Cone Drive Double Enveloping Worm Gearing Design & Manufacturing

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History of The Evolution of Double Enveloping Worm Gearing

Worm gearing is of great antiquity, going back about 2100 years to Archemides, who is generally acknowledged as its inventor. Archemides' concept used an Archemidial spiral to rotate a toothed wheel. Development of the worm gearing principle progressed along conventional lines until about 500 years ago when Leonardo DaVinci evolved the double enveloping gear concept. Worm gearing today is basically divided into three classes or types as follows: (Fig. 1)

1. Those having neither element throated.
2. Those having one (1) element throated (generally the gear) (cylindrical worm gearing).
3. Those having both elements throated (double enveloping worm gearing).

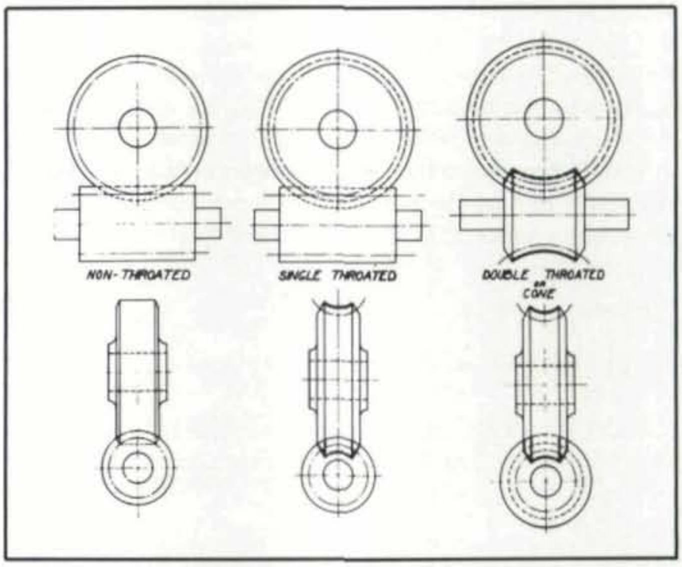

Fig. 1—Diagrams of three classes of worm gearing.

Early worm gearing was made of wood, or wood and metal. Some ancient gears made of stone have been discovered in Sweden where they were used for grinding grain. Most of the early gearing was of the non-throated design. The precise origin of the single element throated gear cannot accurately be established, although Hughes and Phillips were making single element throated gearing in this country as early as 1873.

Historical records indicate that approximately 200 years ago in York, England the famous clockmaker, Henry Hindley, made the first throated worm design. It was used in a dividing machine which he is also credited with originating. The gear was approximately 13" in diameter and had 360 teeth. The teeth were about 1/16" thick at the pitch line, and the helix angle was about 1°. John Smeatson, a contemporary, said in part, "The threads of this screw were not formed upon a cylindrical surface, but upon a solid, whose sides were terminated by arches of circles — the screw and wheel, being ground together as an optic glass to its tool, produced that degree of smoothness in its motion that I observed, and lastly, that the wheel was cut from the dividing plate". It can be determined, from the above, that the worm was throated, but it is not clear whether the wheel was also throated. (Fig. 2)

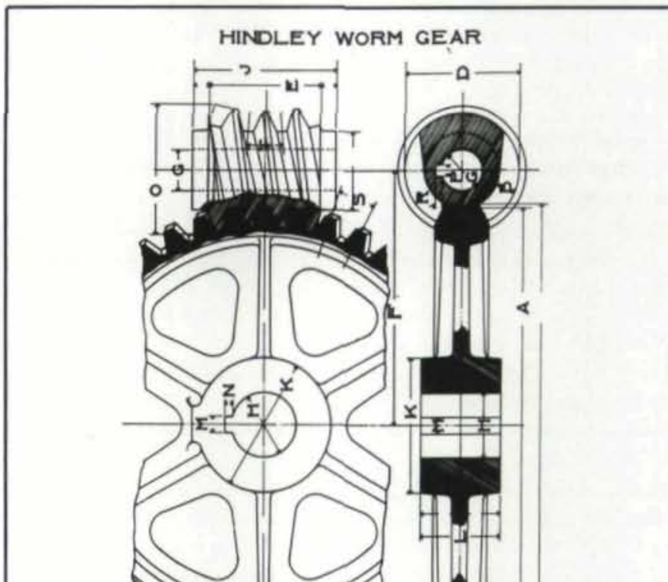

Fig. 2—Typical Hindley worm gear design.

