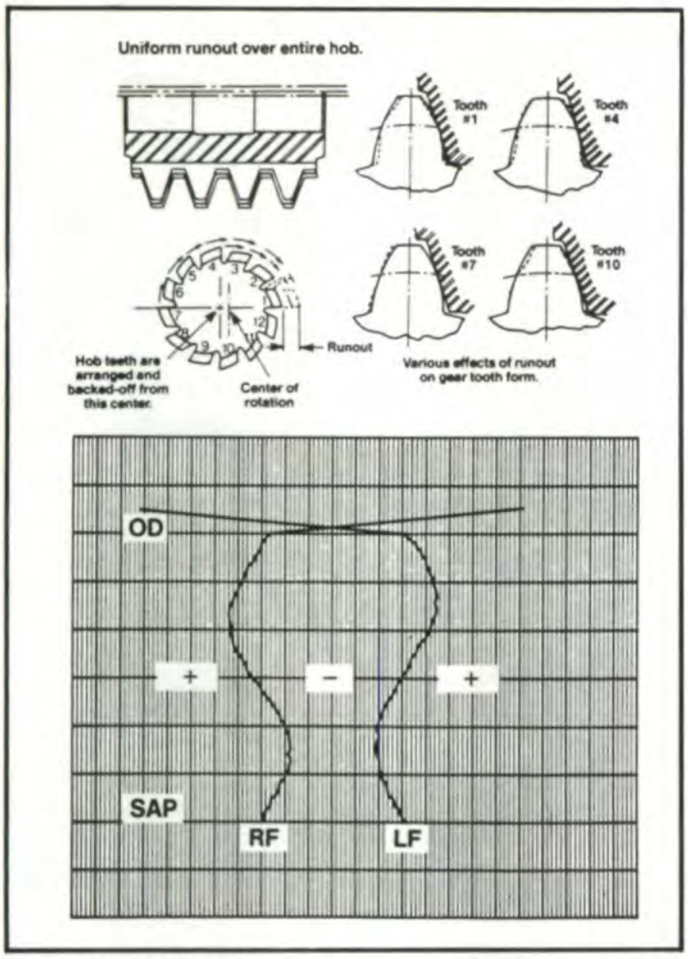


Fig. 10 – Effect of uniform hob runout on hobbed profile.

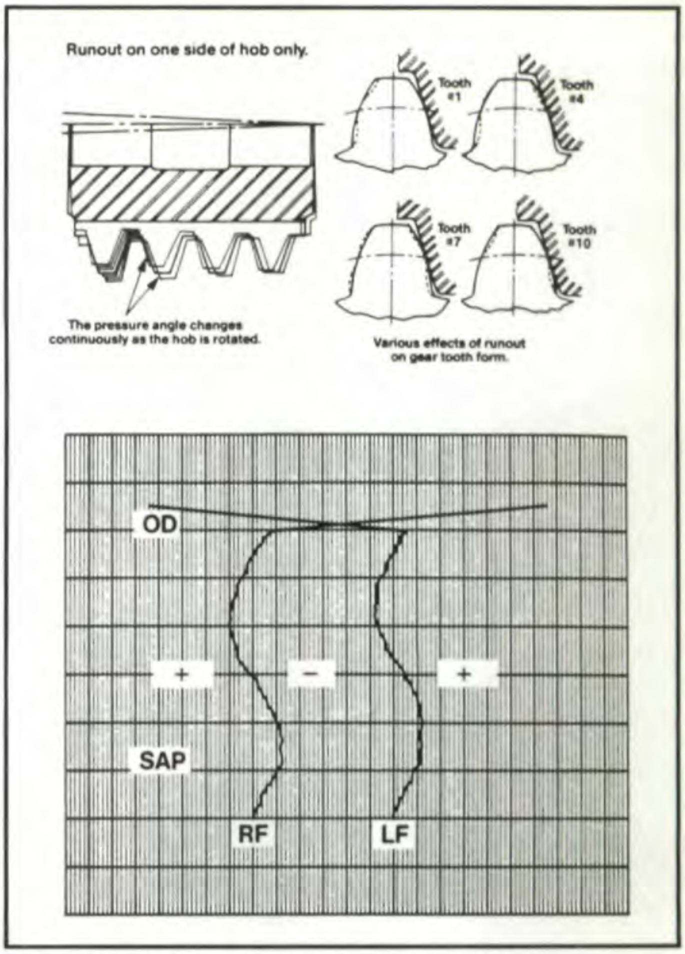


Fig. 12 – Effect of runout on one side of the hob only.

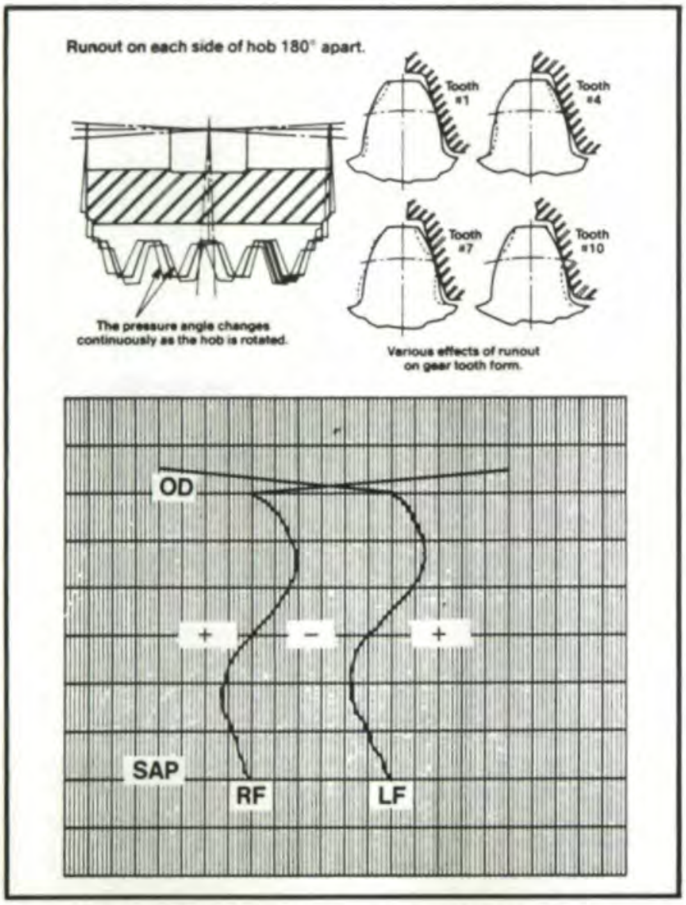


Fig. 11 – Effect of runout on each side of hob, 180° apart.

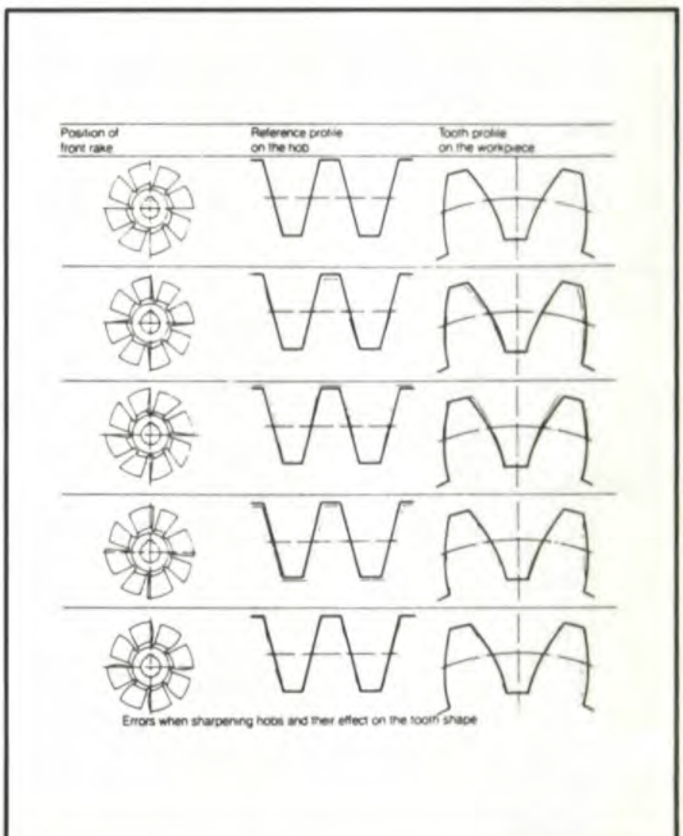


Fig. 13 – Effect of hob resharpening errors on the hobbed tooth profile relative to the basic rack profile of the hob.

