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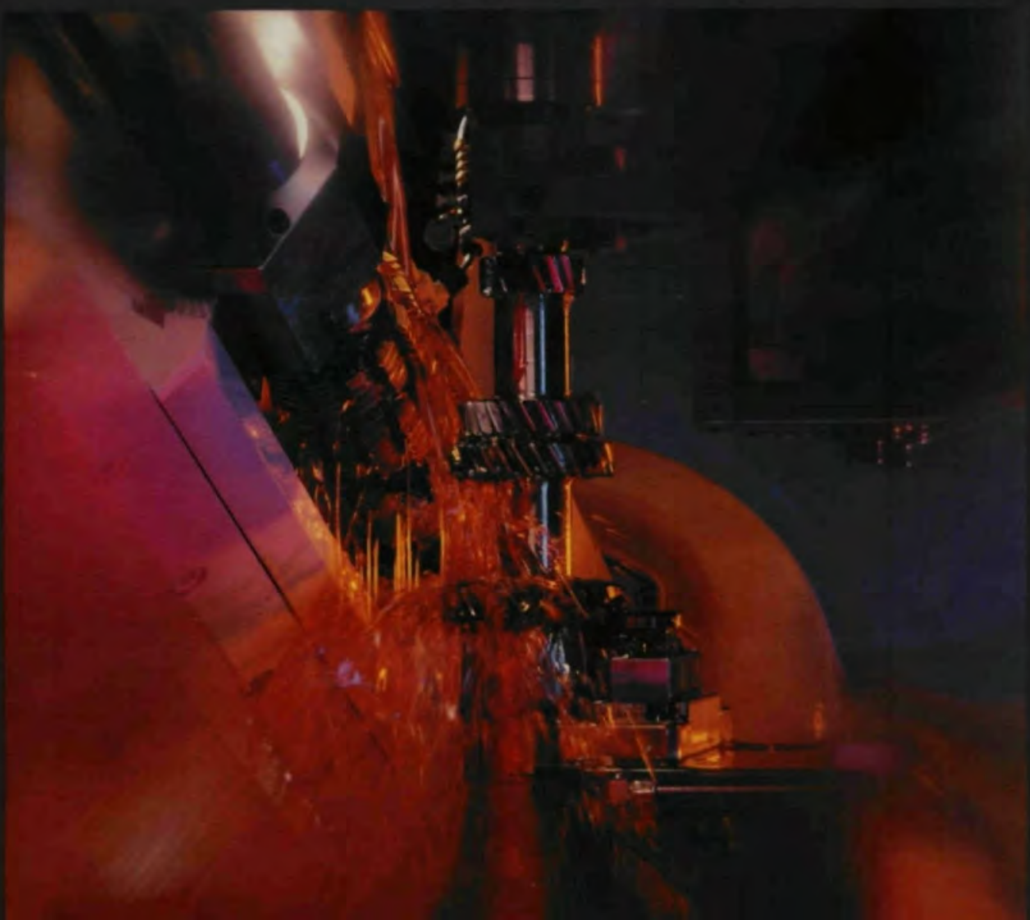
GEAR TECHNOLOGY

The Journal of Gear Manufacturing

SEPTEMBER/OCTOBER 1990



The 332 Report on the Gear Industry

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Involute Tooth Profiles

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GEAR TECHNOLOGY

The Journal of Gear Manufacturing

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September/October, 1990 Vol. 7, No. 5

GEAR TECHNOLOGY, The Journal of Gear Manufacturing (ISSN 0743-6858) is published bimonthly by Randall Publishing Co., Inc., 1425 Lunt Avenue, P. O. Box 1426, Elk Grove Village, IL 60007. GEAR TECHNOLOGY, The Journal of Gear Manufacturing. Subscription rates are: $40.00 in the United States, $50.00 in Canada, $55.00 in all other foreign countries. Second-Class postage paid at Arlington Heights, IL and at additional mailing office.

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

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THE 332 REPORT

Competitive Position of the U.S. Gear Industry in U.S. and Global Markets A Summary

In May of this year the U.S. International Trade Commission made public its Report to the President on the condition of the U.S. gear industry. This 200+-page document is the result of a two-year study by the commission, with the help of the AGMA staff and members. It is the most comprehensive and current analytical coverage of industry conditions and trends presently available. Because of the importance of this report to the industry, GEAR TECHNOLOGY is devoting a good portion of this issue to reprinting the Executive Summary for our readers.

I strongly encourage you to study this summary carefully. It contains both good news and bad news for our industry and a great deal of food for thought. Now the IMTS, where the larger global machine tool market gathers in Chicago to display the newest machinery available, is nearly upon us, providing even more ideas and food for thought. There is no better time to consider the state of our industry and what our strategies for growth, development, research, and investment should be. The Trade Commission has supplied us with the facts we need to plan for the future; it is up to us to use this information to our advantage.

For complete copies of the USITC report, write to Kenneth R. Mason, Secretary to the Commission, U.S. International Trade Commission, Washington, D.C. 20436.

Michael Goldstein

Editor/Publisher

In March 1989, the U.S. Trade Representative requested the U.S. International Trade Commission to conduct an investigation and prepare a report on the competitive position of the U.S. gear industry in U.S. and global markets. The USTR request makes the following observation regarding the U.S. gear industry:

"The U.S. gear manufacturing industry produces components that are essential to most industrial and transportation equipment. The industry, which has experienced a dramatic increase in imports since 1983, is unable to assess properly its trade concerns because U.S. government and private data on the industry's production and trade composition are fragmented and incomplete. The American Gear Manufacturers Association has formally requested assistance providing the industry with a comprehensive set of objective data."

The diversity of the group of companies that comprises the U.S. gear industry complicates the collection and compilation of data on the gear industry. However, through a questionnaire survey of U.S. gear producers, importers, and distributors, as well as domestic and international interviews with industry experts, the Commission was able to develop a considerable database on the U.S. industry and market and provide an assessment of the conditions of competition in the gear industry.

The principal findings of the Commission's assessment of the U.S. gear industry are as follows:

I. Profile of the U.S. gear industry

- *In 1988, the U.S. gear industry consisted of more than 300 firms having shipments of $14.8 billion and production worker employment totaling 84,600 persons.*

Gears and gearing are intermediate products which are essential to a wide range of U.S. finished product industries. The four principal markets for gears and gearing are the motor vehicle, industrial products, aerospace, and marine industries. Approximately 80 percent of gear industry shipments, $11.9 billion, were motor vehicle gearing in 1988. Shipments of industrial gearing totaled $1.7 billion; aerospace gearing shipments totaled $928.7 million; and marine gearing shipments totaled just $275.6 million. The U.S. gear industry exported a total of $2.4 billion in 1988, or 16% of total shipments. U.S. gear consumers imported $2.7 billion in 1988, resulting in

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a gear trade deficit of $316 million in 1988, as import penetration rose to over 18% of total gear consumption.

- *During 1984-88, Canada, Mexico, the United Kingdom, Japan, Australia, and West Germany were the chief foreign markets for U.S. exports of gears and gearing products.*

These markets accounted for 67% of total U.S. exports in 1988. Canada has traditionally been the leading foreign market for U.S. exports of gears and gearing primarily because of the cross border structure of the automobile industry. In total, exports of motor vehicle gears and gearing accounted for 90% of U.S. exports to the 6 leading foreign markets, and most exports were sent to foreign subsidiaries or partners of U.S. firms.

- *Major structural changes took place in the U.S. industry during 1984-88.*

The domestic gear industry has experienced a number of mergers, acquisitions, leveraged buy outs, and joint ventures in recent years, following a period of divestitures prior to 1984. Some U.S. firms have acquired interests overseas to expand their markets, although much of the activity in international acquisitions has been foreign firms investing in new U.S. facilities.

- *The U.S. market for gears and gear products grew by nearly 25% during 1984-88, and accounted for more than one-third of global consumption.*

The U.S. market for gears and gear products is the largest in the world and during 1984-88 rose 25%, from $12.0 billion to $15.1 billion. U.S. imports grew from $1.7 billion to $2.7 billion, or by 57%, during 1984-88. Import penetration rose from 15% in 1984 to 18% in 1988. In 1988, the U.S. market accounted for 35% of global consumption, which is estimated at $42.6 billion.

- *Increased U.S. gear and gearing imports during 1984-88, principally supplied by Canada, Japan, France, and West Germany, were attributable to three factors.*

U.S. imports increased during 1984-88 principally because of (1) U.S. original equipment manufacturers, as a cost-lowering measure, bought less expensive gearing from foreign sources; (2) major Western European and Japanese producers were successful in their concerted efforts to penetrate the U.S. market; and (3) Japanese parts producers supplied the growing number of Japanese-owned auto manufacturers in the United States. In the early 1980s, flagging demand in home markets and the strong dollar made the U.S. gear market attractive to foreign producers. Many U.S. gear consumers were facing difficult market conditions and turned to imported gearing which, largely due to the exchange rate, was often less expensive than the comparable U.S. product. A more recent trend is an increase in imports of gearing by foreign-owned U.S. assembly plants, especially automotive, from their parent companies.

U.S. demand for aerospace gears grew significantly during 1984-88, with imports nearly doubling during this period.

- *In the U.S. market, the largest component of consumption is motor vehicle gearing, a market that is strongly influenced by quality considerations.*

In 1988, apparent U.S. consumption of motor vehicle gears and gearing accounted for nearly 80% of total consumption of gears and gearing; consumption of motor vehicle gearing increased from $9.3 billion in 1984 to $11.9 billion in 1988. Imports accounted for 16% of U.S. apparent consumption of motor vehicle gearing in 1984 and 18% in 1988. A large percentage of these imports are from U.S. subsidiaries located in Canada. Imports from Japan are primarily used in Japanese automotive transplant assembly operations in the United States. The motor vehicle industry is characterized by rapid technological change in virtually all major vehicle systems and producers must be somewhat innovative to remain competitive. Product quality is an especially important consideration for vehicle gear producers and the use of cubic boron nitride grinding technology is becoming a critical element in remaining competitive.

- *In the U.S. market, industrial gears and gear products, the second most important market sector, grew irregularly during 1984-88, but imports' share of the market more than doubled.*

In 1988, apparent U.S. consumption of industrial gears and gear products accounted for 14% of total consumption of gears and gearing; consumption of industrial gears rose from $1.8 billion in 1984 to $2.1 billion in 1988. Imports accounted for 15% of U.S. apparent consumption in 1984, but rose to 27% in 1988. The increase in imports resulted from increasing consumer demand for quality products competitively priced, especially by foreign-owned gear assembly operations. The U.S. market for industrial gearing is directly related to the overall investment in new plant and equipment in the manufacturing sector and to expenditures on public works.

- *U.S. demand for aerospace gears grew significantly during 1984-88, with imports nearly doubling during this period.*

In 1988, apparent U.S. consumption of aerospace gearing accounted for 6% of total consumption of gears and gearing; consumption of aerospace gears increased from $738.0 million in 1984 to $834.0 million in 1988, or by 13%. Aerospace gear imports nearly doubled from $25.0 million in 1984 to almost $50.0 million in 1988 and the ratio of imports to consumption rose from 3 to 6% during this period. The demand for aerospace gears is heavily influenced by the demand for helicopters. Despite a downturn in demand for helicopters, however, overall demand for aerospace gears increased during 1984-88 because of the unprecedented increase in sales of large civil transport vehicles.

- *U.S. demand for marine gearing remained level during 1984-88, but softened toward the end of this period for small marine gearing, as imports obtained a larger share of the market.*

In 1988, apparent U.S. consumption of marine gears accounted for 2% of total apparent U.S. consumption of gears and gearing. During 1984-88, U.S. apparent consumption of these gears rose irregularly, ranging from a low of $249 million in 1985-86 to a high of $275 million in 1988, whereas the import-to-consumption ratio rose from 2% in 1984 to 4% in 1988. Increased imports of large marine gearing occurred in both the government and commercial markets, due, in part, to lower prices. In late 1988, consumption of small marine gears began to fall as sales of pleasure craft softened due to saturation of the market.

