Throughout this table it is assumed that only one tooth is in contact at one time, and that bearing takes place across the full face of the gears. The valves of f vary as the square root of the speed of the pitch circle, since the greater this speed the greater the vibration and shock, as is shown by Lasche's deductions.

The author does not propose to go more fully into the considerations which govern the design of spur gearing, but interesting and complete data and formulae will be found in Lathe Design, by Nicolson & Smith, and in Halsey's book on Machine Design.

It may be mentioned, however, that it is generally agreed that the Lewis formula gives proportions which are excessively strong for low revolutions, and revised formulas for these conditions will be found in both the text books above referred to.

In the design of Laminated gearing the manufacturers use the Lewis formula, but owing to the greater number of teeth in contact reduce the face value arrived at by this formula by 20 per cent.

Several different systems are in use for generating straight cut spur gearing, both by shaping or planing and by hobbing.

The best known shaping systems are the Sunderland, using a cutter taking the form of a straight sided rack, which is the basis of the involute system, and the Fellowes system, using a circular cutter as a planing tool, the shape of the cutter teeth being originally generated from a rack, and both can be accurately ground to correct form.

The accompanying diagrams illustrate the generation of an involute tooth by the Sunderland process.

In the hobbing process the teeth are generated by a revolving hob, the teeth of which are cut to a plain rack section. One hob will cut gears of any number of teeth of one pitch, which in itself is a great advantage over the milling process and its accompanying range of cutters, each of which is only correct for one particular number of teeth of one pitch, and approximately so only for the other numbers of its marked range. For various reasons, however, which are beyond the scope of this paper, the hobbing process cannot be considered to be as satisfactory as the shaping processes.

Double Helical Gearing.—As previously mentioned, the only form of gearing which has proved successful up to the present for transmitting large powers at high rates of speed has been the double helical. To understand the reasons for this, it is necessary to make a comparison of the actions which take place between two teeth in contact in streight and in helical gearing.

In straight gearing we first have the point of a tooth engaging the root of its mate, next the teeth engaged near the pitch line, and finally the root of the tooth engaging with the point of its mate. At the beginning and end of the action one tooth is subject to the maximum sliding action, and only during the middle of the motion does it approach the ideal rolling action. As a consequence of this imperfect action, wear takes place at the root and points of the teeth, and the involute curve is destroyed, as previously demonstrated.

In the helical gear all the phases of engagement above mentioned take place simultaneously—that is to say, the load is partly taken by surfaces which are in rolling contact, and partly by surfaces which are in sliding contact.

As in straight gear there is the same tendency to wear away at the root and points of the teeth, but as the pitch line portion is always carrying portion of the load, the effect of wear at the ends of the teeth tends to concentrate more load on the pitch line portion of the teeth.

The teeth, therefore, tend to wear more evenly and to retain their original involute form, as the least wear of the ends of the teeth will relieve them of all load.

As the teeth retain their involute form motion is transmitted without shock or vibration, and there is no tendency of the gears to become more noisy after lengthy service.

In addition to the above advantage the second feature is the continuity of action which takes place. In a straight gear the number of teeth in simultaneous action is strictly limited, and is definitely fixed by the number of teeth and pressure angle of the gears. In a straight gear of 30 teeth or less with a 20 deg. pressure angle at a particular instant there will only be one tooth in contact with the whole load concentrated upon it. If this gear wheel be designed, however, as a double helical wheel of 22½ deg. spiral angle, then no less than four different teeth will be in simultaneous engagement. This fact results in the load being transferred from tooth to tooth more gradually, and with less shock. It also makes it possible to reduce the number of teeth in the pinion, as ample contact is obtained by merely increasing the face width to meet the conditions.

Ordinary double helical gearing is produced by several methods, viz.: Shaping, by hobbing, and by the use of end milling cutters.

The latest method is the shaping process under the Sunderland patents. The cutters in this process are again straight sided racks as for shaping straight teeth, but two cutters are employed, one for each side of the helix.

They are carried in slides, and the general appearance of the machine and cutters is shown by the accompanying slides.

The feature of gears cut by this process is that the bearing takes place across the whole width of the wheels, as will be shown by the small sample exhibited. The shape of the teeth being generated, they are extremely accurate.

The hobbing process can only be used when a groove is left in the centre of the face of the wheel or the teeth are staggered, as in the Wuest process. The shape of the teeth is also generated in this method of cutting.