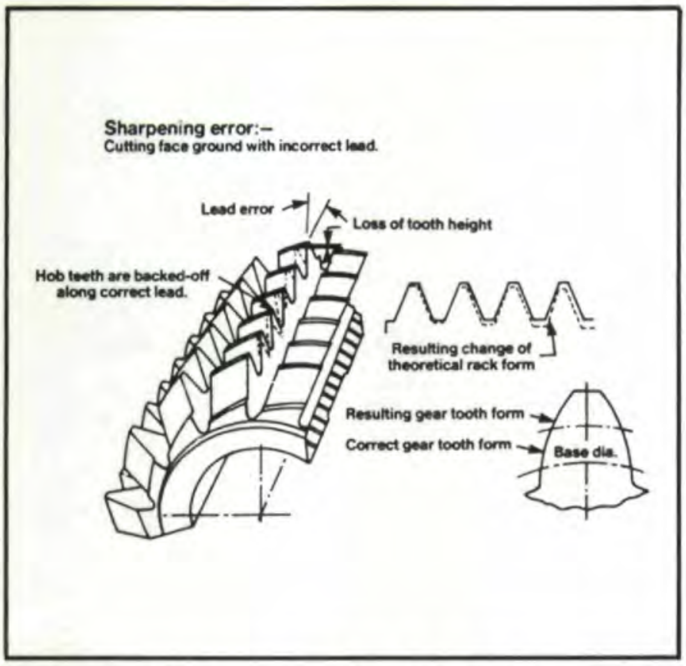


Fig. 14 – Effect of hob flute lead error. Since the hob is a reducing cylinder, incorrect flute lead resharpening destroys the integrity of the hob cylinder end to end, typically causing changes in workpiece size as the hob is shifted across its length.

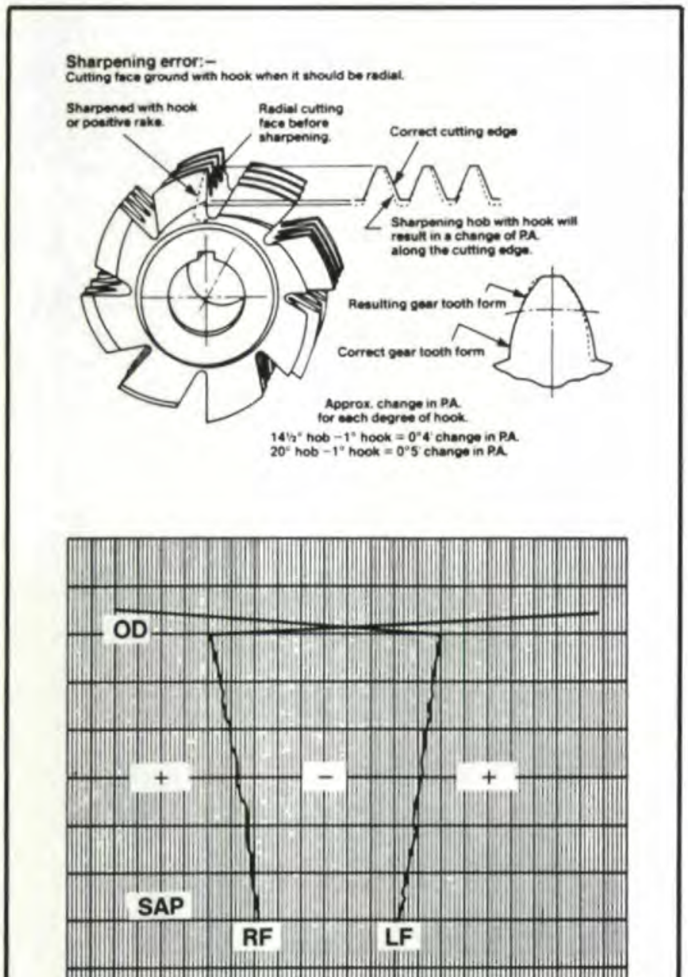


Fig. 15 – Effect of negative rake resharpening error on profile.

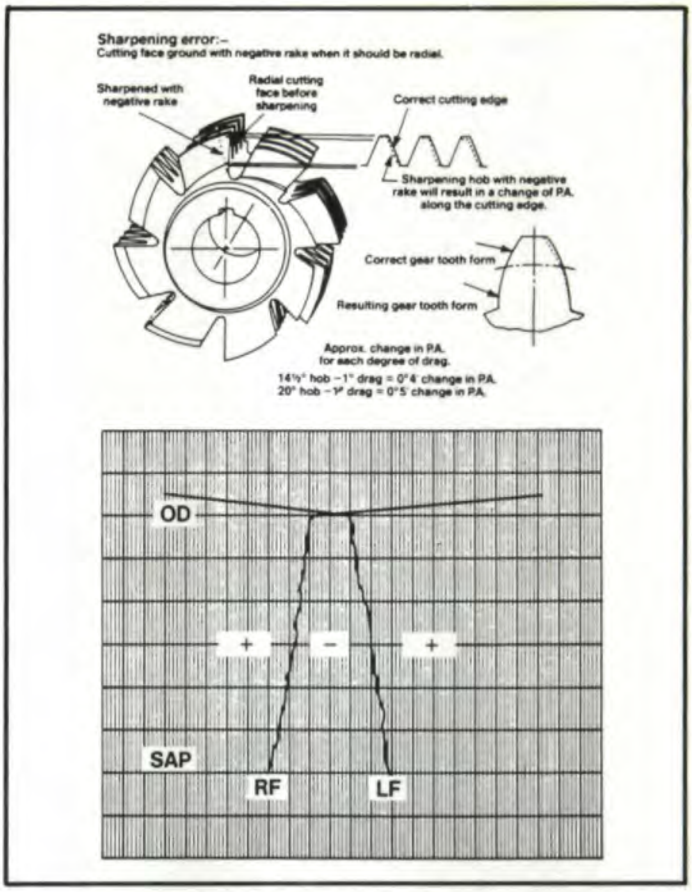


Fig. 16 – Effect of positive rake resharpening error on profile.

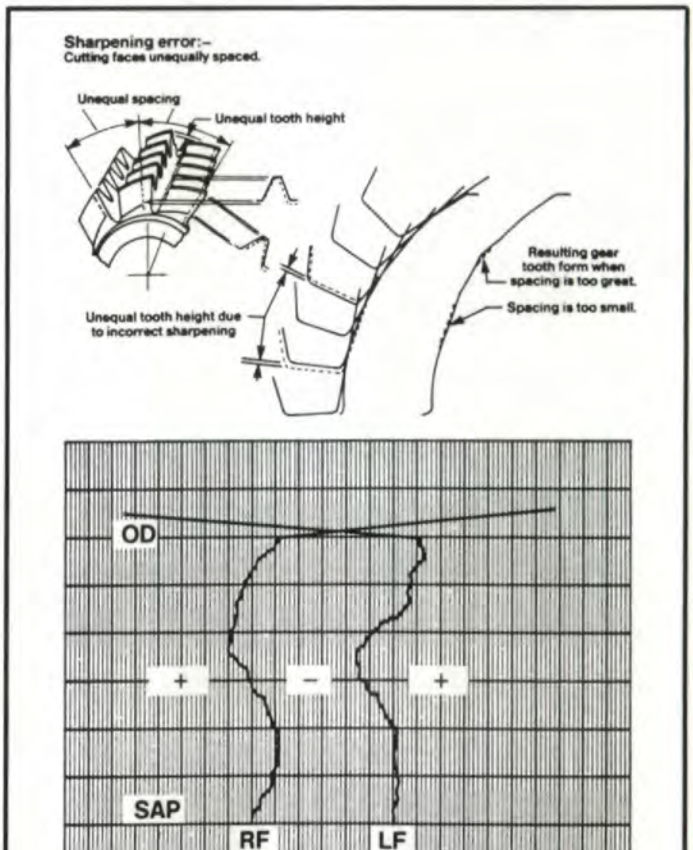


Fig. 17 – Effect of accumulated flute spacing error on profile.