- *The overall number of production workers in the U.S. gear industry declined 3.6% during 1984-88.*

There were an estimated 84,600 production workers in the U.S. gear industry in 1988, down from 87,800 in 1984. Employment declined by 6.4% between 1984 and 1987 and then increased by 2.9% between 1987 and 1988. The overall decrease in employment in the U.S. gear industry reflects increased automation and flat shipment trends of the industrial and marine gear sectors. However, employment showed a slight increase during the last year of the period; this increase can be attributed to an upturn in the market in 1987 which necessitated an increase in employment.

- *Nominal wages for all U.S. gear production workers rose significantly; however, wages in real terms reflected an increase of only 3%.*

Total compensation, including fringe benefits, bonuses, and payments in kind, remained relatively stable for the period, declining by 2% in real terms, although in nominal terms, total compensation costs increased by 11%. Wages also declined in real terms, by 4%, while increasing 8% in nominal terms. Annual productivity per worker rose by 17% in real terms.

- *Skilled personnel necessary for U.S. gear manufacturing operations are in short supply.*

Machinists and trainees with the necessary mathematical skills to become machinists are most in demand. Firms attribute the scarcity of workers to generally low unemployment, insufficient numbers of high school graduates with adequate mathematical and verbal skills, and the low status of blue-collar jobs. On-the-job training has a significant cost, as it requires taking otherwise productive skilled workers away from their tasks in order to train new workers. Subsequently, some firms have worked with vocational schools to develop programs covering rudimentary skills, such as blueprint reading and basic machine operations. Many firms report high retention rates among workers recruited from these schools.

R & D expenditures by U.S. gear producers increased during 1984-88.

- *During 1984-88, U.S. gear manufacturing capacity declined an estimated 9%.*

The decline in capacity is based upon a number of different indicators such as plant closings and declines in employment; however, partially offsetting such changes were increases in productivity, as well as the rationalization of inefficient operations. For example, a decline of 15% for machinery in place was offset by the introduction of newer, more efficient gear-cutting and finishing machine tools which resulted in improved productivity. Decreases in capacity of some firms owned by U.S. producers have partially been offset by new capacity added by foreign-owned gear producers as well as by other U.S. firms.

- *The level of capacity utilization by U.S. producers varied substantially among firms producing for different markets.*

For the U.S. gear industry as a whole, capacity utilization was 71% in 1988, as measured in actual machine hours spent producing gears compared with available machine hours. Many captive producers manufacturing gears and gearing for the automotive and construction equipment industries have been operating at higher levels of capacity utilization, in some instances close to 100%. Most producers of gears and gearing for the aerospace and specific industrial and marine products markets have been operating at lower rates of capacity utilization.

- *The level of profits generated by most U.S. gear producers trended upward.*

The increase in operating margin during 1986-88 was partly attributable to the general improvement in the economy, especially in the automotive and machinery sectors. Net sales rose slightly faster than production related costs. Although the percentage increase in operating income was nearly twice that of sales, net income before taxes rose only 11.4% during 1986-88 as a result of a more than doubling of non-production-related expenses, such as interest expense, plant closing losses, and write-offs of assets.

- *Companies that can convince lenders that they will continue to generate revenues and that they have valuable assets are likely to have an advantage in the capital markets over small job shop operations.*

The ability of gear producing firms to obtain financing and the rates at which they borrow money are determined largely by the financial strength of the individual company. The large proportion of companies in this industry that are small do not have a high net asset value or an expected stream of future revenues from long term contracts. They often find most conventional means of financing unavailable or unaffordable. Gear-producing subsidiaries of large companies, such as captive producers in the automotive market, generally meet their capital needs through their corporate financial centers and thus may obtain capital at lower rates or in different ways than are available to smaller firms. U.S. bank lending rates for short-and medium-term financing needs of the private sector declined from slightly over 12% in 1984 to approximately 9% in 1988.

- *During 1984-88, the trend in capital expenditures for gear-producing machine tools in the United States increased, but continued to lag behind the expenditure levels of foreign producers.*

Total expenditures on gear-producing machine tools by U.S. firms rose 48% between 1984 and 1988, although such expenditures fell 11% between 1987 and 1988 to an estimated $56 million. In spite of the increase during 1984-88, 1988 U.S. expenditures were substantially below the 1980 level. Expenditures for this type of machinery by West German and Japanese producers were significantly higher than for U.S.

firms during 1984-88 and totaled over $130 million in 1988 in each of these two countries.

- *R&D expenditures by U.S. gear producers increased during 1984-88.*

R&D expenditures by the U.S. gear industry rose annually during 1984-88, increasing from $53.8 million in 1984 to $77.7 million in 1988, a 44% gain, but did not keep pace with such expenditures by broader industry groups. Gear industry R&D expenditures represented less than 1% of shipments during 1984-88, whereas the level of R&D for nonelectrical machinery industries, a similar but broader group, totaled 3.5% of sales in 1987. University gear research in the United States has lagged behind that performed in West Germany and Japan. Traditionally, the bulk of gear R&D in the United States is done at the company level and is generally not shared. Several ongoing projects in the United States, especially the work of ASME Gear Research Institute and the Defense Logistics Agency's newly established Instrumental Factory for Gears (INFAC), are designed to improve the competitive position of the U.S. gear industry.

II. Profile of major foreign gear industries.

- *The Japanese gear industry had shipments of $8.4 billion in 1988 and employed an estimated 39,000 persons.*

Japan's gear industry shipments were predominantly motor vehicle gearing, with the bulk of the remainder accounted for by industrial and marine gearing. Japan's aerospace gearing industry is relatively small, but is growing through licensing agreements for larger components, such as engines, and through co-production of aircraft with U.S. and Western European aerospace producers. In 1988, the Japanese gear industry served a domestic market estimated at $6.0 billion, and its exports totaled an estimated $2.5 billion. Approximately 83% of exports were of vehicle gearing. Imports of gearing products totaled just $90 million and consisted mainly of industrial and vehicle gearing. Major foreign suppliers were the United States, France, and West Germany.

- *The West German gear industry had shipments of $4.8 billion in 1988 and employed an estimated 23,000 persons.*

West Germany is a technological leader in industrial gearing; in contrast with other major producers, shipments of industrial gearing accounted for approximately half of production. West Germany is also a leader in marine gearing, especially for diesel engines, and a signficant number of firms produce for this market. The West German gear industry serves a domestic market estimated at $3.2 billion, and exports about half of its production, or $2.2 billion. Imports totaled $521.7 million, accounting for about 17% of domestic consumption, about the same percent as in the United States, and were primarily from France, Italy, and other EC countries.

- *Other important EC suppliers had aggregate shipments of $6.4 billion and employed an estimated 32,500 persons.*

Italy, France, Belgium, and the United Kingdom are all highly industrialized, technologically advanced countries with significant gear producing industries. All four countries are involved in the vehicle and industrial gearing sectors. France, the United Kingdom, and Belgium also produce aerospace and marine gearing. The majority of firms in each country are described as small- to medium-sized firms, operating as subsidiaries of multinational producers, as captive suppliers to the

During 1984-88, U.S. gear manufacturing capacity declined an estimated 9%.

vehicle or aerospace sectors, or as independents operating in niche markets.

- *Other suppliers include Canada, Korea, Taiwan, Mexico, and some of the newly industrialized countries.*

The gear and gear products industry in Canada is closely integrated with U.S. vehicle producers; the gear industry in Korea is also highly dependent on vehicle producers, both domestic and Japanese. Taiwan has designated its gear industry as a "strategic industry," permitting it to have preferential treatment. Major gear producers in Mexico and Brazil produce primarily for domestic consumption. China has an almost unlimited supply of low-cost labor and the potential to become a major supplier in the future, and Singapore is a focal point for transshipments among other Asian countries.

III. Assessment of the global market for gears.

- *Estimated world consumption of vehicle, industrial, aerospace, and marine gearing, measured in terms of U.S. dollars, rose sharply during 1984-88, but experienced only moderate growth when measured in national currencies.*

During 1984-88, the Commission's estimate of world consumption of vehicle, industrial, aerospace, and marine gearing, in terms of U.S. dollars, increased from $20 billion to $25 billion in 1984 to $40 billion to $45 billion in 1988. However, if these measurements utilized national currencies that have appreciated against the dollar, the change in production and consumption would be considerably smaller. For example, during 1984-88, production of gearing in West Germany increased by 107% as measured in U.S. dollars, but production as measured in Deutsche marks rose by 28%.

During 1984-88 the value of the U.S. dollar changed significantly compared with the currencies of countries importing to the U.S.

- *Motor vehicle gearing represents more than 60% of world production and consumption of gears and gear products; the remainder is accounted for by industrial, aerospace, and marine gearing. The United States was a principal supplier to all markets, except marine gearing.*

The largest producers and consumers of vehicle gearing are those countries that have the most significant automotive industries, namely the EC countries, the United States, Japan, and Canada. Korea was a significant producer and consumer, although imports account for an important, but decreasing, part of its total needs. West Germany, the United States, and Japan are the world's largest sources and markets for industrial gearing. The United States is not only the single largest producer of aerospace gearing, but the largest individual market as well.

- *During 1984-88, world capacity in the gear industry grew in most countries.*

The number of facilities and investment in new machinery increased, especially during 1986-88, as the world economic situation improved. This was particularly true in newly industrialized countries; during 1984-88, domestic shipments of Korea and Taiwan, for example, increased nearly 94% to $280 million and 153% to $124 million, respectively. These and other emerging suppliers are expected to become a greater force in the world market over the next 10 years.

- *During 1984-88, the value of the U.S. dollar changed significantly compared with the currencies of many countries exporting gearing to the United States.*

Western European currencies and the Japanese yen appreciated sharply against the dollar in 1986 and subsequent years. Against these currencies, the real exchange rate index increased by 30 to

50% or more during 1984-88. The relative decline of the dollar, all other things being equal, should make U.S. products more price competitive and U.S. imports more expensive.

- *Excluding the non-market economies, 1988 world exports of gearing totaled an estimated $11.2 billion and world imports totaled $8.8 billion.*

In 1988, the largest exporters were Japan and the United States (22% each), West Germany (19%), and France (10%). The major importing countries in 1988 were the United States (31%), Canada (20%), and the United Kingdom (11%). The demand for gearing in these countries was principally for automotive gearing. Japanese automobile transplants in the United States and U.S. automobile producers' subsidiaries in Canada dominated the trade flows within, as well as into and out of, North America. Japan's exports as a share of production were 29%, compared with almost 45% for West Germany, 16% for the United States, and 53% for France.

- *The major suppliers and consumers of gearing in the non-market economies of the world are the Soviet Union, Hungary, East Germany, and China.*

Nonmarket economies supplement their own production with some imports, mainly from Western Europe. Production in these nonmarket economies, as well as in South America, Africa, and South Asia, is mostly destined for internal markets, but is insufficient to meet total demand.

- *Product standards in gear trade are an important marketing tool and the ability to manufacture to a variety of standards is an important asset for gear producers.*

Despite the fact that standards are voluntary, they are often used by private and public procurement officials in tender documents and may attain the status of a de facto requirement in particular countries. One of the most widely used standards is the DIN of West Germany. The American Gear Manufacturers Association (AGMA) has become more active in the International Standards Organization (ISO) during the last few years and has had some success in influencing ISO standards drafting. AGMA standards are receiving wider acceptance because of an emphasis on "serviceability" compared with the more "academic" approach used for developing other countries' standards.

IV. Comparison of U.S. and foreign producers' strengths and weaknesses.

- *Raw material costs are comparable for gear manufacturers worldwide. However, the cost of bearings used in gear products has increased for U.S. producers.*

According to U.S. and foreign industry sources, Japanese, European, and U.S. gear producers face fairly comparable material costs. Since mid-1989, however, U.S. manufacturers have paid a higher price for bearings due to a decline in U.S. production and antidumping tariffs on bearings imported from key foreign suppliers. The costs of the resulting shortages and double-digit bearing price increases have been passed on to customers, reducing U.S. producers' price competitiveness.

- *The United States experienced less growth in real hourly compensation costs for production workers in 1984-88 than did most of its Western European competitors.*

In 1988, world gearing exports were an estimated $11.2 billion and imports were $8.8 billion.