There is no indication of further progress in throated worms

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until 1878 when Stephen A. Morse became interested in a patent on a machine for cutting them. About 1883 the U.S. Government became interested in the Hindley type gearing, and began using it where heavy shock loads were encountered and where an absence of backlash was desirable.

In general, the Hindley double-enveloping worm gearing utilized in this country incorporated an enveloping worm having straight sides in the axis of the worm. Also, the worm was generally bronze and the gear steel. It early became apparent that as a result of the constantly varying diameters and helix angles of the Hindley double enveloping worm, throughout its length, that it would be necessary to utilize a fluted worm as a hob in the production of the throated gear element. The form of the worm (and the fluted hob) prevented the use of tangential hob feed because the larger diameter of the hob at its ends (compared to the center diameter of that hob) would effectively destroy the gear form and gear teeth diameter in the hobbing process. This left the radical feed method as the most viable method for producing Hindley double enveloping worm gearing. In this method, the hob and gear were fed toward one another in a radial direction while geared to the proper time relation for the ratio involved.

For the same reason that the completed elements will not operate satisfactorily on other than the designed center distance, with Hindley designs it was equally impossible to hob a *true* form by radially feeding the hob and gear blank together. This radial feed method resulted in a "destroyed action", whereby, the ends of the hob, because of their larger diameter and rotational arc, removed excessive stock from the flanks of the gear teeth in the infeed process. (Fig. 3)

To minimize this problem, Hindley hobs were made very short, so as to reduce the cutting arc of the hob ends. The negative effect of this was to reduce the effective teeth in contact, since the worm could not be longer than the hob without creating worm/gear interference. After hobbing, Hindley double enveloping worms and gears were extensively lapped in an attempt to broaden the worm gear contact. Such lapping sometimes exceeded 48 hours — using sand and water.

In the early 1920's Mr. Samuel I. Cone of Portsmouth, Virginia, manufactured at the Norfolk Navy Yard, a double throated or double enveloping type of worm geaing, which presently carries his name, i.e., Cone Drive double enveloping

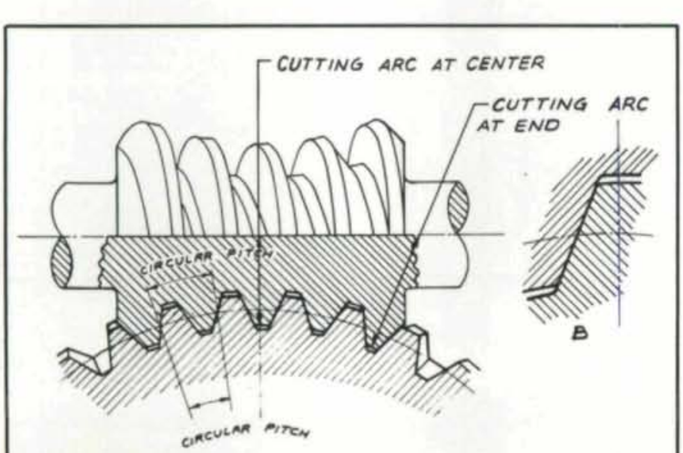

Fig. 3—Cutting arc resulting in "destroyed action".