The end milling process has been chiefly developed by the Citroen Gear Company, and chiefly depends for its accuracy on the care taken in the grinding of the cutter. Except with the greatest of care, this process must be considered inferior to the first-mentioned owing to the smallness of the tool, its small wearing surface and capacity for dissipating heat. In gearing cut by the end-mill process contact does not take place at the centre of the teeth as in wheels which are shaped by the Sunderland process.

Ordinary double helical gearing, as distinguished from turbine gearing, is being largely adopted for many purposes, and the following slides show examples of same. The first consists of one of four ammonia compressor drives, 250 horse-power being transmitted in each case, the reduction being from 485 r.p.m. to 60 r.p.m. In this particular instance the wheel face is 12 inches, the pinion having 17 teeth of 1¾in. pitch. The wheel is of cast steel, and the pinion of high carbon steel. There will shortly be added to this plant three additional compressors, each gear driven and requiring 500 h.p. The gears in this instance will be 17 inches wide, the pinion having 19 teeth of 2¼in. circular pitch. The reduction will be from 375 r.p.m. to 60 r.p.m.

The rolling mill installation consists of double reduction gearing for driving one set of 24in. cold rolls running at 11 r.p.m., and one set of 30in. rolls running at 8 r.p.m. from one motor running at 360 r.p.m. A six-ton flywheel is fitted to the motor shaft, and all the pedestal bearings are mounted on a heavy cast-iron bed, each set of gears being enclosed in a steel case and run in oil.

There is also shown another reversing mill reduction gear and motor, designed to transmit a maximum of 1500 h.p., and reducing the motor speed of 188 r.p.m. to 75 r.p.m. The motor in this instance is a variable speed direct current machine.

Similar equipments to the latter are also used for colliery fan work.

The large pair of pinions for a 36in. cogging mill shown weigh 20 tons per pair. For rolling mill work double helical gearing offers special advantages. The principle of these is that the even transmission and entire absence of vibration allows the finishing rolls to be gear driven for the finest work without showing gear marks on the product. The absence of shock in transmission makes breakages much less frequent than with straight cut spur gears.

Attention is directed to the very solid manner in which all the gears are supported, and it is absolutely necessary for satisfactory service that the shafts be generously proportioned to prevent vibration, and that the pinions be not subject to end pressures which would tend to cause one side only of the gears to take the load.

The problem of securing silent running in double helical turbine gears is a much more difficult one, in view of the fact that pitch line speeds of 6000ft. per minute are quite common. In the majority of instances these gears are hobbed, and great precautions are taken to secure accuracy. Sir Charles Parsons patented the creep system with a view to eliminating the errors due to the parent gear of the cutting machine. Briefly this consists of mounting the gear to be cut on a table made up in two halves, the upper one carrying the gear being rotated slightly faster than is necessary, and the lower one somewhat slower, the resulting relative motion reducing the maximum errors in gears cut by this process to about 1-5th of those normally occurring.

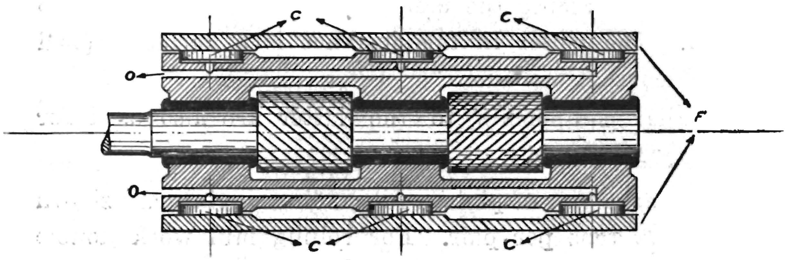

Fig. 7.

F = Fixed Frame. O = Oil Ways. C = Pistons.

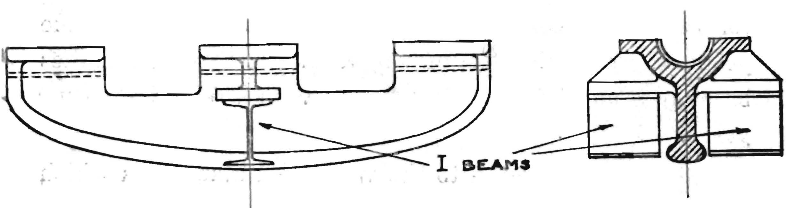

Fig. 8.

In the majority of instances, and especially with big ratios of reduction, a third bearing is provided for the pinion shaft, but some makers do not provide this in gears having a ratio of less than 10 to 1. An excellent example of two bearing mounting designed to transmit 1800 h.p. with momentary overloads of 50 per cent. is shown on the next slide. The ratio of reduction in this case is 7½ to 1, viz., from 3000 r.p.m. to 400 r.p.m.