When adjusted for inflation, hourly compensation costs for U.S. production workers were unchanged from 1984 to 1988; in West Germany, they increased in real terms by 3.5% over the period. In Japan and Canada, however, they fell by 3.1%

and 0.7%, respectively.

- *The supply of skilled labor worldwide has tightened in recent years, and employers are pursuing a variety of training programs to ease the shortage.*

As the current workforce ages, major world producers are finding it difficult to fill entry-level and skilled manufacturing positions. Geography, labor force mobility, and the economy are all factors; in addition, young people are not entering the skilled manufacturing trades. Employers are assuming a major role in training new hires in a wide range of skills.

- *During 1984-88, U.S. interest rates were higher on average than those in West Germany and Japan, but lower than those in other major gear producing nations.*

U.S. bank lending rates averaged 9.6% during 1984-88, compared with 9% for West Germany and 5.8% for Japan. In other European countries, the rates ranged, on average, from 10.5% in the United Kingdom to 16.9% in France.

- *U.S. gear producers are disadvantaged relative to European and Japanese manufacturers in gaining access to capital.*

Domestic producers believe that competing successfully in the future requires current capital expenditures to upgrade equipment. U.S. and foreign industry officials feel that in the United States, investors typically focus on short-term profitability, unlike foreign investors who generally consider return on investment over the long term. One of the results of this is that lending rates for research projects with long lead-times are two to three times higher in the United States than in many other countries. Operating with lower profit margins than their foreign competitors, U.S. firms lack retained earnings, and the majority are not large enough to have easy access to capital markets. In contrast, certain of their foreign competitors have relationships with larger firms and banks which assure more ready availability of capital. In the United States, the integration of financial institutions and industry that is prevalent in countries such as West Germany and Japan, is prohibited.

- *University research and development expenditures in Japan and West Germany far exceeded those of the United States, but technology leaders differ by market sector.*

The United States spent less than $1.0 million in university gear research in 1985, as compared with an estimated $3.8 million in West Germany and $5.0 million in Japan in the same year. Both West Germany and Japan have extensive gear research centers in universities, cooperating and sharing information with private corporations and government agencies. In the United States only a few of these centers exist; almost all research is done at the company level and remains proprietary. While the U.S. is believed to be the leader in aerospace gear technology, it lags behind its competitors in technology for automotive and marine applications, for which West German firms are believed to have an advantage. No clear leader in industrial gearing technology has emerged.

> University research and development expenditures in Japan and West Germany far exceeded those of the U.S., but technology leaders differ by market sector.

- *Most U.S. gear manufacturers lag behind their Japanese and Western European counterparts in adopting new machine-tool technology.*

During 1984-88, U.S. expenditures for gear-making machine tools were $264.0 million, compared with $542.8 million for West Germany and $428.4 million for Japan. The world's leading machine tool manufacturers are located in Japan and Western Europe, particularly in West Germany. As a result, gear producers located in or near those countries can experiment with and integrate the latest in machine-tool technology in their families before it arrives in the United States.

- *The U.S. machine tool industry ranks behind Western European and Japanese machine tool builders for some critical types of machinery.*

Industry sources indicate that the technology and quality of West German, Swiss, and Japanese gear-making machine tools equal or surpass that of U.S. producers. For instance, West German and Swiss machine tool builders excel in bevel gear grinding machine tools, and Japanese manufacturers produce excellent hobbing and grinding machines. Foreign machine tool firms are characterized as large, technologically advanced, multi-product firms known for high quality, moderately priced products; some are subsidiaries of much larger firms. U.S. machine tool firms, while technologically advanced, are smaller and more specialized.

- *On average, the equipment currently in use by U.S. manufacturers is older than that of West German and Japanese producers.*

According to trade surveys, 88% of the gear-cutting and finishing machine tools in use in the United States in 1989 were more than 10 years old; in Japan, only 63% were of that age. West German sources estimate that the average age of critical manufacturing machines is less than 10 years. Older machinery tends to require more frequent maintenance and repair, which reduces its productive time. Also, technology embodied in new machinery enables manufacturers to maximize their productivity.

V. U.S. and foreign industry and U.S. consuming industry views.

- *U.S. manufacturers claim that some government actions have harmed the competitiveness of the U.S. gear industry in global markets.*

U.S. manufacturers claim that antitrust and product liability laws, tax policy, OSHA and EPA regulations, and other government policies harm their competitiveness; moreover, according to U.S. producers, incentives to export are practically nonexistent. In contrast, a number of foreign producers receive support from their governments, which allows them to be more competitive. This support includes accelerated depreciation for new machinery, encouragement for mergers and acquisitions, and, in most European countries, government rebating of VATs. The following specific taxation issues concern many U.S. gear manufacturers: (1) the treatment of depreciation under the Modified Accelerated Cost Recovery System; (2) the corporate alternative minimum tax; (3) the elimination of the Investment Tax Credit; (4) the current tax treatment of capital gains; (5) the treatment of "goodwill" under the U.S. tax code; and, (6) changes in the present tax code concerning foreign tax credits.

- *U.S. gear producers claim U.S. product liability laws inhibit research and development efforts.*

U.S. producers' insurance costs have risen dramatically in recent years in the face of product-liability lawsuits. As a result, according to industry sources, some firms cannot afford the high insurance premiums and have been forced to curtail or eliminate research and new-product-development efforts. Many U.S. firms feel that, in order to avoid product liability problems, they must produce only proven designs with extra measures incorporated to ensure durability and longevity, and to stress design of products to more stringent standards. This hinders them from competing against foreign companies that can more readily offer new products and designs.

U.S. firms maintain that while businesses and manufacturers should be held liable for injuries caused by their products due to their own negligence, liability laws must be uniformly enforced and penalties reasonable. Under the current system, U.S. businesses assert that they can be forced to pay large settlements for injuries that they did not cause; it is now always necessary in the U.S. legal system to show that the target of such a suit was responsible for injuries. U.S. firms maintain that this gives foreign firms a competitive edge over their U.S. counterparts since other industrialized countries have a fault-based standard of liability or other judicial or institutional differences that reduce the uncertainty of liability lawsuits. The fault-based system sets more rigorous standards for the proof of fault and the proof of the absence of contributing fault on the part of the plaintiff.

U.S. distributors cite improved product assortment, price, quality, service, and lead time as the primary areas U.S. producers need to address to remain competitive in the U.S. market.

- *Certain U.S. Department of Defense policies are eroding the U.S. defense industrial base, according to some U.S. producers.*

Some U.S. producers believe they are harmed by the Defense Department's practice of purchasing on initial bid price rather than the life cycle cost of the product. This policy favors the low-cost producer, whether it is the manufacturer that has invested heavily in research and development to produce a superior product or another, perhaps less knowledgeable, producer. Other sources believe that defense weapon systems are increasingly relying on foreign gears and gear products purchased as a result of offset agreements or of contracts awarded to the lowest bidder. Some firms have advocated the strict enforcement of "Buy America" procurement regulations in order to counter shifts in purchases to foreign goods.

- *U.S. distributors cite improved product assortment, price, quality, service, and leadtime as the primary areas U.S. producers need to address in order to remain competitive in the U.S. market.*

Some U.S. distributors criticize U.S. gear manufacturers for not offering a complete assortment of gear products at a competitive price. U.S. distributors believe that if U.S. manufacturers are to retain their market share, they must develop products that are competitive in terms of quality and price, increase communications with customers, shorten lead times, and build export marketing networks. Others feel that cost structure and design factors must be reexamined to reduce prices and R&D must be increased. Foreign producers believe that U.S. production is primarily intended for the domestic market and is therefore not truly competitive with the assortment of products available from foreign sources.

- *U.S. producers expressed concern over the way gears and gear products are currently classified under U.S. Government statistical programs.*

They are concerned that a large part of current domestic industry activity is not covered by the Standard Industrial Classification system. Similarly, import statistics of products from other countries (especially

Canada) to the United States are not collected in categories that are useful to the domestic industry.

- *U.S. producers expressed concern over the current pattern of foreign investment in the United States.*

U.S. producers are facing increased competition from foreign-owned firms that are locating in the United States in order to increase their market share. Such firms are not investing in existing U.S. operations, but are constructing new facilities or are establishing marketing agreements with U.S. distributors. Foreign automobile manufacturers are locating in the United States and are sourcing gears from their home countries.

- *U.S. industry sources allege unfair trade practices by foreign suppliers, citing as an example import prices that are substantially lower than U.S. producers' prices, despite unfavorable exchange rates for the imports.*

Foreign suppliers state that price differences are a result of their different gear production technology and the production of gears for different applications. Domestic firms advocate the implementation of reciprocal trade agreements between the United States and those countries exporting to the United States, and matching U.S. import tariffs with those faced by U.S. exports.

- *U.S. firms indicated that trade barriers significantly inhibit the free flow of U.S. exports into major foreign markets.*

Trade barriers named included high tariffs, import licensing requirements, technology transfer requirements, subsidies, local content requirements, exchange and other monetary or financial controls, and discriminatory sourcing. Countries most often cited with signficant barriers to trade include Japan, Argentina, Australia, Brazil, the EC member states, India, Mexico, Korea, China, and the Eastern Bloc.

- *According to U.S. manufacturers, finding and retaining skilled labor is difficult and current training programs are inadequate and outdated.*

A number of countries report a similar lack of skilled workers. Those U.S. firms that offer in-house training report that many employees leave for higher paying jobs with other firms. Unlike the United States, where training programs receive little or no government financing, assistance is provided for training programs in the EC and Japan.

In some countries, such as West Germany, vocational training and apprenticeship programs are used to train a skilled labor force. In other European countries and in Japan, however, such programs are not widespread and manufacturers express concerns similar to their U.S. counterparts regarding attracting younger employees to these programs.

Based on comparisons of the U.S. gear industry with the U.S. gross national product (GNP) and broader industry groups, growth in the total U.S. gear industry shipments have lagged behind that of the GNP, and the motor vehicle sector kept pace with that of the durable goods sector, and surpassed the growth in all manufacturing. Employment in the U.S. gear industry fell slightly during 1984-88, whereas it rose 3% annually in the motor vehicle industry and less than 1% in all manufacturing during the same period. Capital expenditures, especially among U.S. vehicle gear producers, increased substantially during 1984-86, as new machinery was required for new generations of automotive transmissions, and then declined. Such expenditures increased at an average annual rate of 4%, compared with 7% for all manufacturing during the period. ■

To obtain a complete copy of the 332 Report, write to Kenneth R. Mason, Secretary to the Commissioner, U.S. International Trade Commission, Washington, D.C., 20436. Ask for USITC Publication 2278.