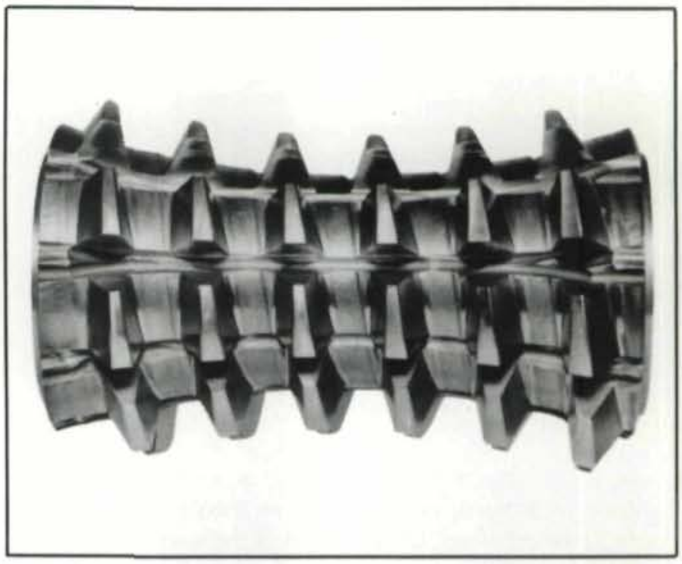

Fig. 4—Typical double enveloping gear hob.

worm gearing. Mr. Cone had developed and patented a rational method of generating the elements of double enveloping gearing which permitted the cutting of both elements, without interference, when operated at center distance. The Cone double enveloping principle utilized a hob made by gashing a worm (Fig. 4) but having thread flanks thinner in cross section than the worm from which it was evolved. When such a hob was radially fed into a gear blank to proper center distance, the gear was merely roughed out, in that its tooth form was oversize to the extent that the hob was made undersize. The "destroyed action" was still there. However, the hob and gear were now on center distance and there was stock available for truing up the gear tooth form. Radial feed here was not the answer because the hob and gear were *already* on center distance. Tangential feed would not work because it would alter the pressure angle and tooth form of the gear tooth. In the Cone principle, rotational feed is used, whereby the relative rotational position of the hob and the gear blank are changed. This creates

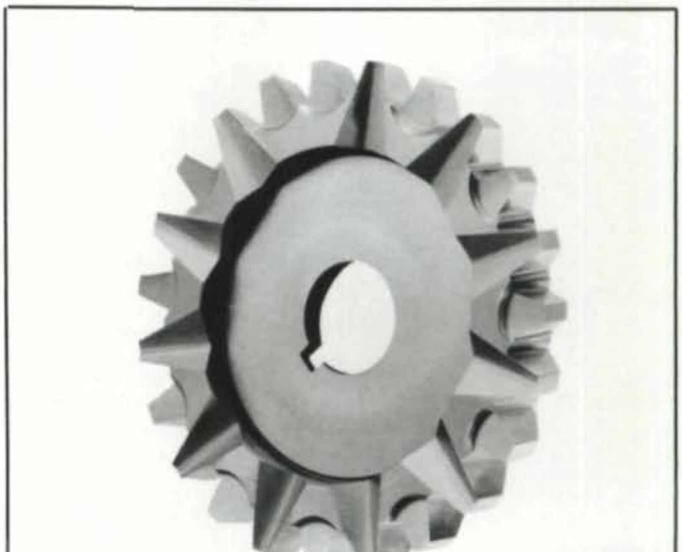

Fig. 5—Typical double enveloping worm cutter.

a cutting action to bring the geartooth to size, remove the "destroyed action", and produces a true tooth form. This rotational feed is not unlike taking up backlash in the gearset.

The double enveloping worm is produced in a manner similar to the double enveloping gear. A cutter representing the midplane section of a gear is used. (Fig. 5)

The teeth in the cutter, insofar as mating form is concerned, are identical with the teeth in the gear. (While this is basically true — certain liberties are taken in both hob and cutter design to provide better cutting action, chip clearance, uniform cutting pressures, etc.). Here also, the cutter teeth are undersize, but true in form to permit recutting of the thread flanks once correct center distance has been reached. Rotational side feed is again used in this finishing operation.

While the above hobbing procedure seems simple, there are many factors adding to the complexities of achieving economical manufacture. Hob and cutter heat treat distortion, true form backoff, generating and hobbing large gearsets up to 50" center distance, indexing and non-indexing ratios, special hobbing and generating machines with rotational side feed features, individual hobs and cutters for each center distance and ratio were only a few of the problems which had to be solved by Cone Drive engineers to effectively amd economically manufacture double enveloping gearsets. Double enveloping worm gearing is manufactured in this country primarily by Cone Drive, Franke Gear, Western Gear and Vard.