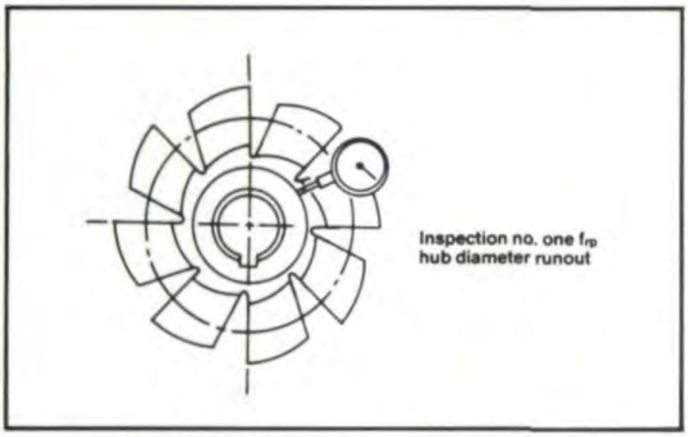


Fig. 18 – Hob diameter runout inspection check.

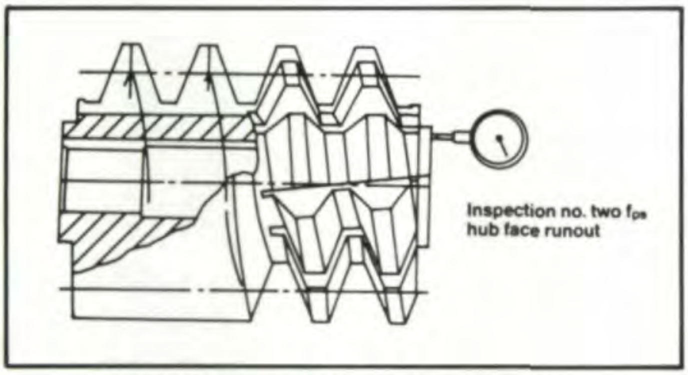


Fig. 19 – Hob clamping face runout inspection check.

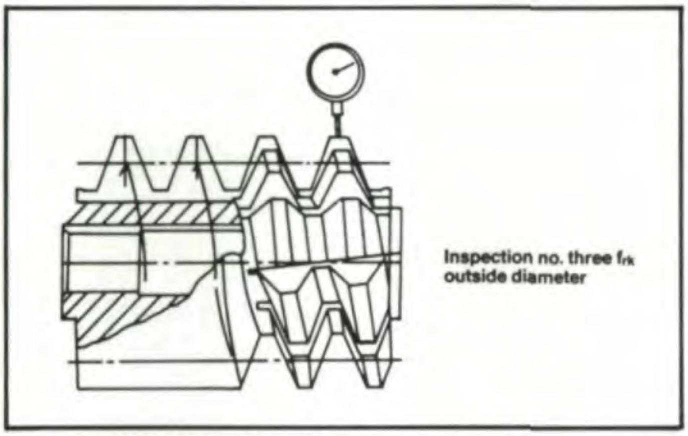


Fig. 20 – Hob outside diameter inspection check.

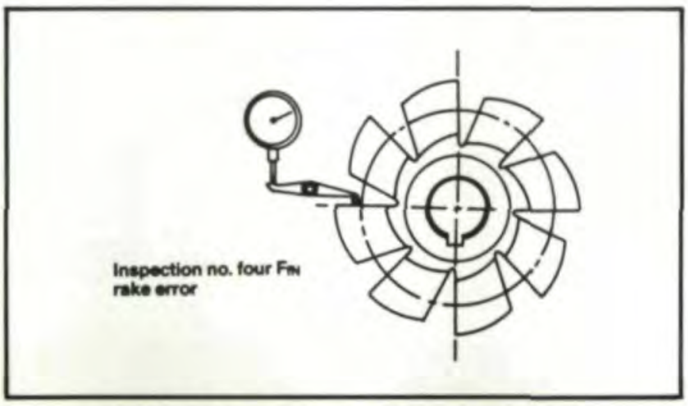
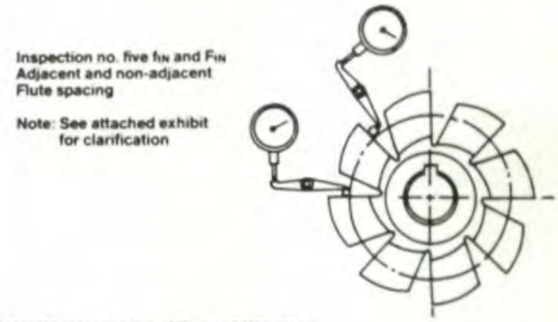


Fig. 21 – Hob rake to cutting depth inspection check.



Example of inspection of flute spacing error:

FLUTE NO. (N)	MEASUREMENT	CORRECTION	ftN	FtN
1	+.0004	−.0002	+.0002	+.0002
2	0	−.0002	−.0002	0
3	+.0004	−.0002	+.0002	+.0002
4	+.0001	−.0002	−.0001	+.0001
5	+.0002	−.0002	0	+.0001
6	−.0001	−.0002	−.0003	−.0002
7	+.0006	−.0002	+.0004	+.0002
8	0	−.0002	−.0002	0

Correction value = sum of measured errors divided by total number of flutes.

ftN = measured value minus correction value

fuN = maximum deviation between any two *adjacent* flutes. From above: fuN = deviation of −.0003 and +.0004
fuN = .0007

Ft(N+1) = Ftn = ft(N=1)
Note: This will give the particular value of FtN for one flute.
The decisive value will equal the maximum deviation between any two values of FtN. From above: FtN = deviation of +.0002 and −.0002
FtN = .0004

Fig. 22 – Hob adjacent and non-adjacent flute spacing check.

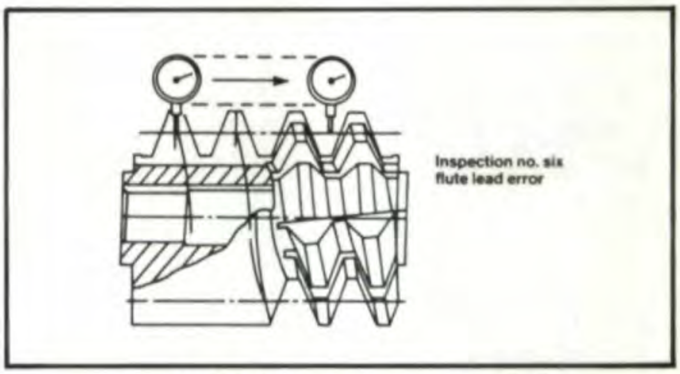


Fig. 23 – Hob flute lead error inspection check.

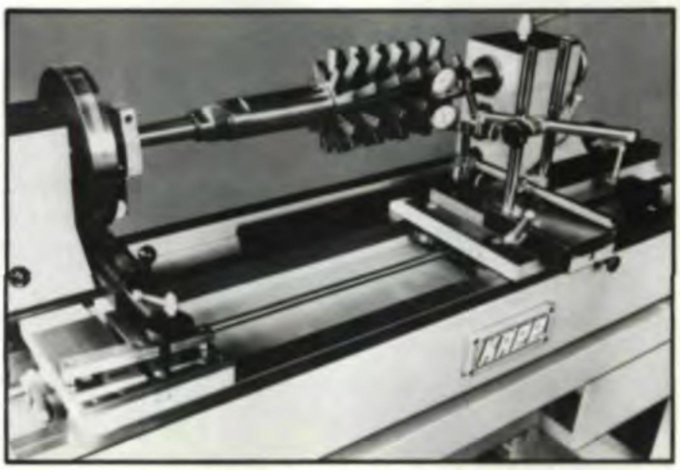


Fig. 24 – Hob checking unit.