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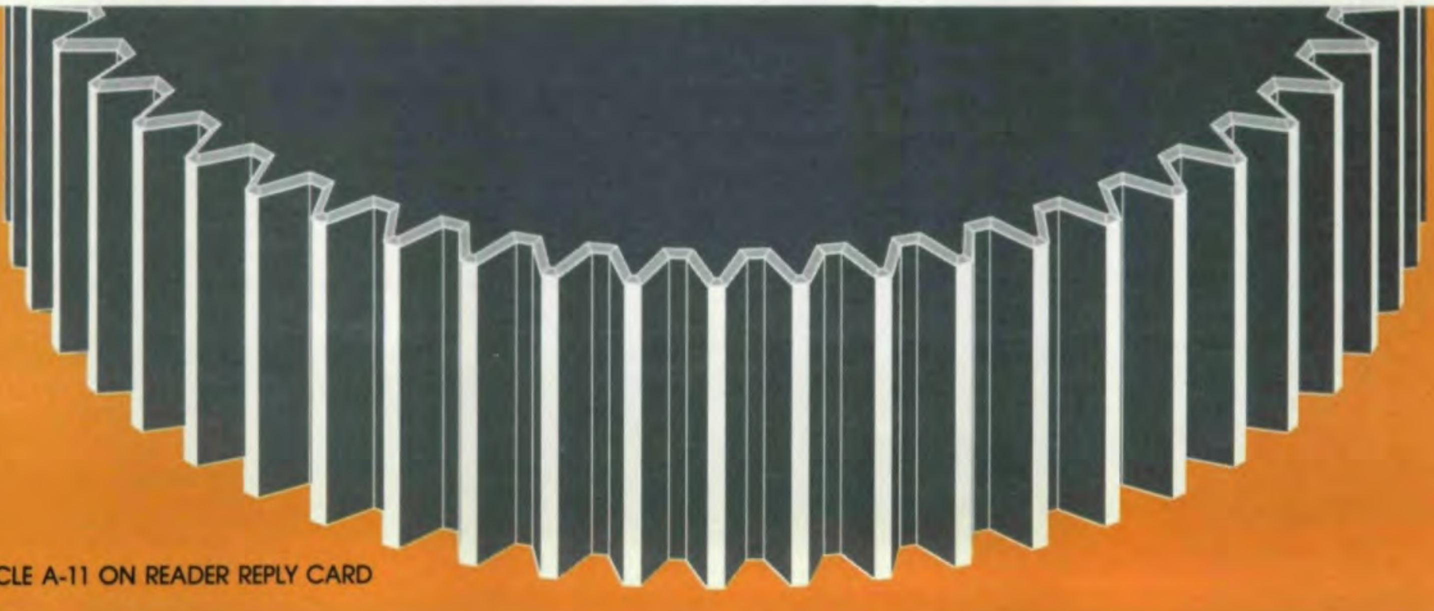

CIRCLE A-11 ON READER REPLY CARD

Accurate and Fast Gear Trigonometry

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Summary:

An accurate and fast calculation method is developed to determine the value of a trigonometric function if the value of another trigonometric function is given. Some examples of conversion procedures for well-known functions in gear geometry are presented, with data for accuracy and computing time. For the development of such procedures the complete text of a computer program is included.

Existing Complications

The many trigonometric functions in gear geometry may cause peculiar complications in the calculation of quantities. Very often the value of a certain function is known, and the value of another function has to be calculated, sometimes with available inverse functions, and sometimes without any other help other than laborious procedures. Such a complication already occurs in the first application of the definition of an involute.

An involute is defined on a base circle with fixed radius r_b. The polar coordinates of a point of an involute are

$$\frac{r_b}{\cos\alpha_t} \text{ and } \mathrm{inv}\alpha_t$$

(See Fig. 1.) The pressure angle α_t is a parameter in the two functions $r_b/\cos\alpha_t$ and $\mathrm{inv}\alpha_t$. Leaving the fixed radius r_b out of consideration, the two functions to be examined are the secant, $1/\cos\alpha_t$, and the involute function $\mathrm{inv}\alpha_t$. A point on the involute is determined by the variable transverse pressure angle α_t or by one of the above functions. If one of these functions is known, the other can be calculated.

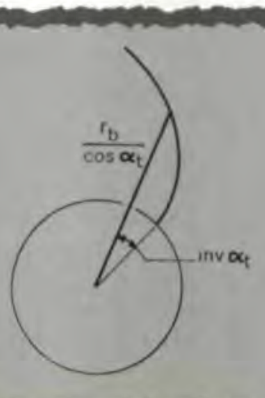

Fig. 1

From a given secant, $1/\cos\alpha_t$, the involute function can be determined easily in two steps.

$$\left.\begin{aligned} \mathrm{TALT} = \tan\alpha_t &= \sqrt{\left(\frac{1}{\cos\alpha_t}\right)^2 - 1} \qquad (1) \\ \mathrm{inv}\alpha_t = \tan\alpha_t - \alpha_t &= \mathrm{TALT} - \arctan(\mathrm{TALT}) \qquad (2) \end{aligned}\right\}$$

For instance, this sequence of calculations will be applied in the determination of the sum of addendum modification coefficients in a gear pair with given center distance. The square root and the inverse function, arctan, are standard functions in all computer languages.

The reverse calculation, starting with the involute function $\mathrm{inv}\alpha_t$ and finding the secant $1/\cos\alpha_t$, is of equal importance in gear calculations, but direct transcendental functions are not available in any computer language. Special iteration processes had to be written in each program. However, direct conversion procedures may be developed which are accurate, fast, and easy.

Common Calculation Methods

A common calculation method is an iteration process with three characteristics:

- *The introduction of one or two initial values,
- *The repeated use of an approximation formula or a set of formulae,
- *The procedural decision in each step based on the comparison of an intermediate result with an agreed tolerance value.

The *"regula falsi"* iteration method has two initial values, one below and the other above the expected solution. The approximation formula is a chord between two points, yielding a next point to be examined. The comparison decides whether the procedure can be stopped or with which points the procedure has to continue.

The "tangent" iteration method can be applied if a formula for the tangent of the function is known. Then only one initial value suffices. The approximation formula includes the tangent, yielding just one point to continue the procedure. The comparison only concerns the tolerance. An excellent example of such an iteration process[1-2] is

$$\left.\begin{array}{lr} \phi_1 = (3.\ \mathrm{inv}\alpha_t)^{0.330} & (3) \\ \phi_{j+1} = \phi_j + \dfrac{\mathrm{inv}\alpha_t + \phi_j - \tan\phi_j}{\tan^2\phi_j} & (4) \\ j = 1,2,3,\ldots \text{ until } \mathrm{abs}(\phi_{j+1} - \phi_j) < \text{tolerance} & (5) \\ 1/\cos\alpha_t \approx 1/\cos\phi_{j+1} & (6) \end{array}\right\}$$

In this process the involute function $\mathrm{inv}\alpha_t$ is given, and the transverse pressure angle α_t is approximated, after which the secant can be calculated. The iteration formula (Equation 4) replaces the true function by its tangent. (See Fig. 2.) The initial value (Equation 3) may be as decisive for the number of steps as the agreed value of the tolerance. In this case the initial value was adequate and only by a trial-and-error numerical assay could it be slightly improved to

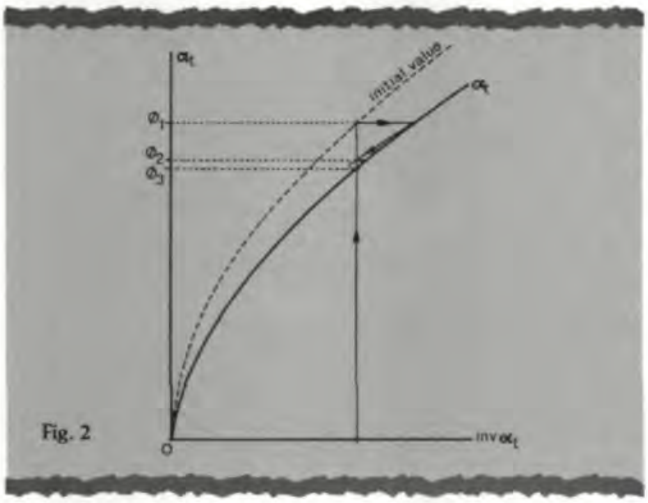

Fig. 2

$$\phi_1 = (2.80.\mathrm{inv}\alpha_t)^{0.333} \quad (7)$$

A significant improvement of the determination of the secant by any amendment in this iteration process could hardly be expected. Nevertheless, a development with a long history resulted in a new efficient procedure. In retrospect, the new method has a clear relation with the above iteration process, although "iteration" is abandoned.

A Promising Method

A very interesting method was presented by Jennings.[3] The main idea was to

* derive an expression to approximate the function for small values of the variables, and
* modify the expression to approximate larger values.

An expression suitable for small values is easily derived from the first terms in infinite series.

$$\tan\alpha_t = \alpha_t + \frac{1}{3}\alpha_t^3 + \ldots \quad (8)$$

$$\sec\alpha_t = \frac{1}{\cos\alpha_t} = 1 + \frac{1}{2}\alpha_t^2 + \ldots \quad (9)$$

For small values of α_t, Equations 8 and 9 simplify into

$$\mathrm{inv}\alpha_t \approx \frac{1}{3}\alpha_t^3 \quad (10)$$

$$\sec\alpha_t - 1 \approx \frac{1}{2}\alpha_t^2 \quad (11)$$

Elimination of the right-hand terms yields

$$\sec\alpha_t \approx 1 + 1.0400(\mathrm{inv}\alpha_t)^{2/3} \quad (12)$$

The approximation (Equation 12) does not hold for larger values, but Jennings was successful in manipulating a term $(\sec\alpha_t - 1)$ into (12) in such a way that it became a cubic equation with the root

$$\sec\alpha_t \approx 1 + 1.0400(\mathrm{inv}\alpha_t)^{2/3}\{1 + 0.3082(\mathrm{inv}\alpha_t)^{2/3}\} \quad (13)$$

The approximation (Equation 13) was introduced in gear calculations by Tuplin,[4] who recommended simple but effective methods preferably to be carried out with no other help than a slide rule. With respect to its simplicity, this approximation is remarkably accurate in a rather large range:

* For α_t between 0° and 53°, the approximation remains below the exact value, with a maximum error of -0.000252 for $\alpha_t = 46.2°$.
* Above $\alpha_t = 53°$, the approximation rises above the exact value with a rapidly increasing error.

Polynomial Method

The next step, References 5-8, applied in Reference 9, is obvious. Equation 13 looks like a series expansion in which the coefficients and the number of terms may be adapted to better accuracy. Such a development was not only needed for the move from the slide rule to the computer in a special case, but it also offers a general method for the conversion of several transcendental functions. The method follows.

* Determine a simple expression to approximate the function for small values of the variable. It may be an exponential expression inspired by the first terms of infinite series.
* Determine the range of the variable in which the result has to be sufficiently accurate.
* Apply a least square root method,[10-11] to establish a polynomial of a certain degree (concerns the number of terms in the formula) and with a certain number of decimal places (in the coefficients of the formula).
* Check the result and possibly select another degree or another number of decimal places.

The result of the above easy effort is an accurate and fast conversion procedure that can be applied in any computer program. Theoretically, it may be considered to be an improvement of the tangent iteration method, since it replaces the rectilinear tangent (one-degree polynomial) by a smooth curve (multi-degree polynomial). That smooth polynomial

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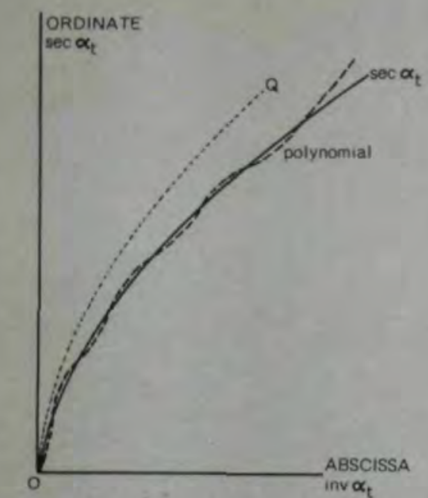

Fig. 3

curve approximates the function so closely that "iteration" is unnecessary. (See Fig. 3.)