Design of Double Enveloping Worm Gearing

The design of Cone double enveloping worm geaing is based on a different tooth form concept. Instead of involute or other curved tooth forms, most double enveloping gears have straight sided forms on both the gear teeth and worm threads with this form tangent to a common base circle. (Fig. 6)

As with cylindrical worm gearing, experience over the years has established the practical design proportions for helix angles, pressure angles, number of teeth in gear, gear widths, relationship of worm P.D. to worm root diameter, tooth thickness, backlash, etc.. Most of these are outlined in AGMA specification 342.02, which contains design formulas as well as tables of recommended proportions. While the average engineer can use these formulas to design his own gearsets, double enveloping manufacturers prefer to establish these designs themselves, in most cases to specifically suit the application. Since a hob and cutter or other suitable tools must be made for each center distance and ratio, it is desirable that a gearset design be of the proper proportions the first time through, because changes to the design proportions cannot be readily accomplished without changing the tooling.

The straight sided form, as well as the side feeding operation in manufacturing, enable double enveloping worm gears to have variable tooth thickness. This gives considerable latitude in design. Normally we hold to a 55%-45% ratio with the gear 55% of the circular pitch and the worm 45% of the circular pitch. This gives a much more balanced design since the worm, which is made from steel, is the stronger member (120,000 PSI yield) and the gear, made from bronze, is the weaker member (25,000 PSI yield). By making the gear tooth thickness greater than the worm thread thickness, the two members are more nearly equal in relative strength. Obviously, if we can normally

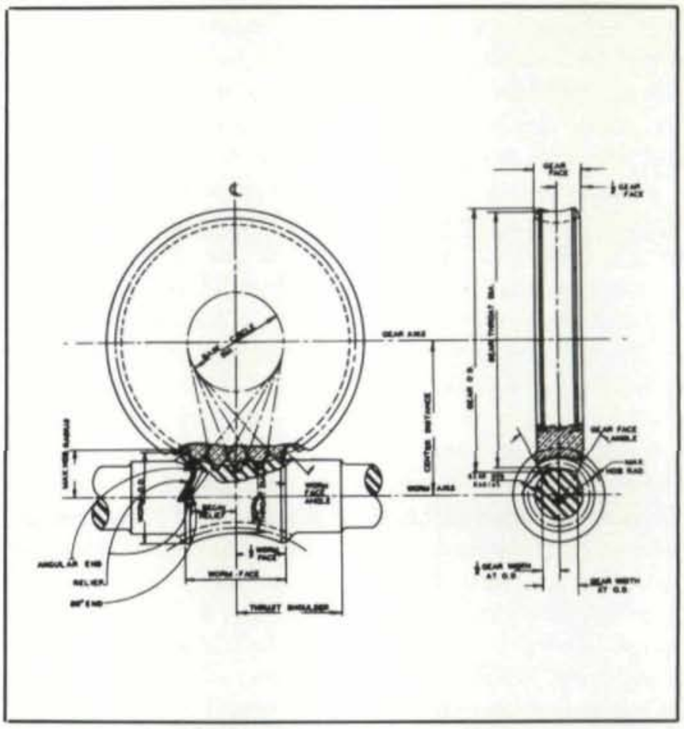

Fig. 6—Cone drive double enveloping worm gear design.

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obtain a 45/55 tooth thickness relationship, then specific applications of an unusual nature which would benefit from a 60/40 or other worm/gear thickness relationship can be provided.

Another advantage to the side feeding feature of Cone double enveloping worm and gear manufacture is selective backlash. Since we can produce worm and/or gear tooth thickness exactly to suit, we can readily create close backlash gearsets. We have produced numerous designs and manufactured hundreds of gearsets with backlash in the .000 to .0002 range on design center distance. Obviously, it is extra work and more expensive to make such close backlash designs — but it can be and is done.