2. The hob cutting faces sharpened with negative rake (Fig. 15)

3. The hob cutting faces sharpened with positive rake (Fig. 16)

4. The hob cutting faces sharpened with unequal spacing (Fig. 17).

Fig. 14 shows the effect of sharpening the reducing cylinder of a straight fluted hob with a lead error. This occurs often in older hob sharpeners with misaligned centers. Because the hob is a reducing cylinder, sharpening more off one end of the hob than the other results in a tapered hob. As the hob is shifted across its usable life in the hobbing machine, a change in the size of the workpieces will be evident. Often this error is assigned to the hobbing machine and valuable production hobs are wasted while maintenance crews attempt to find the source of the error.

The same error can exist in helically fluted hobs. Wear of the sine bar or misaligned centers contribute to this off lead problem.

Figs. 15 and 16 show two common resharpening errors on radial rake designed hobs – positive and negative rake. The effect of positive or negative rake sharpening on a radial rake designed hob is a change in the pressure angle of the basic rack form of the hob. This produces either a lesser or greater pressure angle on the gear tooth, which can result in excessive gear wear, gear noise and shock loading.

Some hobs are deliberately designed with hook (positive rake) or with negative rake and must be sharpened accordingly to prevent the introduction of pressure angle errors.

Fig. 17 illustrates the condition of unequally sharpened hob flutes, resulting in unequal spacing of the cutting edge positions relative to the thread helix. Due to the flank or cam relief on the hobs, unequally spaced flutes will cut either high or low from the nominal enveloping helix, producing a "wandering" profile.

Usually worn index plates or worn pawls are the source of this problem. Excessive stock removal during the resharpening can crowd the grinding wheel, also causing unequal flute spacing.

Inspecting the Resharpened Hob

Figs. 18 through 23 illustrate the six basic checks which can be performed to insure that the hob resharpening conforms to the tolerance level of the hob purchased class accuracy. These simple checks can be performed on bench centers or with a hob checking unit such as shown in Fig. 24.

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THE INTERRELATIONSHIP OF TOOTH . . .

(continued from page 34)

TERMS

The following is a list of terms and definitions as used in Equation (1) through (11).

- Ainv — Perform the arcinvolute function.
- Acos — Perform the arccosine function.
- Asin — Perform the arcsine function.
- Sin — Perform the sine function.
- Cos — Perform the cosine function.
- Inv — Perform the involute function.
- BD — Diameter of the gear base circle.
- BTN — Normal tooth thickness at the gear base circle.
- CP — Contact point of the ball and the tooth side.
- D — Exact ball diameter to contact the tooth at the midpoint between the outside circle and the form point.
- DBALL — Dimension over the balls.
- DD — Diameter of the circle to any designated point on the involute surface.
- DS — Diameter of the standard ball.
- FD — Diameter of the gear form circle.
- HADD — Helix angle at diameter DD.
- K — Designator to determine external or internal – K is + 1 for external gears; K is – 1 for internal gears.
- OD — Diameter of the gear outside circle.
- PACB — Transverse pressure angle at the center of the ball.
- PACP — Transverse pressure angle at the ball contact point.
- RCB — Radius to the center of the ball.
- RCP — Radius to the contact point.
- ROB — Radius over the ball.
- RUB — Radius under the ball.
- TTDD — Normal tooth thicknes at diameter DD.
- Z — Number of teeth on the gear.

$$\mathrm{PACB} = K * \left[\frac{\pi}{Z} - \left[\frac{(\ \mathrm{BTN}}{\mathrm{Cos}\ (\mathrm{BHA}) * \mathrm{BD}} \right] + \right. \qquad (3)$$

If Z is even:

$$\mathrm{DBALL} = 2 * \mathrm{RCB} + K * \mathrm{DS} \qquad (11)$$

Measurement Over 1 Wire

$$M_1 = R_W + (d_w/2) \qquad (12)$$

Chordal Addendum Specification

$$a_c = a + (T_{Mc}^2 \cos^2\Psi) \ / \ (4\ d_M) \qquad (13)$$

Chordal Tooth Thickness Specification

$$t_c = T_{Mc} - (T_{Mc}^3 \cos^4) \ / \ (6\ d_M^2) \qquad (14)$$

Measurement Over 2 Wires
(Even Number of Teeth)

$$M_2 = D_w + (d_w/2) \qquad (15)$$

(Odd Number of Teeth)

$$M_2 = 2\ R_w[\cos(90/N)] + d_W/2 \qquad (16)$$

Span Measurement Specification

$$M_s = D \cos\Phi[\pi/(2N) + \mathrm{inv}\Phi] + (n - 1)(\pi/P_d \cos\Phi) - (T_{Ms}) \cos\Phi \qquad (17)$$

Tooth Thickness and Space Width

$$\pi/P_d = t + s \qquad (18)$$

Change in Arc Tooth Thickness vs. Change in Center Distance

$$\Delta t = 2 \tan \Phi \Delta C \qquad (19)$$

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(continued from page 10)