The conversion of an involute into a secant may serve as an example of this method. The infinite series (Equations 8 and 9) were already simplified into Equations 10 and 11. The coefficients in (10) and (11) can be neglected, since the polynomial will produce new coefficients. It suffices to determine the exponent of an auxiliary expression. In this example it is

$$Q = (\text{inv}\alpha_t)^{2/3} \tag{14}$$

The relevant input of the polynomial program is

*the involute function, referred to as ABSCISSA,

*the secant function, referred to as ORDINATE,

Table 1 – Polynomial Program

```
program POLYNOMIAL(input,output);
var
   AL,EXPNT,fl,h,p,pp,pl,Q,XABSC,xl,Y,YAPPR:
   real; a,alfa,b,beta,f,pn,pnl,x: array[0..17] of real;
   DCML,i,j,k,n,NMAX: integer;
function ABSCISSA: real;
begin ABSCISSA:=sin (AL)/cos(AL)-AL
end;
function ORDINATE: real;
begin ORDINATE:=1/cos(AL)
end;
function EXPONENT: real;
begin EXPONENT:=2/3
end;
function DECIMROUND(X:real): real;
var MM,XF,XR,XX: real; M,DR: integer;
begin XR:=abs(X); XF:=round(XR);
   XR:=XR-XF; DR:=DCML; MM: = 1;
   repeat M:=10*M; DR:=DR-1
   until (DR=0) or (M=1000);
   XX:=round(M*XR); XR:=M*XR-XX;
   MM:=MM*M; XF:=XF+XX/MM
   until DR<=0; if X>=0 then
   DECIMROUND:=XF else
   DECIMROUND:=-XF
end;
procedure CHECKLIST;
begin writeln;
   writeln('ALFA ABSCISSA ORDINATE
   POLYNOMIAL ERROR');
   for i: = 0 to 81 do
   begin AL:=0.01745329252*(i);
      XABSC:=ABSCISSA; Y:=ORDINATE;
      if XABSC<=0 then Q:=0
      else Q:=exp (EXPNT*1n(XABSC));
      YAPPR:=b[NMAX]; for j:=1 to NMAX do
      YAPPR:=b[NMAX-j]+Q*YAPR;
      writeln(i:3,XABSC:14:10,Y:14:10,YAPPR:14
      :10, (YAPPR-Y):14:10)
   end
end;
```

```
begin write('degree of polynomial = ');
   readln(NMAX);
   write('places of decimals = '); readln(DCML);
   EXPNT:=DECIMROUND(EXPONENT);
   writeln('EXPONENT =', EXPNT:15:11);
   for i:=1 to 17 do
   begin AL:=0.07*(i-0.9); pn[i]:=0; pn1[i]:=1;
   write('*');
    x[i]:=exp(EXPNT*ln(ABSCISSA));
   f[i]:=ORDINATE
   end; for n:=0 to NMAX do
   begin write (n:2); p1:=0; x1:=0; f1:=0;
   for i:=1 to 17 do
    begin h:=sqr(pn1[i]); p1:=p1+h;
     f1:=f1+f[i]*pn1[i]; x1:=x1+x[i]*h
    end; a[n]:=f1/p1; b[n]:=a[n]; if n<NMAX
      then
    begin if n=0 then beta[1]:=0 else beta[n
      +1]:=pl/p;
     alfa[n+1]:=x1/p1; for i:=1 to 17 do
     begin pp:=(x[i]-alfa[n +1])*
       pnl[i]-beta[n+1]*pn[i];
       pn[i]:=pn1[i]; pn1[i]:=pp
     end; p:=p1
    end; gotoXY(1,whereY); ClrEol
   end; for j:=0 to NMAX do
   begin for k:=NMAX-j-1 downto 0 do
    begin
      b[k+j]:=b[k+j]-alfa[k+1]*b[k+j+1];
     if k+j<>NMAX-1 then
      b[k+j]:=b[k+j]-beta[k+2]*b[k+j+2]
    end; b[j]:=DECIMROUND(b[j])
   end; writeln('COEFFICIENTEN');
   for j:=0 to NMAX do writeln(j:9,b[j]:16:11);
   CHECKLIST
end.
```

*the value of the exponent, referred to as EXPONENT,
*the degree of the polynomial. (For example, try 8.)
*the places of decimals, (For example, try 10.)

The full text of the program is shown in Table 1.

The program is assumed to be used for gear trigonometric functions, expressed in the pressure angle. The 17 target points for the polynomial cover a range for the pressure angle from 0.4° to 64.6°. The polynomial may have the best accuracy near these target points, but the final check should examine the less accurate points. Therefore, the final check uses pressure angles in whole numbers, and the polynomial target points lay between them.

The polynomial program yields the values of the exponent and the coefficients to be applied in the conversion procedure. Table 2 presents an example of such a procedure or function. The function name is written with a letter K instead of the letter c in secant to emphasize that it is a function of the involute INV instead of a function of an angle.

Similarly, the cosinus function and the angle itself can be computed with conversion procedures. (See Tables 3 and 4.)

Each coefficient being determined in the polynomial program depends on previously calculated ones. To be sure that the output of coefficients of the polynomial program to be written in the conversion procedure, with a certain number of decimal places, is the key to an accurate result, the coefficients are rounded off in the polynomial program immedi-

Table 2 – Conversion Procedure SEKANS(INV)

```
function SEKANS(INV:real):real;
var Q:real;
begin if INV>0 then
   begin Q:=exp(0.6666666667*ln(INV));
     SEKANS:=1.0000000001+
       Q*( 1.0400419016 + Q*(0.3245063564+
       Q*(-0.0032156523 + Q*(-0.0088935917+
       Q*(0.0030544551 + Q*(-0.0002575881+
       Q*(-0.0001768974 + Q*(0.0000558091))))))))
   end else SEKANS:=1
end;
```

SEKANS is accurate to 1 unit
of the tenth decimal for a pressure angle up to 46°,
of the ninth decimal for a pressure angle up to 61°.

Table 3 – Conversion Procedure KOSINUS(INV)

```
Q:=exp(0.6666666667*ln(INV));
KOSINUS:=0.9999999997+
   Q*(-1.0400418300 + Q*( 0.7571779479+
   Q*(-0.4467429639 + Q*( 0.2242268843+
   Q*(-0.0972737495 + Q*( 0.0356321395+
   Q*(-0.0098523283 + Q*(0.0014817285))))))));
```

the secant function 1/KOSINUS is accurate to 1 unit
of the ninth decimal for a pressure angle up to 42°, and
of the eighth decimal for a pressure angle up to 57°.

Table 4 – Conversion Procedure ALFA(INV)

```
Q:=exp(0.3333333333*ln(INV));
ALFA:=-0.0000000278+
   Q*( 1.4422443987 + Q*(0.0001360164+
   Q*(-0.4014067126 + Q*( 0.0073800066+
   Q*( 0.0852577851 + Q*( 0.0366671471+
   Q*(-0.0498112377 + Q*( 0.0118002431))))))));
```

ALFA in radian is accurate to 4 units of the eighth
decimal (0.000002°) for a pressure angle up to 61°.

The secant function 1/cos(ALFA) is accurate to 1
unit
of the eighth decimal for a pressure angle up to 29°,
to 1 unit
of the seventh decimal for a pressure angle up to
61°.

ately after their coming into being. Special attention has to be paid to the accuracy of the exponent. If the rounding off in the conversion procedure differs from that in the polynomial program, then the loss of accuracy in the result may be as serious as unnecessary.

For application purposes the accuracy and the computing time are important features. The accuracy of the conversion procedure is very high. To achieve the same accuracy, an iteration procedure needs three steps up to about 40°, or four steps up to about 55°. Comparing the different functions in Tables 2, 3, and 4, shows that the direct calculation of the SEKANS is the most accurate one. The functions 1/KOSINUS and 1/cos(ALFA) are significantly less accurate, *in spite of the excellent accuracy of ALFA itself. The benefits* of accuracy go together with the advantage of time-saving, as is shown in Table 5.

Table 5 – Time in Milliseconds

given $1/\cos\alpha$, computed $\mathrm{inv}\alpha$	
(using Equations 1 and 2)	Time 19 ms
given $\mathrm{inv}\alpha$, computed SEKANS	
(using polynomial degree 6)	Time 33 ms
(using polynomial degree 8)	Time 36 ms
given $\mathrm{inv}\alpha$, computed $1/\cos\alpha$	
(using iteration 2 steps)	Time 90 ms
(Using iteration 3 steps)	Time 115 ms

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Acknowledgement: *Thanks to Mr. William L. Janninck for help with the technical editing of this article.*

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Back To Basics

Involute Splines

Rudolf Och
Frenco, Altdorf,
West Germany

Gears and Splines

Engineering design requires many different types of gears and splines. Although these components are rather expensive, subject to direct wear, and difficult to replace, transmissions with gears and splines are required for two very simple reasons:

1) Motors have an unfavorable (disadvantageous) relation of torque to number of revolutions.

2) Power is usually required to be transmitted along a shaft.

Due to the increasing number of motor driven components, the use of splines does not diminish, but increases. In general, there are two different kinds of tooth based systems — gears and splines.

Operation of gears

Gears always transmit torque from one axis to another. This is obtained by direct contact or indirect contact through chains or Vee-belts. Usually the number of revolutions is changed at the same time. Examples are spur gears, bevel gears, helical gears, and herringbone gears. (See Fig. 1.)

Throughout the world, gears are the subject of standards, literature, lectures, design classes, seminars, software, and specialists. However, there is very little information on splines. Therefore, from this point, we will deal only with splines.

Operation of splines

Unlike gears, splines are only applied for the transmission of torque on the same axis. Again, in general, splines are necessary for only two reasons.

1) Parts with torque transmission have to be separated due to production and assembly requirements. (Transmissions, steering components)

2) The driven part must be movable on the driving part. (Speed reducers, clutches)

The main criterion for splines is secure torque transmission. Additional requirements are little clearance, good centering, low noise, low wear, and few axial forces. These demands are very high for a part of such geometric complexity.

The requirements and designs vary depending on the kind of use. Accordingly, there are many names for these spline forms:

- Fit splines
- Straight-sided splines
- Splined shafts and hubs
- Sliding profiles
- Short splines
- Serration shafts and hubs

The designation "spline" serves as a title for all profiles of the above types which are inserted into one another, (See Fig. 2.) with the exception of the racktooth system. This system functions similarly in some respects, however, it has to be regarded separately from splines. Although it transmits torque axially, it cannot be simply inserted into the mating component, but rather requires an additional axial pressure force. (See Fig. 3.)

Splines and Forms of Flanks

The flank form of splines is not of consequence in actual operation. In practice there are only three different forms of tooth flanks between minor and major diameters.

Straight-sided

Straight-sided profiles have keys (teeth) with straight and parallel tooth flanks. (See Fig. 4.) The number of teeth varies from 4 to 12. The large tooth thickness from minor to major diameter allows the transmission of very high torques. However, there is a lack of centering efficiency in the straight-sided tooth flanks, therefore, the centering has to be on the minor and major diameters. The torsional clearance must then be increased to take eccentricity of the tooth flanks to the centering diameter, as well as the spacing errors which always exist, into account. With wear, there will quickly be an additional radial clearance and, at the beginning, little line of contact. (See Fig. 5.) There is a further disadvantage for all straight-sided splines regarding the line of contact. A surface contact will only exist on the flanks after wear or when bending forces occur.

Serration

Serration splines have straight flanks similar to straight-sided splines, however, they are angular. This angle causes a centering effect of the tooth flanks and does not require any additional diameter centering fit. (See Fig. 6.)

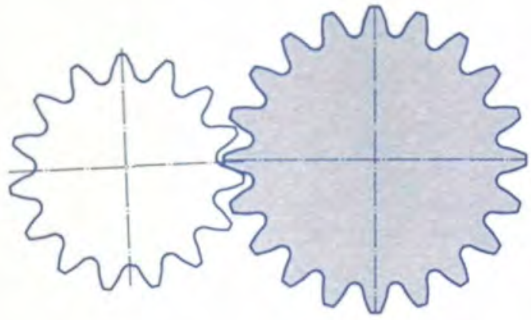


Fig. 1 – Mating spur gears.

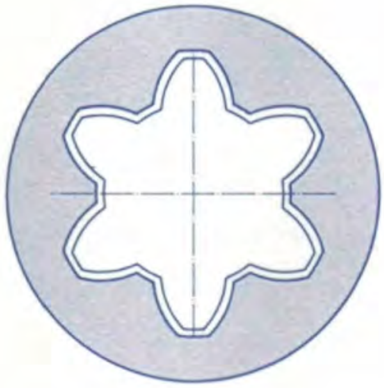


Fig. 2 – Matching splines.

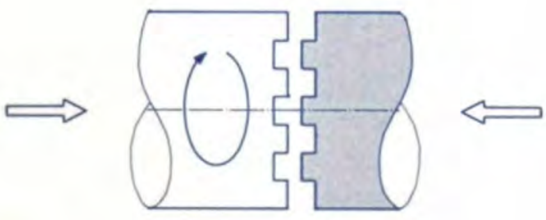


Fig. 3 – Mating rack-tooth system.

Fig. 4 – Straight-sided profile.

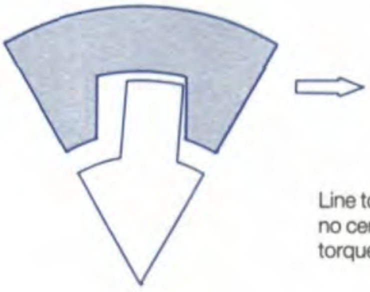


Fig. 5 – Straight-sided profile 1 tooth.