Gear material for double enveloping worm gearing follows the normal worm gearing practice of using SAE 65 or 65N bronzes-statically chill cast or centrifugally cast. Forged bronzes and some of the maganese aluminum bronzes are sometimes used for gear materials. They are primarily noted for their strength and not their bearing characteristics. Aluminum bronzes are often used in place of regular tin or nickel/tin bronzes where additional strength is required, not, however, without a certain penalty in bearing characteristics. Most aluminum bronzes must be restricted in rubbing velocity to work effectively in worm gear applications. Whereas we use tin bronzes up to 2000 ft/minute rubbing speed with splash lubrication, the aluminum bronzes are restricted to 600-800 ft/minute.

Gear bronzes are a unique material and one should not assume that just any tin-bronze alloy will suffice. It takes a precise blend of tin and copper in the right proportions and cast in a precise manner to create the proper dendritic formation and the correct amounts of the alpha/delta phase so necessary to make a *good* bearing bronze for worm gearing. Much of the secret in obtaining an effective gear bronze results in the structural formation of the material with hard load carrying phases of a high tin concentration finely dispersed throughout a matrix of bronze which tends to cushion the tin particles. (Fig. 7) (Fig. 8)

This provides an effective bearing surface to carry the load. The copper phase should be sufficiently ductile so that it will yield and flow with that load. This "flow-ability" assists in creating the broad area of contact so prevalent in double enveloping worm gearing and also compensates for minor errors in manufacturing and assembly.

Most gear bronzes have physical characteristics of 45,000-50,000 PSI tensile, 22,000-25,000 PSI yield, 10%-12% elongation and hardness ranges of 85-120 BHN. For a comprehensive listing of gear bronzes and their physical characteristics, reference should be made to AGMA 240.01.

Fig. 7—Dendritic structure in alpha matrix tin bronze centrifugally cast-50x.

Fig. 8—Delta particles in dendritic alpha matrix tin bronze centrifugally cast-50x.

Manufacturing Methods For Cone Drive Double Enveloping Worm Gearing

Manufacturing of double enveloping worm gearing beyond normal preparatory stages involves basically:

1. Hobbing
2. Generating
3. Matching and lapping
4. Assembly

Hobbing and generating is done on the same machine. On gearsets up through 18.000" center distance, there are specially designed hobber/generators which mount the work piece and the cutting tool in exact position and location with respect to each other. (Fig. 9)

Center distance is set by dial indicators, side and end positions are set by the use of gage blocks. The radial feed mechanism and rotational feed mechanism are geared into the machine so that the hobber/generator is in effect semi-automatic in operation. Gearsets larger than 18.000" center distance up

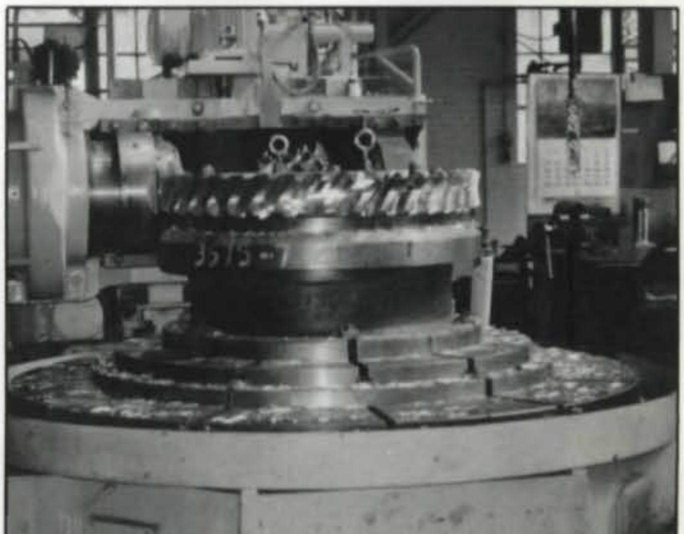

Fig. 9—Double enveloping worm-gear-hobber-generator.