TABLE I

Quantity	Example of Usage	US Customary Units	SI Units	Typical Values
Acceler.: longitudinal	vehicle, gravity, valve train	ft/sec^2, g, in/sec^2	m/s^2	dragster: 20m/s^2, earth gravity: 9.8 m/s^2, cam follower: 1000 m/s^2
angular	crankshaft, governor	deg/sec^2, rad/sec^2	rad/s^2	dragster wheel: 40 rad/s^2
Angle: figure descrip.	geometric pictures twist, dynamics	deg, min, sec	rad[deg]	one radian is 57.3 degrees
engineering	of rotation	rad, degree	rad	6.28 rad is one revolution
Area	cross-section, land measure	in^2, ft^2, acre	m^2	office space per person: 10 m^2, discharge port: 4 mm^2, nozzle orifice: 0.12 mm^2
Coeff. of therm. exp.	shrink fit, volumetric growth	1/°F	1/K	aluminum alloy: 22 μm/m•K, steel: 11 μm/m•K, kerosene: 1 dm^3/m^3•K, water: 0.18 dm^3/m^3-K, water: 0.18 dm^3/m^3-K
Density	mass density, specific weight	lb/ft^3, lb/gal	kg/m^3	diesel fuel: 870 kg/m^3 water: 1000kg/m^3, aluminum alloy: 2700 kg/m^3, steel: 7850 kg/m^3
Dimension	mech. engineering design	in	mm	rotor dia: 23.410, rotor dia. tolerance: ±0.004, car length: 5730
Energy: general	work, heat, electricity	ft-lb, Btu, kW-h	J	60 W bulb: 5 MJ in one day, diet soft drink: 4 kJ, 1 MJ of electricity costs 2¢
specific	energy content in fuel	Btu/lb	J/kg	diesel fuel: 42 MJ/kg, methanol: 20 MJ/kg
Flow	See mass flow or volume flow.			
Force	weight, thrust, drag	oz, lb, ton	N	big man exerts 1000 N force of gravity on earth
Frequency: angular	See angular velocity.			
cycle	vibration pulsing	c/min, c/sec	Hz	athlete's heart at rest: 1 Hz, middle "C": 261.6 Hz, engine vibrates at 300 Hz
rotational	rpm, rps	r/min, r/sec	1/s	induction motor: 30 1/s, small genset: 60 1/s, truck turbocharger: 2000 1/s
Fuel cons.: transport	highway economy	mile/gal	km/dm^3	car: 10 km/dm^3, truck: 2 km/dm^3 (dm^3 – liter)
Specific	See specific fuel consumption			
Heat	See energy.			
Length	distance	mil, in, ft, yd	m	dime thick.: 1mm, man height: 170 cm, "ideal" form: 90-60-90, jets fly at 10 km alt.
Load	See mass or force			102 kg mass exerts 165 N force of gravity on moon, 1 kN on earth, 275 kN on sun
Mass: object	weight, load, inertia	oz, lb, ton, slug	kg	1 dm^3 contains 1 kg of water, average man has mass of 70 kg, truck has mass of 30 Mg
flow	flow rate, fuel consumption	lb/min	kg/s	car engine fuel flow: 2 g/s, rocket engine: 100 kg/s
Modulus of: elasticity	longit.(E), shear(G), volumet.(K)	lb/in^2	GPa	steel: E–200 GPa, G – 80 GPa; diesel fuel: K – 1.5 GPa
section	bending and twist calcul.(W)	in^3	m^3	rod of 10 mm diameter: W – 100 mm^3 in bending, 200 mm^3 in torsion
Moment of: area, 1st	centroid of shape	in^3	m^3	
area, 2nd	bending and twist calculation	in^4	m^4	rod of 10 mm diameter: 2nd moment – 500 mm^4 in bending, 1000 mm^4 in torsion
force	torque, bending couple	ft-lb, in-lb	N•m	1dm^3 engine torque: 100 N•m, discharge fitting installation torque: 80 N•m
inertia	dynamics of rot., Flywheel Effect	lb-in-sec^2, lb-ft^2	kg•m^2	flywheel of a car engine: 0.1 kg•m^2, truck engine: 2 kg•m^2
Power	heating, engine	Btu/min, hp	W	av. man: 100 W cont., small kerosene heater: 2000 W, House furnace: 40 kW, car: 60 kW
Pressure	stress, vacuum, injection	psi, inHg, inH$_2$O	Pa	filter Δp: 70 Pa, earth atm.: 100 kPa, tire: 200 kPa, BMEP: 1.2 MPa, diesel injec.: 50 MPa
Spec Energy Consump.	brake spec. energy consumption	Btu/hp-h	non-dim.	engine 33% efficient has BSEC of 3
Spec. Fuel Consumption	brake spec. fuel consumption	lb/hp-h	g/MJ	diesel engine: 80 g/MJ, gasoline engine: 100 g/MJ
Specific Gravity	ratio of grav. forces or densities	dimensionless	not appl.	relative density of water is 1, of steel 7.85, if water is the reference substance
Speed	See frequency or velocity			
Spring rate: longitud.	spring force per change in length	lb/in	N/m	valve spring: 10 kN/m, car suspension: 30 kN/m, railroad car spring: 4MN/m
torsional	spring torque per change in angle	deg/100 ft-lb	N•m/rad	torsionally soft drive shaft: 600 N•m/rad. torsionally stiff drive shaft: 300 kN•m/rad
Stress	See pressure.			strength of steel-tensile: 900 MPa, bending: 200 MPa
Temperature	fever, melting point	°F, °R	K [°C]	human body: 37°C, ice melts at 273 K (0°C), steel is hot forged at 1300 K
Time: daily schedule	clock, events of daily life	h, min	[h, min]	60 min in 1h, 24 h in 1 day
engineering	flow measurement, elapsed time	h, min, sec	s	1 ks is 17 min, 100 ks is approx. 1 day, 1 revolution takes 20 ms at 50 rev./s
Torque	See moment of force.			
Vacuum	See pressure.			100% vacuum: 0 Pa absolute pressure, 50% vacuum: 50 kPa pressure differential
Velocity: longitudinal	vehicle, boat, fuel plume, cam	mile/h, knot	m/s	athlete: 10 m/s, speed lim.: 25 m/s (88 km/h), sound in air: 333 m/s, in water: 1444 m/s
peripheral	grinding wheel, fan tip	ft/min	m/s	grinding speed: 50 m/s
angular	2π X freq. of rotation	deg/sec, deg/min	rad/s	100 rev./s is 628 rad/s
Viscosity: kinematic	fuel property, lubrication	centistokes	mm^2/s	water: 1mm^2/s, diesel fuel: 5 mm^2/s at 20°C, lub. oil: 10 mm^2/s at 100°C
Volume: content	tank, milk, cylinder	qt, gal, in^3, ft^3	m^3	1 quart is approximately 1 dm^3, beverage can is 1/3 of dm^3
flow	fuel flow, air flow rate	ft^3/min, yd^3/min	m^3/s	air usage of a marathon runner: 1 dm^3/s, car engine: 100 dm^3/s
Weight: Object	See mass or force.			
specific	See density or specific gravity.			
Work	See energy.			

(continued on page 50)

SI UNITS – MEASUREMENTS . . .

(continued from page 48)

Acceleration: angular

1 degree/sec^2	is	0.0175	rad/s^2

Angle

1 degree	is	0.0175	rad
1 min	is	0.291	mrad

Area

1 acre	is	4047	m^2
1 ft^2	is	0.093	m^2
1 hectare	is	0.01	km^2
1 in^2	is	645	mm^2
1 yard^2	is	0.836	m^2

Coefficient of thermal expansion

1/°F	is		1.8/K

Density (i.e., specific mass. For relative density see specific gravity.)

1 lb_m/ft^3	is	16.02	kg/m^3
1 lb_m/in^3	is	27.68	Mg/m^3
1 lb_m/gal (US)	is	119.8	kg/m^3
1 kg/dm^3 or g/cm^3	is	1000.	kg/m^3

Note: The $_m$ and $_f$ subscripts are intended to distinguish between the two meanings of oz., lb., ton and the like. When these units refer to mass, the subscript m is used; when to force, the subscript f is used. In SI, the unit of mass is formed from the word gram, and the unit of force from the word newton. Whereas lb_m and lb_f are numerically the same for a given object on Earth, newton and gram differ. The lb_m/lb_f distinction can be ignored in the common measurements of daily life, as the distinction makes no difference there. In some branches of engineering, however, and in converting to SI units, understandng the distinction is vital for avoiding errors. Some specific weight tables list the values derived from force units, not mass units. The potential error resulting from the use of the above conversion factors is 0 to 0.5%.

Dimension (is always in mm in Mech. Eng.)

1 ft	is	305
1 in	is	25.4
1 yard	is	914

Energy: general (including work)

1 Btu	is	1.055	kJ
1 cal (thermochemical)	is	4.19	J
1 Cal (usage in nutr.)	is	4.19	kJ
1 ft-lb_f	is	1.36	J
1 kg-m	is	9.81	J
1 kW-hr	is	3.6	MJ

Note: kg* = kg force, sometimes written kp.

Energy: specific

1 Btu/lb_m	is	2.33	kJ/kg
1 cal/g	is	4.19	kJ/kg

Flow: mass

1 lb_m/min	is	7.56	g/s
1 lb_m/sec	is	0.454	kg/s

Flow: volumetric

1 ft^3/min	is	0.472	dm^3/s
1 gal (US)/h	is	1.05	cm^3/s
1 gal (US)/min	is	63	cm^3/s

Force (incl. load, weight, etc., where they pertain to force)

1 dyne	is	0.01	mN
1 kg**	is	9.81	N
1 oz_f	is	0.278	N
1 lb_f	is	4.45	N
1 ton (short) force	is	8.90	kN
1 ton (metric) force	is	9.81	kN

Frequency: cycle

1 cpm	is	1/60	Hz

Frequency: rotational

1 rpm	is	1/60	1/s

Note: 1/s = s^{-1}. Both symbols represent the SI unit of rotational frequency, popularly known as speed, speed of rotation, revolutions per second, rps, and the like.

Fuel consumption: transportation (incl. economy)

1 lb_m/h	is	0.126	g/s
x liter/100 km	is	100/x	km/dm^3
1 mile/gal (US)	is	0.43	km/dm^3

Note: 235.2/mpg = liter per 100 km
235.2/liter per 100 km = mpg

Fuel consumption: specific

1 lb_m/hp (US)-h	is	169	μg/J
1 g/kW-h	is	0.278	μg/J

Length (See also dimension.)