Fig. 6 – Serration.

AUTHOR:

RUDOLPH OCH *is president and owner of Frenco, a West German manufacturer of spline and profile-related gaging and workholding equipment. He is a graduate mechanical engineer and holds several patents for spline gages, arbors, and testing methods. Mr. Och serves on the ANSI spline committee.*

Favorable flank angles are between 50° and 90°. However, the teeth are rather small compared to straight-sided splines and, therefore, the transmission torques are very low. (See Fig. 7.) A second disadvantage is that line contact cannot be eliminated by serration due to the straight flanks. Therefore, serration splines are sensitive to wear and are only used for non-moveable connections.

Involute

Side fit. At present the best connection is achieved by the use of involute tooth flanks. (See Fig 8.)

The contact of tooth and space is always a surface independent of the fit clearance. This characteristic can only be obtained with the involute form.

The centering effect is very good, and the distribution of force from top of the tooth (addendum) to root of the tooth (dedendum) results from the involute curve. (See Fig. 9.) Splines with involute flanks have a very high line of contact in the nonworn condition. This reduces increase of clearance due to wear within the lifetime of the spline, compared to straight-sided splines. For these reasons the spline with involute flanks is the most frequently used connection. (See Fig. 10.)

The tooth flanks can optionally be made steeper or shallower by varying the pressure angle. Different pressure angles influence force transmission, notch effect, and producibility. Pressure angles of 30°, 37.5°, and 45° are most commonly used.

Diameter fits are possible with involute flanks for systems having great numbers of revolutions at high speeds. That necessitates more precise centering and reduced runout. In practice, these fits are rarely used. Side fit splines with involute flanks are in the majority and offer the biggest range of use.

Diameter fit. Both torque transmission and centering are done on the tooth flanks at the same time with side fit profiles. Therefore, the precision of the centering depends on the quality of production of the tooth flanks. Here certain difficulties arise, as the tooth flanks are not ground for reasons of economy. But if a very precise centering is important for operation, it is possible to produce a considerably more accurate centering using minor and major diameters. (See Fig. 11.) These are special cases which result in extra cost, yet, are cheaper to produce than ground tooth flanks. Usually a major diameter fit is chosen in these cases. The major diameter of the internal spline is broached exactly, (using a concentricity broach) and the major diameter of the external spline is ground cylindrically. This provides the most economical production of a diameter fit.

Different pressure angles. Depending on the pressure angle, the tooth flanks become steeper or shallower. The most commonly used pressure angles are 30° for sliding fits and 45° for force (interference) fit. The pressure angle of 37.5° is rarely used. (See Fig. 12.)

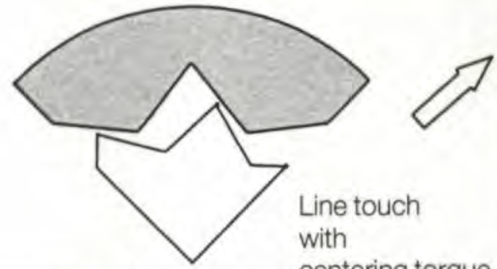


Fig. 7 – Serration profile 1 tooth.

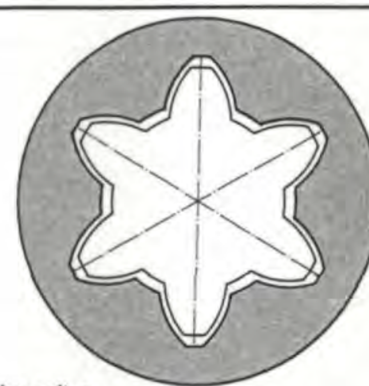


Fig. 8 – Involute spline.

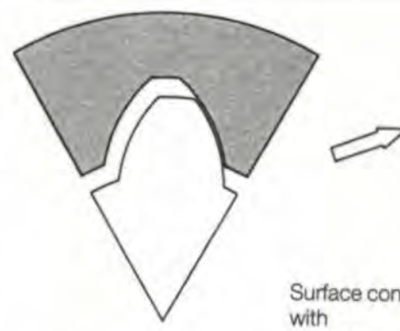


Fig. 9 – Involute spline 1 tooth.

Fig. 10 – Involute spline with load.

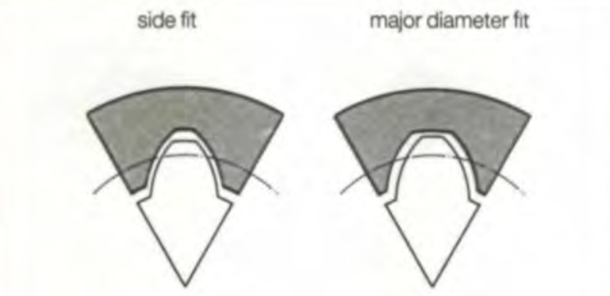


Fig. 11 – Possibilities of fits.

Fig. 12 – Pressure angles.

Pressure angle 30° is the most common profile for sliding fits. Relatively high torques are transmissible. This pressure angle is not very advantageous for the production process "rolling", due to the necessary high volume portions of deformation.

With a 45° pressure angle the centering effect is very good. There is, however, more wear due to the smaller tooth heights with sliding connections. The increased notch effect demands a (full) fillet root. This pressure angle is ideal for the production process "rolling", therefore, it is the angle of preference for force fits.

Pressure angle 37.5° is a compromise between 30° and 45°. Such profiles are often used for the advantage of a 30° spline (rigidity, stability, tightness of fit), but to avoid the disadvantages of the 30° to manufacture. Sometimes, this pressure angle is used for reasons of reducing the notch effect on thin wall mating parts.

Geometry of minor and major diameters. Splines with pressure angles of 30° commonly have flat addenda and root radii. Splines with 37.5° or 45° pressure angles are generally made with fillet roots and flat addenda because of the notch effect. Diameter fits often possess tip chamfers due to root radii arising in production of the matching parts. (See Fig. 13.)

In addition to the minor and major diameters, the form diameter at the root is always required. (See Fig. 14.)

Side Fit Profiles

Clearance of fit

Side fit profiles obtain both the centering and the torque transmission with the tooth flank contacts. Under load, the centering effect is independent of the torsional clearance of the internal spline to the external spline. However, in a no-load condition, a gap occurs between the internal and external profiles, and the resultant centering effect degenerates with direct relation to the amount of gap.

For the above reason, it is desirable to have as small a gap as possible, creating a close fit clearance between tooth and space. (See Fig. 15.) To attain this effect, close manufacturing tolerances must be maintained. In practice, however, standard production processes produce an ever increasing fit clearance over time. In special cases, a negative fit clearance in the form of an interference fit is required. In production, the amount of interference is very difficult to control and is subject to the same fluctuations as a clearance fit.

Contact area

Of all form fitting connections, splines are among the most difficult to calculate and predict. For example, a standard 1.00" spline with 24 teeth has 48 individual lines of contact. When an internal and external spline each having 24 teeth are

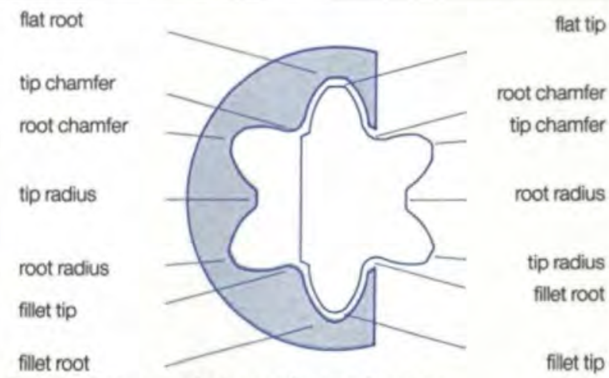


Fig. 13 – Geometry of minor and major diameters.

Fig. 14 – Diameters.

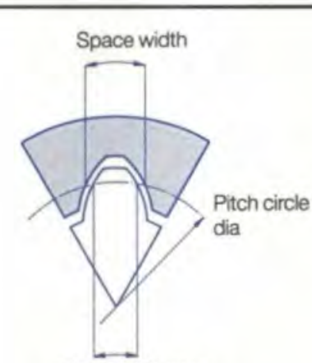


Fig. 15 – Tooth thickness space width.

inserted together, the design theory is to have any equal symmetrical fit clearance at all 24 teeth. However, inspection of the mating profile systems shows some spaces to be slightly smaller or slightly larger than others. The smallest widths of the internal spline are entirely responsible for the efficiency of the entire spline system.

The equal distribution of size and form fluctuations within both profiles directly influences the number of contacting tooth flanks under load. For clearance fit designs, this number is not important. However, on interference (force) fits, the line of contact at the tooth flanks has an enormous effect on the necessary force required during assembly. A poor line of contact influences performance of the spline as well as increases fatigue of material. As a rule it is desirable to have a good line of contact, and this can only be obtained by designing and manufacturing splines with little size and form deviations within the profile.

Effective spline

In rare cases, when an internal spline is mated with an external spline, the quality of fit may resemble a cylindrical (non-profiled) fit. In side fit profiles, this fit is achieved through perfect contact of the tooth flanks with the spaces. The same applies to the system basic sleeve – a basic shaft where an absolutely round and cylindrical bore or shaft will never be possible. Likewise, a spline will not be absolutely round or equally cylindrical over its entire length. Production is responsible for nonuniformities of form where irregularities will always exist.

Not only the size, but also the existing form errors are important for the clearance of the fit. The amount of influence of size and form to the clearance fit is different on various contours.

With regards to cylindrical (non-profiled) fits, the actual size of the components determines the fit much more than the form. Also a cylindrical form can be produced more easily and accurately.

The converse is true with splines. Splines only can be produced with relatively big deviations. The quality of fit of a round bore is always determined by the internal effective circle, and the fit quality of a shaft by the external effective circle.

Form errors reduce the effective size of bores and increase the effective size of shafts. (See Figs. 16-17.) Cylindrical fits always have form deviations, however, they are not as big as for splines. If accurate round fits are requested, they will be ground after heat treatment. The grinding of a round geometry is an acceptable and economic solution.

The cost of grinding splines is prohibitively high and is a process that is usually avoided. Even with the need for hardened structural parts, a rework usually will not follow heat treatment. At the time of production of soft (green) splined parts, big form deviations arise. Additionally, heat treatment makes the deviations of contour worse. The effective tooth thickness and space width are greatly influenced by

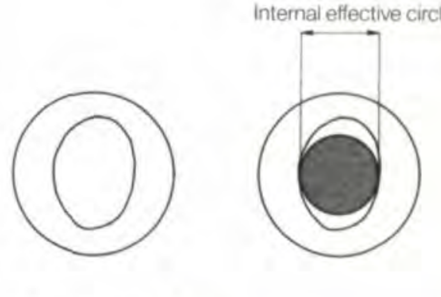


Fig. 16 – Form errors of a bore.

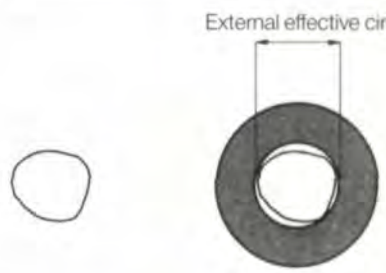


Fig. 17 – Form errors of a shaft.

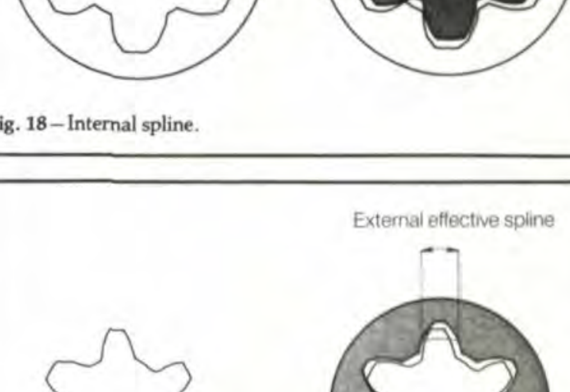


Fig. 18 – Internal spline.