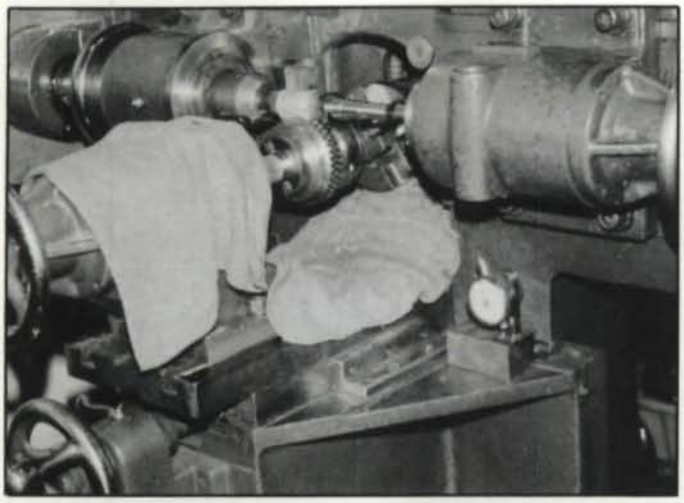

Fig. 10—Matching lapping machine.

to 52.000" center distance are manufactured on commercial hobbers which have been considerably altered to suit double enveloping gearing and have rotational feed boxes attached to the machine system.

Hobs and cutters are manufactured at our own facility where material, heat treat, back off, grinding, spacing, lead, pressure angle, etc., can be controlled within exacting limits.

Cone double enveloping worms are not ground. This precludes certain material selections and hardness. Cone double enveloping worms are made from 4150 resulphurized steel heat treated to Rc 35-38 or nitrided with a 87-15N case (28-30 Rc core). Generating steel this hard has an obvious effect on cutter life. Hob and cutter life controls the processing after hobbing and generating. Tooth form will change as tools dull. As a result, within the usable life of a sharpening cycle, some tooth forms will be minutely different than others.

To provide the customer with a uniform and consistent quality gearset, most double enveloping worms and gears are matched on center distance, end position, and side position for a quality check. (Fig. 10)

While this is not a mandatory procedure, in that many customers purchase worms and gears separately and use them at random, it is, nevertheless, the ideal procedure to achieve the high performance capabilities of the double enveloping product. When a worm and gear do not produce contact patterns up to standards, the worm and gear are lightly lapped to produce the desired contact. Such a lapping procedure is very minimal—no more than 1 - 2in-lbs. load and 3 - 8 gear revolutions in each direction of rotation. Total lapping time is generally less than one (1) minute.

Assembly of double enveloping worm-gears, to *ideal* contact, necessitates that the set be on true gear side position, worm end position, and center distance. We recommend, for *maximum initial performance* of a double enveloping worm gearset, that center distance, gear side position and worm end position tolerances be in accordance with the following table.

Center Distance	Tolerance
Up to 6"	±.001
6" to 12"	±.002
Over 12"	±.003

Comments have been made about the additional effort involved in controlling worm end position. If you provide control of center distance and gear side position, and both cylindrical and double enveloping worm gearing requires that you do so, then controlling worm end position is only one more step, and certainly the technique is readily available. We are consistently holding worm end position along with center distance and side position while producing 2000, and more, speed reducers per month at our facility. The effort in providing worm end position is minimal when compared to the benefits of the greatly increased load capacity which will be realized with double enveloping worm gearing. In addition, we have found that double enveloping worms and gears — off slightly on center distance, worm end position, or gear side position (or any combination) rapidly seat against each other during the break-in and are soon providing full contact and design capacity. The straight sided conjugate form, parallel to a common base diameter, enables the double enveloping design of worm gearing to regenerate to an ideal matching relationship.

Worm gear efficiency is worthy of discussion. Testing has established that center distance for center distance and ratio for ratio cylindrical worm gearing and double enveloping gearing will have the same basic efficiency values. Since Cone Drive manufactures not only worm gearing but helical and herringbone gearing as well, we feel that we are also somewhat qualified to compare worm gear efficiencies with helical efficiencies. That worm gear efficiency is less than that of helical gearing will not be disputed. However, the overall variations between these two (2) types of gear efficiencies are not as great as generally assumed. For years it has been commonly stated that helical and herringbone gearing have efficiency losses of 1%-2% at mesh. This is a reasonably valid value. However, what most people fail to consider is the fact that helical and herringbone gearing at 1%-2% mesh inefficiency must be installed in gearboxes where bearing losses, oil seal drag and churning losses within the bearbox add to the mesh loss. When these losses are added to mesh loss, then the overall gearbox efficiency is considerably less.