1 ft	is	0.305	m
1 in	is	25.4	mm
1 mile (nautical)	is	1.85	km
1 mile (statute)	is	1.61	km
1 yard	is	0.91	m

Mass (incl. load, weight, etc., where they pertain to mass)

1 carat	is	0.2	g
1 oz_m (avoirdupois)	is	28.35	g
1 oz_m (troy)	is	31.10	g
1 lb_m	is	0.454	kg
1 slug	is	14.6	kg
1 ton (short) mass	is	0.907	Mg

Modulus of: elasticity

1 lb_f/in^2	is	6.89	kPa

Modulus of: section

1 in^3	is	16.4	cm^3

Moment of: area, 1st

1 in^3	is	16.4	cm^3

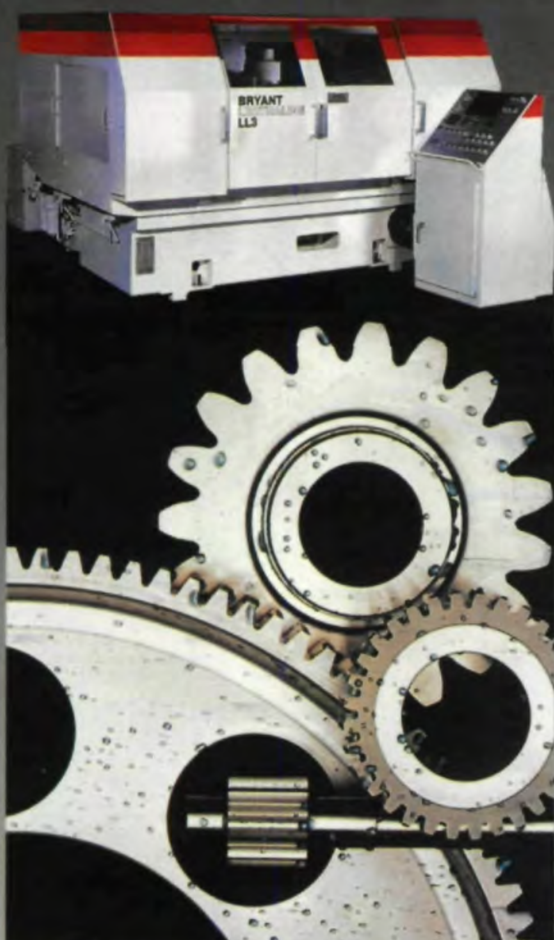
Moment of: area, 2nd

1 in^4	is	41.6	cm^4

(continued on page 52)

Tight grinding tolerances? Try a Bryant Grinder.

Overton Gear did — and returned for seconds!



The search ends! Quality and precision solutions found with Bryant Lectraline® CNC grinders.

Overton Gear and Tool Corporation of Addison, Illinois, a prime manufacturer of quality gears for commercial, industrial, and military applications, was prepared to purchase new precision grinding equipment. According to Overton President Tony Waitekus, "We were looking elsewhere when we heard of the innovatively designed Bryant Lectraline Grinder. We liked what we saw and were convinced that to compete and meet tighter tolerance requirements we had to go Bryant."

Since its installation, the Bryant Lectraline LL3-55 CNC grinder has been performing a multitude of precision ID, OD, and face grinding operations on a wide variety of gear types and sizes. According to A. Flores, Overton's Quality Control Manager, the Lectraline, in automatic cycle, maintains repeatability tolerances of 0.0002″ total size with 0.000040″ total straightness on a 2″ diameter, 2″ long, bore — without in-process gaging!

Mr. Waitekus states, "We could see the increased productivity and multiple machine capabilities the Lectraline had to offer." Mr. Flores adds, "With the Bryant we have experienced less scrap, less rework, a higher quality product, and production done in a fraction of the original time." Waitekus concludes, "The Bryant Lectraline more than met our expectations. The service and assistance we have received has been outstanding. Having the fullest confidence in the machine's performance we ordered a second."

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CIRCLE A-7 ON READER REPLY CARD

SI UNITS – MEASUREMENTS . . .

(continued from page 50)

Moment of: force (incl. torque and couple)

1 ft-lb_f	is	1.36	N•m
1 in-lb_f	is	0.113	N•m
1 in-oz_f	is	7.06	mN•m
1 kg-m	is	9.81	N•m

Note: The unit was also written as N•m/rad.

Moment of: inertia (incl. Flywheel Effect)

1 lb_f-in-sec^2	is	0.113	kg•m^2
1 lb_f-ft-sec^2	is	1.36	kg•m^2
1 kg-cm-sec^2	is	0.0981	kg•m^2
1 GD^2 (kg-m^2)	is	0.25	kg•m^2
1 GD^2 (kg-m-sec^2)	is	0.0255	kg•m^2
1 WD^2 (lb-ft^2)	is	0.168	kg•m^2
1 WD^2 (lb-in^2)	is	0.00117	kg•m^2
1 WR^2 (lb-ft^2)	is	0.0421	kg•m^2
1 WR^2 (lb-in^2)	is	0.00029	kg•m^2

Note: The unit was also written as kg•m^2/rad^2. GD, WD and WR represent the so-called flywheel effect. Flywheel effect has sometmes been used in place of the polar mass moment of inertia. The different symbols reflect the fact that flywheel effect may not have exactly the same meaning from country to country, or even within a country. To correctly convert them is really tricky.

Momentum

1 lb_m-ft/sec	is	0.138	kg•m/s
1 lb_m-in/sec	is	0.0115	kg•m/s

Power (incl. heat rate)

1 Btu/h	is	0.293	W
1 ft-lb_f/min	is	0.0226	W
1 hp (US, mech. eng.)	is	0.746	kW
1 hp (metric)	is	0.735	kW
1 ton (refrigeration)	is	3.52	kW

Pressure (incl. stress)

1 atm (internat.)	is	101.3	kPa
1 bar	is	100.0	kPa
1 kg/cm^2	is	98.1	kPa
1 $lb_f ft^2$	is	47.9	Pa
1 lb_f/in^2	is	6.89	kPa
1 inHg (60°F)	is	3.38	kPa
1 inH_2O (60°F)	is	0.249	kPa
1 mmHg (16°C)	is	133	Pa
1 mmH_2O (16°C)	is	9.80	Pa

Specific force of gravity (For specific mass see density.)

1 lb_f/ft^3	is	157	N/m^3
1 lb_f/in^3	is	271	kN/m^3
1 kg/dm^3	is	9.81	kN/m^3)

Specific gravity (meaning relative density)

Conversion of this nondimensional quantity to density in kg/m^3 or to specific force of gravity in kN/m^3, is a matter of multiplication by 1000 or by 9.81, respectively, when water is the reference mass.

Spring rate: longitudinal

1 lb_f/ft	is	14.6	N/m
1 lb_f/in	is	175	N/m

Spring rate: torsional

x deg/100 ft-lb_f	is	7.8/x	kN•m/rad
1 kg m/rad	is	9.81	N•m/rad
1 lb_f-ft/rad	is	1.36	N•m/rad
1 lb_f-in/rad	is	0.113	N•m/rad

Note: The unit was also written as N•m/rad^2.

Temperature

1°F=1°R (increment)	is	0.556	K
1°F=461°R (scale)	is	−16.8°C=256.4K	

$t_{°C}=(t_F-32)/1.8$

$t_K=(t_F+460)/1.8$

$t_K=5t_R/9$

Velocity (incl. angular)

1 deg/sec	is	0.0175	rad/s
1 ft/min	is	0.0051	m/s
1 ft/sec	is	0.305	m/s
1 in/sec	is	0.0254	m/s
1 km/h	is	0.278	m/s
1 knot (international)	is	0.515	m/s
1 mile (statute)/h	is	0.447	m/s

Viscosity

1 centipoise	is	1	mPa•s
1 centistokes	is	1	mm^2/s

Volume

1 barrel (US liq. exec. oil)	is	0.12	m^3
1 barrel (oil)	is	0.16	m^3
1 ft^3	is	0.028	m^3
1 gal (US)	is	3.79	dm^3
1 in^3	is	16.4	cm^3
1 oz (US, liquid)	is	29.6	cm^3
1 quart (dry)	is	1.101	dm^3
1 quart (US, liquid)	is	0.946	dm^3
1 $yard^3$	is	0.765	m^3

Weight – See mass or force. For specific weight see density, specific force of gravity, or specific gravity.