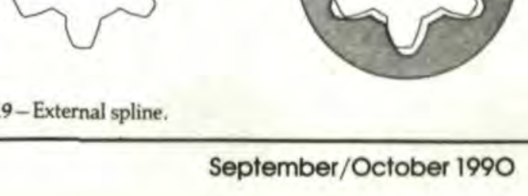


Fig. 19 – External spline.

these factors. The effective tooth thickness and space width are termed the "effective spline". (See Fig. 18-19.)

Fit system actual-effective

The actual (real) measurable size of tooth thickness and space width at the pitch circle diameter (PCD) is called "actual". The more difficult to measure size of the tooth thickness and space width which makes the effective spline, is called "effective".

The compounding effect of many form errors cause an increased effective tooth thickness on external splines. This makes the external spline appear to have a larger actual size than the mating part. The compounding form errors on internal splines result in a reduced effective space width. As above, this makes the internal appear to have reduced actual size as compared to its mating part.

The most important form errors occuring are

a) Profile error. (See Fig. 20.)
b) Spacing error. (See Fig. 21.)
c) Lead error. (See Fig. 22.)

In addition to these primary deviations, the following errors may also exist:

Concentricity error
Torsion (twist, distortion)
Damage
Eccentricity
Dirt contamination
Surface finish deviation

The summation of all the single form deviations can only be determined by fitting of an "ideal" mating part (go gage).

Unlike cylindrical fits, the manufacturing tolerance and the form tolerance are distinguished separately on splines. The manufacturing tolerance is the tolerance of the space width and tooth thickness at the circular pitch diameter. This is a required measurement for the adjustment and wear of tooling. The common designation for this specification is "actual" tolerance, and from this the size "max actual" and "min actual" are derived. (See Fig. 23.)

In addition to the actual manufacturing tolerance discussed above, splines will also have form deviations. These form errors ultimately decrease the apparent size of the spaces on internal splines and increase the apparent size of the tooth thickness on external splines. This size is called "effective". (See Fig. 24.)

Much like the deviation of size having a tolerance due to unavoidable process changes in production, deviations in form also have tolerance band governing the total amount of errors. The name of this "form deviation" tolerance band is "effective tolerance".

Internal splines have a decreasing tolerance limit called "minimum effective", which is the minimum size of the internal effective spline. External splines have an increasing effective tolerance. The limit of the external effective spline is called

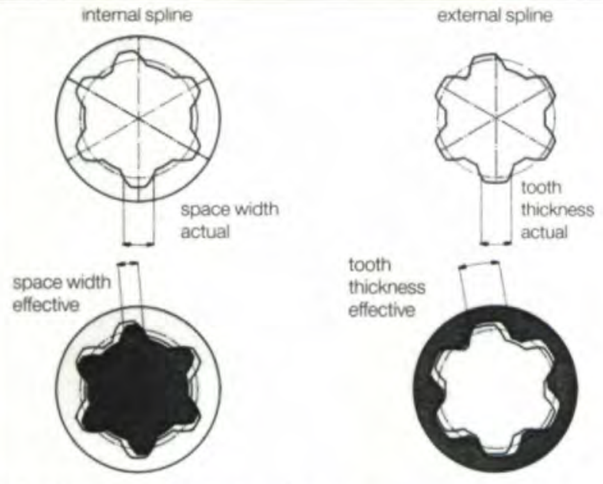


Fig. 20 – Effective result of profile errors.

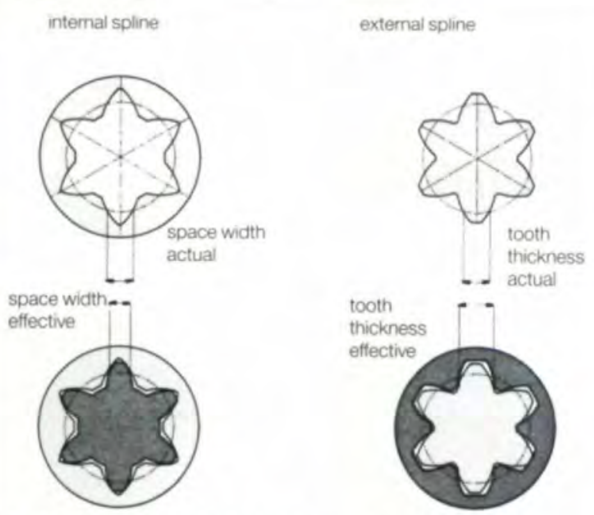


Fig. 21 – Effective result of spacing errors.

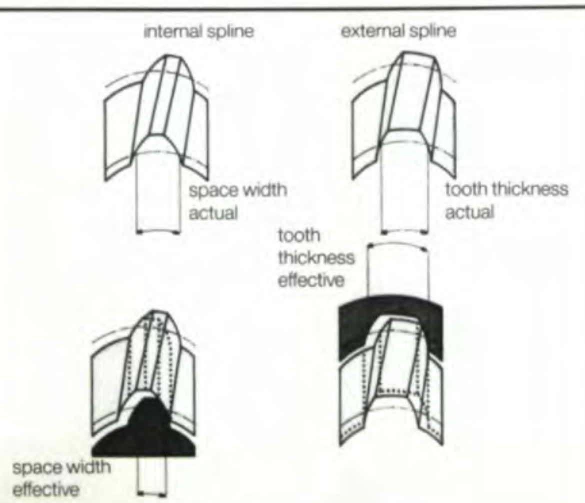


Fig. 22 – Effective results of lead errors.

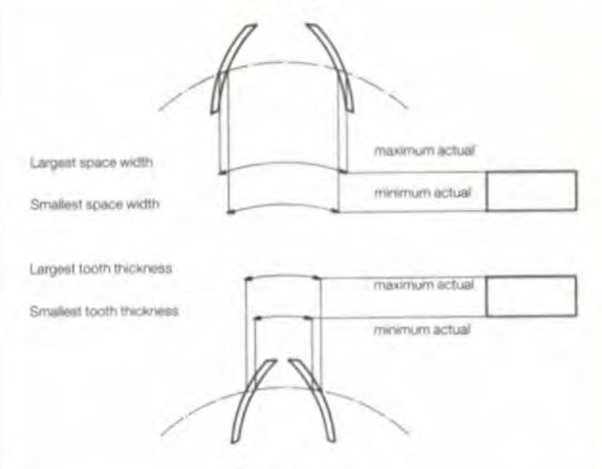


Fig. 23 – Size tolerances.

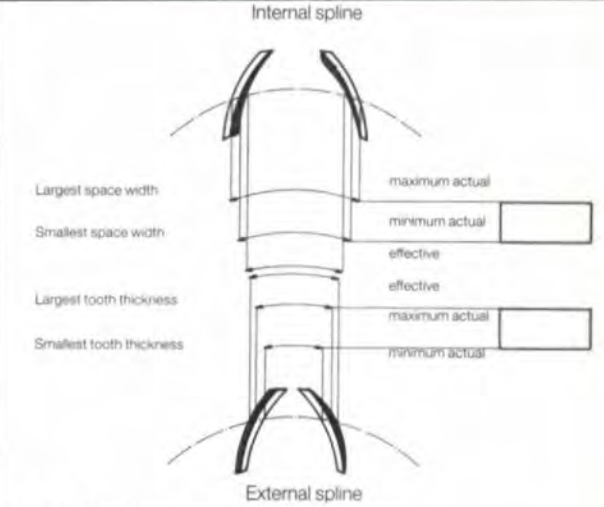


Fig. 24 – Size tolerances and effective errors.

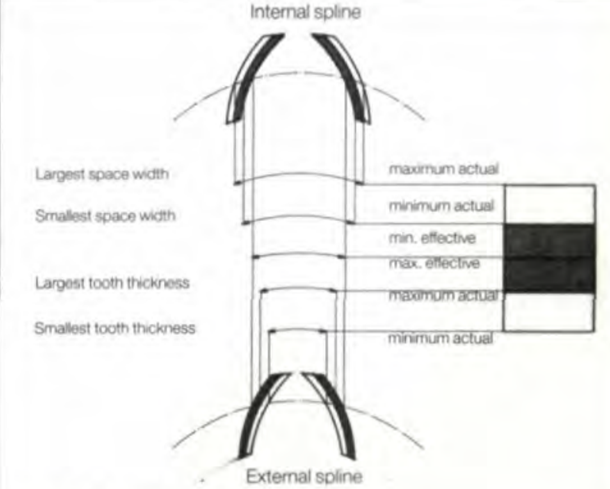


Fig. 25 – Tolerance system.

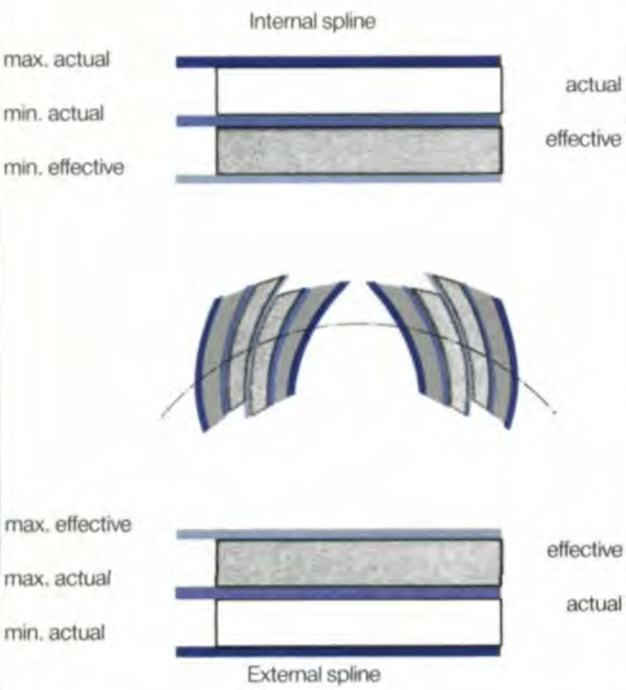


Fig. 26 – Tolerance limits.

Internal Spline	External Spline
top manufacturing limit	bottom manufacturing limit
bottom manufacturing limit (for reference only)	top manufacturing limit (for reference only)
bottom effective limit	top effective limit

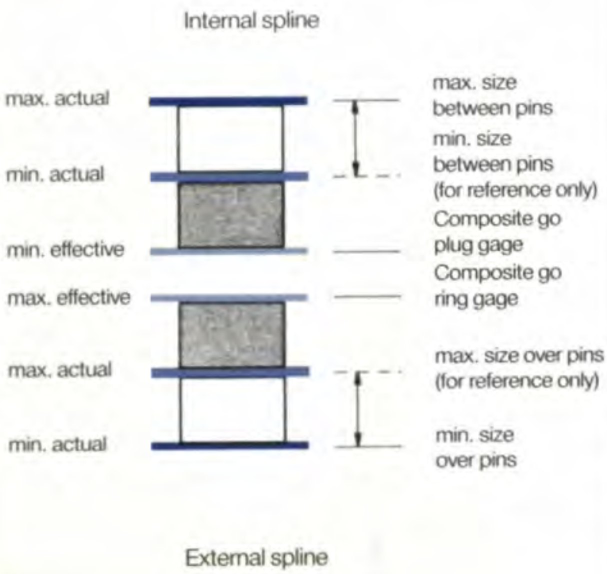


Fig. 27 – Spline tolerancing system using no-go gages.

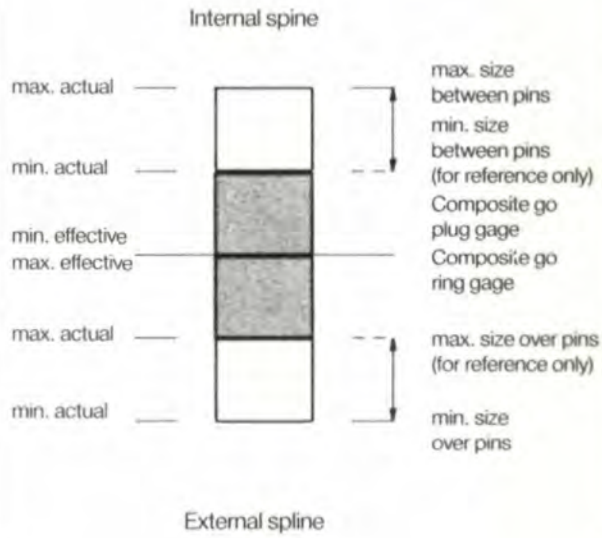


Fig. 28 – Two-tolerance zone system.