Turbine gears provided with floating pinion bearings are of two types, one in which the frame carrying the pinion bearings is hydraulically supported, and the other in which the floating action desired is obtained by the flexure of some part of the frame supports.

Fig. No. 7 shows the first and Fig. No. 8 the second. The first is most commonly known as the Westinghouse system, and the second is known as the Melville and Maccalpine systems, after the inventors. They are both designed with a view to permitting the alignment and position of the pinion shaft to be controlled entirely by the acton of the teeth in contact, and not by the fit or alignment of the bearings. Variations of tooth pressure due to errors in cutting are provided for, and excess pressure at any one point is relieved and transferred to the point on the working face of the pinion, where the tooth pressure is for the instant below normal.

This is accomplished in the first instance by the excess bearing pressure on one end of the pinion shaft, causing displacement of the oil pistons at that end, the displaced oil flowing to the opposite end; and in the second instance, by flexure of the web of the I. beams supporting the pinion bearing frame.

It is interesting to note that in the Westinghouse system the oil necessary to float the pinion frame is pumped by the bearings themselves. This is accomplished generally by the fact that if oil is fed into the point of minimum pressure in a bearing it will be carried and discharged at the point of maximum pressure if a suitable opening is provided, and at a pressure equivalent to the bearing pressure. Further, by the use of pressure gauges and from the known area of the oil cylinders taking the reaction due to the tooth pressure, the horse-power transmitted can be easily calculated. The hydraulic floating frame can therefore be used as a dynamometer.

The problem of dealing with the tremendous pressures at the high speeds common to turbine work due to inaccurate cutting, the value of which will be apparent from the previous reference to Lasche's investigations, has led to the manufacture of flexible turbine gear wheels, which may be regarded as being somewhat similar to the Laminated gears previously described. This type of wheel is due to Alquist, and has been adopted by the General Electric Company of America for use in connection with the Curtis turbines manufactured by them.

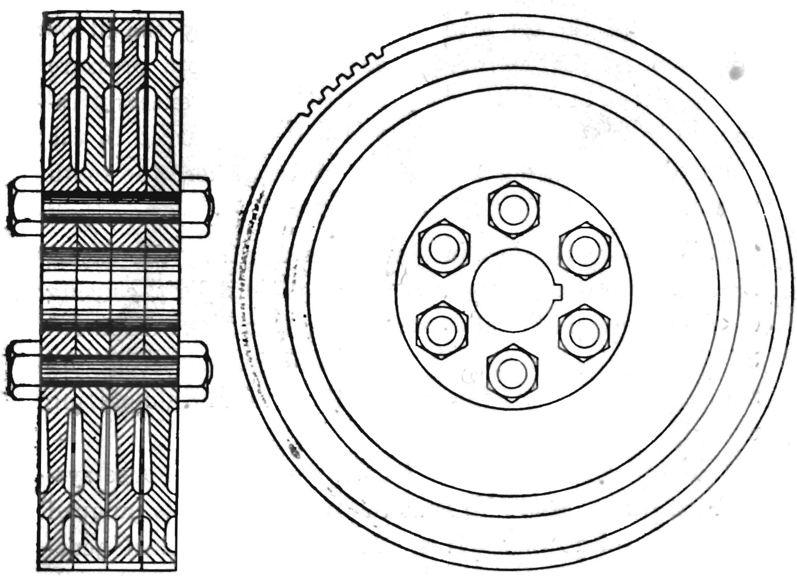

Fig. 9.

A section of one of these wheels is shown in Fig. 9, and the next lantern slide shows a double reduction turbine gear of this type for transmitting 1750 h.p., with a total ratio of reduction of 23.4 to 1, the actual speeds being 3200 r.p.m. to 137. This photograph is from one of two sets purchased by the U.S.A. Government for installation in the U.S.S. Nevada.

The gear wheels in this type are built up of a number of plates which are machined to a form which will give them a certain amount of lateral flexibility. The plates are bolted together near the hub, where there is contact between them. They also bear against each other near the rim, but after the teeth have been cut the rim contact is relieved, a gap of about 1/100 inch being provided between adjacant plates. Each plate is thus free to deflect laterally under the side pressure due to its diagonal engagement with the pinion, and a very small deflection is sufficient to afford the desired distribution of the load. For further particulars those interested are referred to a paper read before the American Society of Naval Architects and Marine Engineers, Nov. 1916, by W. L. R. Emmet.