Kg* ≐ kg force, sometimes written also as kp.

Use of this Table:

1) Converting to SI units: Multiply the known value by the equivalent.
   Example: Flow of 15 lbm/min. How much is it in g/s?

   Since 1 lbm/min is 7.56 g/s,
   15 lbm/min is 15 X 7.56 = 113.4 g/s
2) Converting to U.S. units: Divide the known value by the equivalent.

(continued on page 56)

ULTRA-CASE™

High speed induction gear hardening cuts costs and eliminates carburizing

ULTRA-CASE:

- **Eliminates carburizing.**
- **Is easily integrated into your gear manufacturing cell.**
- **Eliminates dimensional distortion problems.**
- **Reduces overall manufacturing costs.**
- **Processes a wide variety of gears.**
- **Utilizes proven high performance technology.**

The new IEH 50 KHz solid state ULTRA-CASE System offers you surface hardened gears at production line speeds. ULTRA-CASE enables you to harden gears twice as fast as any other comparable induction system with less equipment and at lower cost.

Gear Expo '87

See us in Booth #324.

AN AJAX MAGNETHERMIC GROUP COMPANY

Unlike currently available systems, ULTRA-CASE requires no pre-heat cycle which often affects core hardness. ULTRA-CASE 50 KHz also eliminates the use of outmoded RF sets.

Let Industrial Electric Heating put ULTRA-CASE to work for you. Send us a sample part or let us make a production run. We can even handle some of your heat treating jobs in our plant. For more information, write to:

Industrial Electric Heating, 3949 Lyman Drive, Hilliard, Ohio 43026; Phone (614) 876-0456, or P.O. Box 991, Warren, Ohio 44482; or call (216) 372-8574, telex 98 2482, fax (216) 372-8644.

IEH

Industrial Electric Heating

CIRCLE A-21 ON READER REPLY CARD

CLASSIFIED

Rates: Classified Display—$110 per inch (minimum 3") Type will be set to advertiser's layout or *Gear Technology* will set type at no extra charge. **Word Count:** 35 characters per line, 7 lines per inch.

Payment: Full payment must accompany classified ads. Mail copy to *Gear Technology*, P.O. Box 1426, Elk Grove Village, IL 60007. **Agency Commission:** No agency commission on classifieds.

Closing date: Ads must be received by the 25th of the month, two months prior to publication. **Acceptance:** Publisher reserves the right to accept or reject classified advertisements at his discretion.

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- Custom gear design including non-standard pressure angles for more strength.
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The COSTIMATOR® computer aided cost estimating system insures speed and consistency in the difficult task of estimating the costs of all types of gears.

Used by small shops and Fortune 500 companies throughout the country.

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(413) 733-1972

CIRCLE A-39 ON READER REPLY CARD

GEAR DESIGN & ANALYSIS PROGRAMS FOR IBM COMPATIBLE

Worm, Spur, Helical will be demonstrated at BOOTH 805

Buckingham Gear Books also available

For more info call:

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591 Parker Hill
Springfield, VT 05156
Tel. 802-885-5052

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HOBBER REBUILDING SPECIALISTS

Having trouble meeting today's demand quality control tolerances?
Let our factory trained and experienced staff return your machine to optimum operating condition.
We specialize in repairing, rebuilding and modernizing all makes of hobbers.

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- Gould & Eberhardt
- Barber Colman

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522 Cottage Grove Road
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CIRCLE A-41 ON READER REPLY CARD

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GEAR TOOTH GRINDING & HONING ONLY

Production Quantities
3/4" P.D. to 27.5" P.D.;
3.5 D.P. and 11" Face

Gear Tooth Finishing is our Only Business

We have no turning, hobbing or shaping capability

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CIRCLE A-42 ON READER REPLY CARD

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SERVICE:

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PRECISE

PRECISE INSPECTION
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Roseville, Michigan 48066
313-445-6959

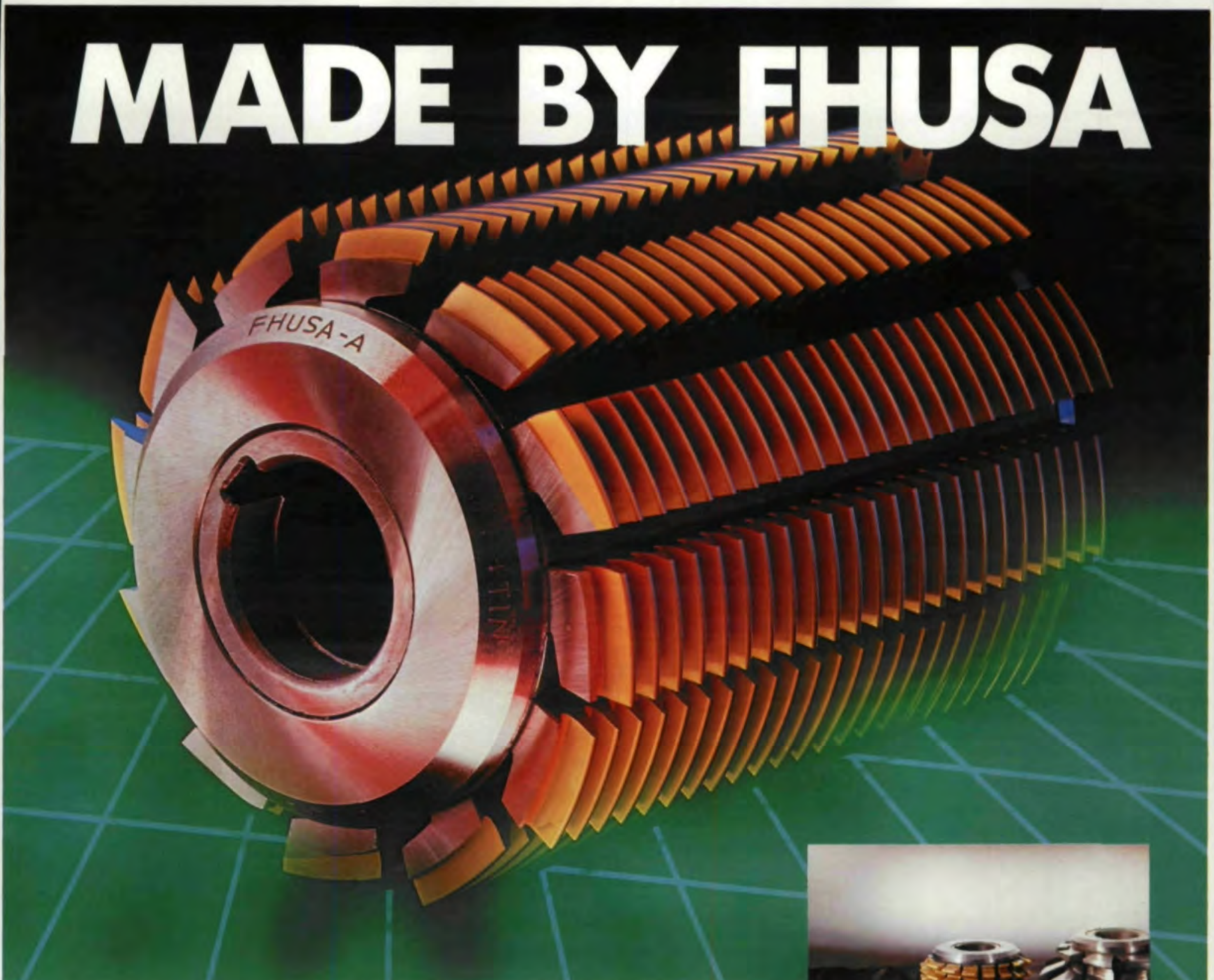
CIRCLE A-44 ON READER REPLY CARD

HELP WANTED

WANTED: Gear set up man for close tolerance work on late model gear machinery. Requirements: shaping, hobbing and inspection experience. Midwestern location; good pay and fringes. Send resume to: Box AT, Gear Technology, P.O. Box 1426, Elk Grove, IL 60007

Bring in new customers for your business by advertising in GEAR TECHNOLOGY, The Journal of Gear Manufacturing.
Call (312) 437-6604

There's still time . . . closing date for a classified ad in the Nov./Dec. issue is **Sept. 25th**



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FHUSA is ready to design the hob you need, however hard it may be the work to be developed.