"maximum effective". (See Fig. 25.)

Fit diagrams

To show the tolerance zones (ranges) the block diagram seems to be most suitable. (See Fig 26.) The tolerance limits are as follows:

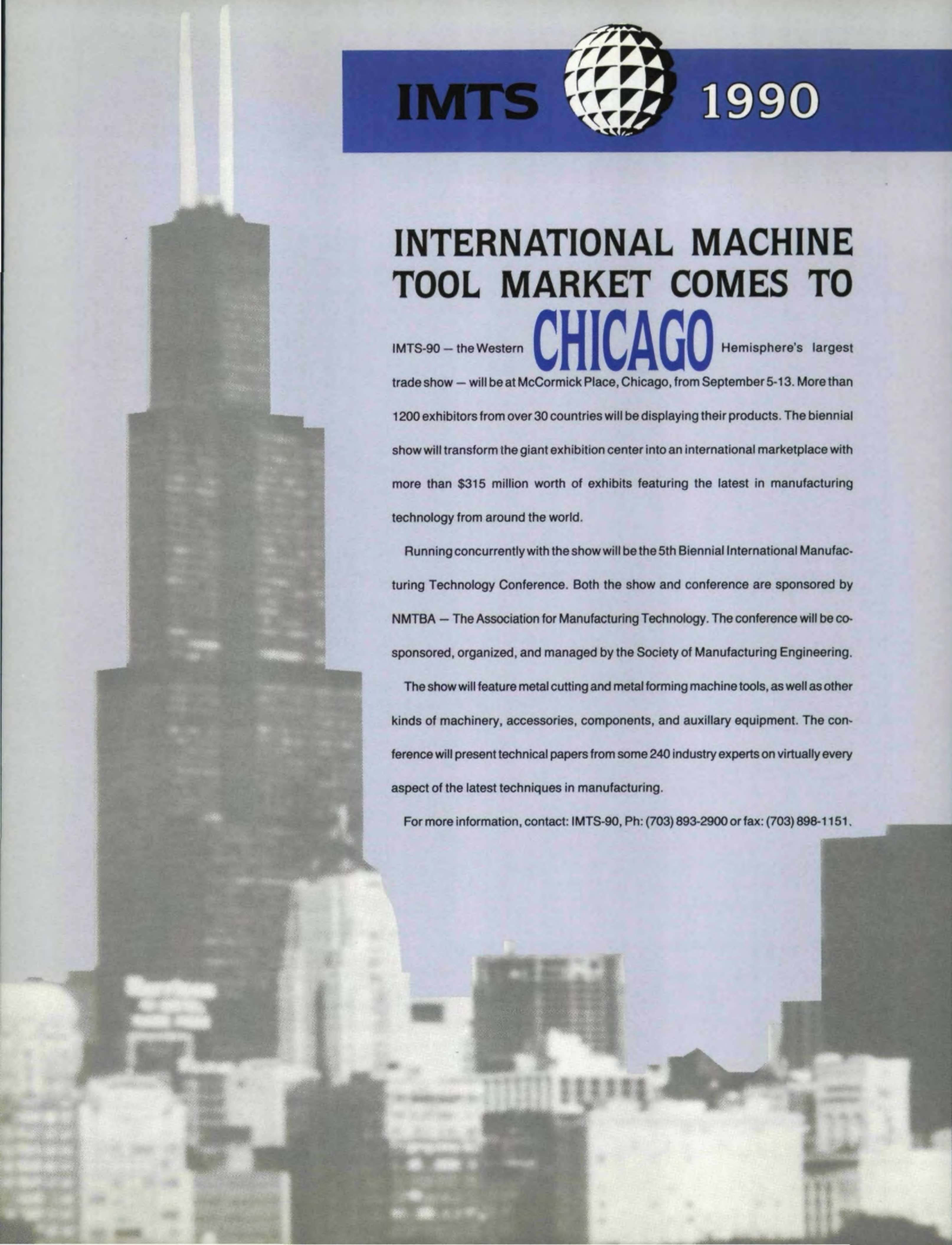
Internal spline: The red tolerance limit (maximum actual) is converted to the measurable feature "dimension *between* pins". The minimum actual limit only serves as reference for manufacture. The minimum effective clearance of the spline is checked as an attribute with a go gage plug.

External spline: To measure the red tolerance limit (minimum actual) convert it to dimension *over* pins. In this case, the limit "maximum actual" only serves as a manufacturing reference. The maximum effective spline is checked by a go ring gage.

Spline standards allow the use of sector no-go gages in place of measurement between/over pins. This method of size measurement, however, may not be 100% accurate. Size measurement must see as few form errors as possible. No-go sector gages will check profile errors as well.

The simple fit diagram is very helpful in understanding this spline tolerancing system. (See Fig. 27.)

The use of the two-tolerance zone system has never been more important than now. (See Fig. 28.) The increasing emphasis on quality and maximum material condition measurement helps us understand the need for continued use of this tolerance system in the future. □



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Running concurrently with the show will be the 5th Biennial International Manufacturing Technology Conference. Both the show and conference are sponsored by NMTBA — The Association for Manufacturing Technology. The conference will be co-sponsored, organized, and managed by the Society of Manufacturing Engineering.

The show will feature metal cutting and metal forming machine tools, as well as other kinds of machinery, accessories, components, and auxillary equipment. The conference will present technical papers from some 240 industry experts on virtually every aspect of the latest techniques in manufacturing.

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FROM THE INDUSTRY. . .

Approximating an Involute Tooth Profile

Carl F. Billhardt
Battelle Columbus Laboratories
Columbus, OH

On many occasions a reasonably approximate, but not exact, representation of an involute tooth profile is required. Applications include making drawings, especially at enlarged scale, and laser or EDM cutting of gears, molds, and dies used to produce gears. When numerical control (NC) techniques are to be used, a simple way to model an involute can make the NC programming task much easier.

Previously we had found that a second order polynomial (A*X*X + B*X + C) gave a very good fit to an involute. Out of curiosity, we thought to test how well a circular arc (a special case of a second order polynomial) fit an involute. A program, GEARFIT, was developed to do this, although the algorithm can be used without this specific program. GEARFIT does the following:

- Request data for gear parameters (e.g. pitch diameter and pressure angle) from the user.
- Generate N points on the involute between the minor and major diameters, using standard involute equations.
- For every combination of three points on the involute:
  - Find the circle which fits the three points,
  - Determine the distance from the center of the trial circle to every point on the involute,
  - Record the maximum error, relative to the radius of the arc fit to the current three points.
  - Select the circle which had the least maximum error.

The number of trial circles which GEARFIT generates is given by the expression:

NC = NP! / (3! * (NP - 3)!), where NP = number of points on involute.

This can be simplified to

NC = NP * (NP-1) * (NP-2) / 3!
NC = NP * (NP-1) * (NP-2) / 6

For 25 points on an involute, 2300 trial circles will be generated. For each trial circle, the distance to 25 points must be found to find the maximum error. If only 12 points are generated on the involute, the number of trial circles is reduced to 220. Running the program on an IBM PS/2 with a math coprocessor and measuring time by a wall clock, 18 seconds were required to solve the case with 25 points; 2 seconds were required for the 12-point case.

The best fit arc for 25 points had a maximum error of 0.0002; the 12-point case had a best fit error of 0.0003. These errors are typical for all gear configurations tried to date. This indicates the faster solution based on fewer points is close enough for most applications.

One test of GEARFIT used the following parameters (dimensions in inches):

Pitch diameter: 2.3125
Pressure angle: 25.0
Root diameter: 2.142 - 2.162
Outside diameter: 2.433 - 2.438
Arc tooth thickness: 0.0922 - 0.0952
Radius over one 0.1080 diameter pin: 1.2259 - 1.2289

The results from GEARFIT are presented in Table 1.

A CAD system was used to visually verify the results. The points on the in-

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GSF 400 CNC 6

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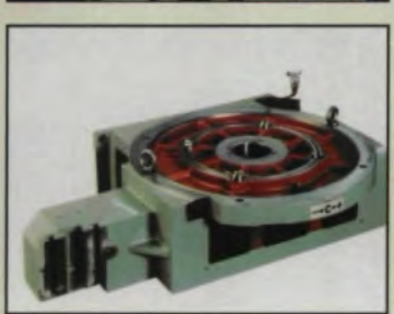

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Table 1
Results of GEARFIT

Input Data:		
Pitch diameter:	2.3125	
Pressure angle-degs.	25.0	
Base diameter:	2.0958	
Minor diameter:	2.1520	
Major diameter:	2.4350	
Tooth thickness:	0.0937	
Pin Diameter:	0.1080	
No. of Teeth:	37	
Results:		
Center of arc:	0.9525	0.4466
Radius:	0.4444	
Best fit maximum error:	0.0003	
Tooth thickness:	0.0940	
Radius over pin:	1.2276	
Points giving best fit:	2 12 23	

Points on involute and error:

1	1.0758	0.0199	−0.0002
2	1.0817	0.0215	0.0000
3	1.0875	0.0232	0.0001
4	1.0934	0.0250	0.0002
5	1.0993	0.0270	0.0002
6	1.1051	0.0290	0.0002
7	1.1109	0.0312	0.0002
8	1.1168	0.0335	0.0002
9	1.1226	0.0359	0.0002
10	1.1284	0.0384	0.0001
11	1.1342	0.0411	0.0001
12	1.1400	0.0438	0.0000
13	1.1458	0.0466	−0.0001
14	1.1516	0.0495	−0.0001
15	1.1574	0.0525	−0.0002
16	1.1631	0.0556	−0.0002
17	1.1689	0.0588	−0.0002
18	1.1746	0.0620	−0.0002
19	1.1803	0.0654	−0.0002
20	1.1860	0.0688	−0.0002
21	1.1917	0.0723	−0.0002
22	1.1974	0.0760	−0.0001
23	1.2031	0.0797	0.0000
24	1.2087	0.0834	0.0001
25	1.2144	0.0873	0.0003

volute were entered. The approximating arc was generated and used to determine the center of the 0.1080 diameter pin. No discernable error could be found between the approximating arc and the involute points, except at extreme magnification. The distance from the gear center to the outside edge of the pin was determined by the CAD system to be 1.2277. This compares to 1.2276 predicted by GEARFIT; both values are well within the allowable limits specified for the radius over one pin.

In all cases tested, the error in using an arc to represent an involute has been less than + / − 0.0005 inch. This is well within acceptable limits for preforming operations or manufacturing by laser or EDM techniques. Using an arc to represent the involute allows the tangent points of blending root and tip radii to be quickly and accurately located. Most NC controls have circular interpolation, so the involute profile can be approximated by a single statement when the center, radius, and endpoints are known. ■

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TECHNICAL CALENDAR

SEPTEMBER 4-6, 1990. Advanced Machining Technology III Conference. McCormick Place, Chicago, IL. Contact Society of Manufacturing Engineers, (313) 271-1500.

SEPTEMBER 5-13, 1990. IMTS-90, McCormick Place, Chicago, IL. Largest trade show in the Western hemisphere with exhibitors from around the world. Contact IMTS-90, Ph:(703) 893-2900.

SEPTEMBER 11-13, 1990. Short Course on Gear Noise. The Ohio State University, Columbus, OH. Seminars on gear noise and related subjects. Contact Mr. Richard D. Frasher, College of Engineering, OSU, (614) 292-8143.

October 29-31, 1990. AGMA Fall Technical Meeting. Hilton International Hotel, Toronto, Canada. The focus will be on new technology. New papers on advanced applications, stress analysis, scoring avoidance, bevel gear optimization, and wormgear design will be presented. Contact AGMA, (703) 684-0211.

NOVEMBER 28-30, 1990. Fundamentals of Gear Design. University of Wisconsin at Milwaukee. Mini-course covering basic gear systems for designers and users. Contact Richard G. Albers, Center for Continuing Engineering Education, University of Wisconsin-Milwaukee, 929 N. Sixth St., Milwaukee, WI 53202. (414) 227-3125.