How far the introduction of floating pinion bearings and flexible gear wheels has improved the operation of turbine helical gears the author is unable to say, but it is interesting to note that British manufacturers appear to have adhered to solid wheels with solid bearings, and relied upon extreme accuracy of workmanship and liberal ratings to secure satisfaction. The special features above referred to are, as far as the author knows, only common to American practice.

Worm Gearing.—This type of gearing is used in cases when high ratios of transmission are required, and also where smoothness of action is necessary. It is commonly believed among engineers, however, to be of low efficiency, whereas this is not the case. The efficiency is almost entirely dependent upon the angle of the worm, as will be shown by Fig. No. 10. This curve shows that a maximum efficiency of 93.3 per cent. is reached with a worm angle of 43.58, and co-efficient of friction of .035. With a worm angle of 20 deg. the efficiency is 90 per cent. Certain tests carried out by Prof. W. K. Kennerson for the Brown & Sharpe Company, and described in the Transactions A.S.M.E. Vol. 34, gave efficiencies as high as 97½ deg. In these cases the worm angles were 45 deg. with ball bearings on both worms and wheels. The following slide shows a large set of double worm gearing driving a set of deep well pumps, a gas engine being the prime mover. This worm gear set weighs over 15 tons.

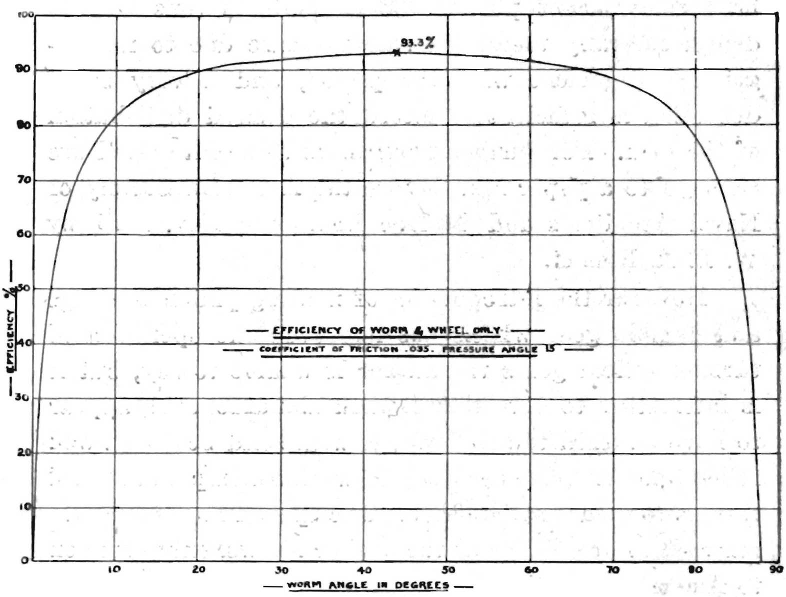

Fig. 10.

In this type one worm is right hand and the other left hand, the two being inter-connected by spur gearing. This arrangement is adopted in order to neutralise the side thrust of the worm wheels, which in worms with a helix angle of 45 deg. is, of course, equal to the pressure due to the load transmitted.

A feature of the worm gear which is often taken advantage of is that it can be designed to be self-locking, and this form is frequently used for lift gearing and hoists. In this type the efficiency cannot be higher than 50 per cent. if the gears themselves are to be self-locking. As, however, there is always some friction in the worm bearings and other parts of the machinery, the efficiency of the gears only may be considerably higher, and the gear still remain self-locking. A point to be remembered, however, is that a worm gear which may be self-locking while at rest may not be so when moving, owing to the fact that the co-efficient of friction with motion is considerably less than when the gearing is at rest.

The considerations which enter into the proper design and manufacture of worm gearings are too extensive to be entered upon now, but those interested will find considerable information on the subject in a book published by the Industrial Press of New York, entitled "Spiral and Worm Gearing."

The author is not in the position of being able to supply any new data affecting the design of gearing generally, but presents these notes in the hope that the modern tendencies in gear design to which attention is drawn may be of sufficient interest to promote discussion.

DISCUSSION.

The President: We are much indebted to Mr. Snashall for the able manner in which he has delivered his lecture on Gearing. The subject is one of which we have all had some experience, and I hope we will have an interesting discussion.

Mr. Saunders: I have pleasure in moving a vote of thanks to Mr. Snashall for his excellent paper; he seems to have dealt with his subject in such a thorough manner that there seems to be very little room for criticism, but if any member present can show us where Mr. Snashall has erred we shall be glad to hear him.