MADE BY FHUSA, THE CHALLENGE OF AN ORIGIN THAT GUARANTEES ITS EFFICIENCY.

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CIRCLE A-25 ON READER REPLY CARD

SI UNITS – MEASUREMENTS . . .

(continued from page 52)

Example: Flow of 15 g/s. How much is it in lbm/min?

Since 1 lbm/min is 7.56 g/s,
15 g/s is $15 \div 7.56 = 1.98$ lbm/min

3) Converting when the x is in the formula: Place the known value for the x.

Example: Fuel consumption of 20 liter/100 km. How much is it in km/dm^3?

Since x liter/100 km is 100/x km/dm^3
20 liter/100 km is $100/20 = 5$ km/dm^3

Rounding Off Converted Numbers

One frequently encounters complicated, beyond-the-decimal point "accurate" conversions. Following are typical cases indicating practical considerations one must apply when converting.

Torque – Torque values, when they refer to fasteners, are inherently inaccurate. They are usually rounded off to a simple number within 20% of the nominal value. Therefore, the data in N•m will seldom need to be expressed in numbers more precise than rounded off to the nearest 5 N•m or a unit value.

Example: 15-18 ft-lb (20-25 N•m)
not 20.3–24.4 N•m
50–60 in-lb (6-7 N•m)
not 5.7–6.8 N•m

For very small values, no greater precision than 0.5 N is needed.

Example: 10–1 in-lb (1–1.5 N•m)
not 1.13–1.7 N•m

Dimension – A statement that something is 20 inches long may mean a length of between 19 and 21 inches if it is a piece of firewood or 20.000 inches if it is a part of a machine. Similarly, 2.625 inches may mean a length of between 2½ to 2¾ inches or exactly 2.625 inches. The knowledge of application and circumstances involved in making the measurement is needed to determine conversion precision. The 20 inches may correctly be converted to both ½ m and 508.00 mm, the choice depending on the origin and purpose of the information.

TECHNICAL CALENDAR

SEPTEMBER 16-18
GEAR NOISE SEMINAR—Ohio State University.

This course will cover general noise measurements and analysis, causes of gear noise, gear noise reduction techniques, dynamic modeling, gear noise signal analysis and modal analysis of gear boxes. For further information, contact Mr. Richard D. Frasher, Director of Continuing Education, College of Engineering, 2070 Neil Avenue, Columbus, OH 43210. (614) 292-8143.

OCTOBER 4-6
AGMA – GEAR EXPO '87

Cincinnati Convention Center
Cincinnati, OH

OCTOBER 5-7
AGMA – FALL TECHNICAL MEETING

Hyatt Regency Cincinnati
Cincinnati, OH

For further information contact: AGMA 1500 King St., Suite 201, Alexandria, VA 22314 (703) 684-0211.

OCTOBER 7-9
COMPUTER-AIDED GEAR DESIGN PROGRAM
Wisconsin Center, Madison, WI

The University of Wisconsin-Milwaukee presents a three-day seminar for engineers designing and specifying gears or gear drives. It will provide a frame work in which the student may develop a computer-aided design system for his or her individual needs. Basics of both gear design and use of microcomputers will be covered. For more information, contact John M. Leaman, Center for Continuing Engineering Education, Univ. of Wisconsin-Milwaukee, 929 North Sixth Street, Milwaukee, WI 53203. (414) 227-3110.

NOVEMBER 17-19
SME GEAR PROCESSING & MANUFACTURING CLINIC
Michigan Inn, Southfield, MI

Three days of presentations and discussions. Topics to be covered include gear finishing, hardening, broaching, grinding, shaping, inspection and chart evaluation, brush finishing and deburring, hobbing and hob design, choice of materials for cutting tools and shaper cutter design. Tuesday evening Nov. 17, will feature tabletop exhibits of the latest gearing products. For more information about attendance of exhibition space, contact Joe Franchini at SME, One SME Drive, P.O. Box 930, Dearborn, MI, 48121. (313) 271-1500, ext. 394.

CALL FOR PAPERS

ASME 5TH INTERNATIONAL POWER TRANSMISSION AND GEARING CONFERENCE

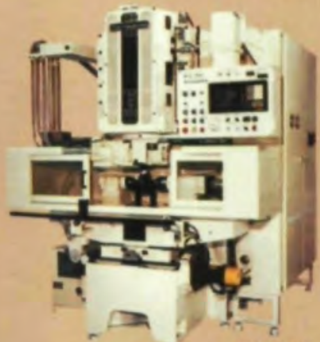
Deadline for submissions for this spring, 1989, conference is **December 31, 1987.** For more information contact: Donald L. Borden, P.O. Box 502, Elm Grove, WI 53122. (414) 784-9363.

When it comes to gear machining needs, Mitsubishi cuts the leading edge.

GA15CNC

SA25CNC

GA25CNC



FA30CNC

GA63CNC

Gear Cutting Tools

Mitsubishi mechanical and electronic technologies come together today in a full lineup of the most advanced gear cutting machines on the market. Whether your requirements call for hobbing, gear shaping or gear shaving, including flexible gear manufacturing systems, Mitsubishi offers the ideal solution for any need. All machines are unsurpassed in flexibility, speed, accuracy and rigidity — resulting in new standards of productivity. For full information on the entire lineup, along with available tools, contact us today.

SPECIFICATIONS

Model		Max. Workpiece Dia. (in)	Max. Workpiece Pitch (D.P.)	Main Motor (HP)	Machine Weight (Lbs)
Gear Hobbers	GA15CNC	5.9/7.9	6.35	10	8,800
	GA25CNC	9.8	4	10	17,600
	GA40CNC	15.7	4	10	18,700
	GA63CNC	24.8	1.8	20	24,200
Gear Shapers	SA25CNC	9.8	4	7.5	11,000
	SA40CNC	15.7	4	10	15,600
	SA63CNC	25.6	3.2	24.7	21,100
Gear Shaver	FA30CNC	12.2	3.2	7.5	11,500

GEAR EXPO '87
Booth #323

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CIRCLE A-8 ON READER REPLY CARD


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CNC
HOBBING
systems compatible
floor TO Floor
auto-load

BACK ISSUE INDEX

FORGET EVERYTHING YOU ALWAYS KNEW ABOUT PRODUCING GEARS

The production of parallel-axis — spur and helical — gears has just undergone such a major change that most machinery in use today is obsolete.

Gleason CNC Systems compatible hobbing machines have written a whole new set of rules on how to produce spur and helical gears profitably. The Gleason CNC hobbers offer highest volume, highest degrees of flexibility and fastest changeover. What's more you can get automatic hob and fixture change plus automatic part load/unload.

That's why auto transmission plants throughout the world are equipped with Gleason hobbers. And that's why Gleason's production hobbing capability is rapidly becoming the world standard for gear production.

The real Gleason plus is the worldwide network of service and application engineering support that Gleason provides. The result consistently is this: maximum equipment reliability, maximum uptime and highest quality gear output.

Call now to learn more about the parallel-axis gear production of today — and tomorrow. The Gleason Works, 1000 University Avenue, Rochester, NY 14692. Detroit (313) 353-5205.

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