We consider that a single reduction helical or herringbone gearbox will have approximately 95% overall efficiency, a double reduction will have 92%-93%, a triple reduction will have 89%-90% and a quadruple reduction will have approximately 86%-87%. It should also be recognized that the higher ratio helical and herringbone gearboxes require multiple gearsets and each one of these gearsets must be suitably mounted in its own bearing mounting arrangement. The multiplicity of bearings and gears compound the bearing and churning losses within the gearbox.

A worm gearbox within the normal ratio range of 5:1-100:1 generally accomplishes the ratio change using a single gearset. The churning losses will be higher on a 5:1 design but the churning losses on the 100:1 design will be substantially reduced because of the very low rotational speed of the gear. We consistently find 5:1 ratio worm gearboxes running at 95% efficiency, which compares on a par with helical boxes of this ratio. With a 20:1 ratio reducer, the helical box will have either two

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(2) reductions at approximately 92%-93% overall efficiency or three (3) reductions at about 89%-90% efficiency. The worm gearbox with a 20:1 ratio will have about 85%-87% efficiency. A 30:1 ratio helical reducer will generally require three (3) meshes with approximately 89%-90% efficiency. The 30:1 wormgear speed reducer will have an efficiency of approximately 83%-84%. You can see the helical box *is* more efficient, but certainly not to the degree often claimed.

There are other inherent advantages in worm gearing which must be considered in evaluating the application and the type of gearing intended for that application. Double enveloping worm gearing will take a momentary overload of 300%, whereas helical gearboxes are only designed for 200%, momentary overload. Helical gearboxes restrict motor starting capacity to 200%, whereas double enveloping worm gearboxes permit 300%. Generally speaking, worm gearboxes are smaller in overall size and weight, and in terms of horsepower capacity, generally less expensive. In addition, with compactness of the double enveloping wormgear principle, double enveloping gearboxes are more compact and weigh less, horsepower for horsepower, than cylindrical gear reducers.

This paper was published for the National Conference on Power Transmissions 1979 and reprinted in "Technical Aspects of Double Enveloping Worm Gears, a Cone Drive Publication.

Design of the Involute . . .
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generally supposed. In other words, bearing pressures are not greatly affected by an increase in the pressure within the usual limits. This condition is graphically presented in Fig. 14. To construct this diagram, draw a line *A B* at right angles to the line of centers and tangent to both pitch circles. Then draw a line *C D* tangent to the base circles and passing through the pitch point *E*; this line representing the pressure angle. Now drop a perpendicular at any point *G* on line *A B*, passing through line *C D* at point *F*. With *E* as a center and *E F* as a radius scribe an arc. Increases in the load on the supporting bearings due to changes in pressure angle can be determined graphically by noting the changes in distance *H*, as the pressure angle changes. It is apparent that the load-increase is the ratio of lengths *E G* to *E F*, and is, therefore, proportional to the secant of the pressure angle.

The second column in Table II gives the secants of various pressure angles listed in the first column, and ranging from 14½ up to and including 30 degrees.

The last column lists in terms of percentage, the increase in the load as compared with 14½ degrees. It will be noticed that an increase in the pressure angle from 14½ to 20 degrees, results in an increased load on the supporting bearings of only 3 percent.

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Scoring Load Capacity . . .
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Conclusion

A new method for scoring load capacity rating, based on the calculation of a mean, weighted flank temperature, the integral temperature, has been described. The limiting temperatures necessary, for the definition of a scoring safety factor, can be obtained from any available gear oil test. The method is valid for all types of oils as straight mineral, mild and EP-oils, as well as, synthetic oils where gear scoring tests are available. The method was checked with more than 300 scoring tests on test rigs and more than 100 practical gears with and without scoring damages. A good correlation was found for the Integral Temperature Criterion, and it was obviously superior to the Total Temperature Method, as well as, to the Scoring Index Method.

The method has been modified for bevel and hypoid gears(10) and even in this field of application a good correlation between calculated scoring factors and field experience was achieved.

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