RECENT DEVELOPMENTS IN THE DESIGN OF REDUCTION GEARING

A description of the mechanism of toothed gearing was given in an article in No. 3 issue of these papers, and it was shown therein that the permissible load per inch length of pinion face varied with some function of the radii of curvature of the engaging teeth. If arrangements can be made to increase this radius of curvature, the permissible load can also be increased, and the length of pinion face correspondingly reduced. As the length of the pinion is reduced the pinion becomes stiffer, and both torsional deflection and bending are reduced until a point is reached when the centre supporting bearing of the pinion can be omitted, the result being that there is a substantial saving in the weight of the gearing.

Two interesting new forms of tooth have been developed with the object of enabling such an increased load to be safely carried as will permit of this saving being realized.

For many years the tooth form used for main gearing has been standardised, the standard hob form being shown in Fig. 6 of the article referred to above and reproduced in Fig. I of this article.

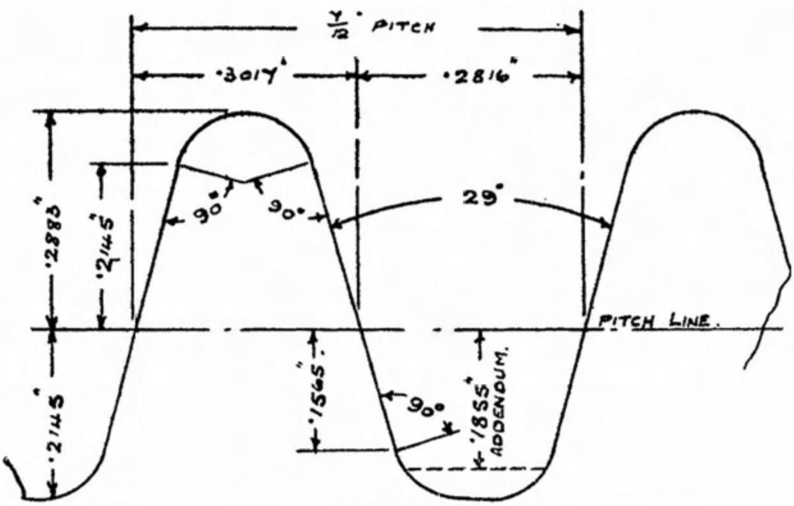

FIG. 1.

In the first of the new forms referred to, the involute tooth contour has been retained, but the tooth is formed wholly above the pitch line in the case of the pinion teeth and wholly below it in the case of the gear wheel teeth. The tooth form of the pinion tooth, therefore, lies between the addendum and pitch circles, a fact which gives rise to the name given to this form, viz., the " All Addendum " tooth. In addition to the alteration described above, the angle of obliquity has been altered from $14\frac{1}{2}°$ to $22\frac{1}{2}°$. The detail of the hob form is given in Fig. 2.

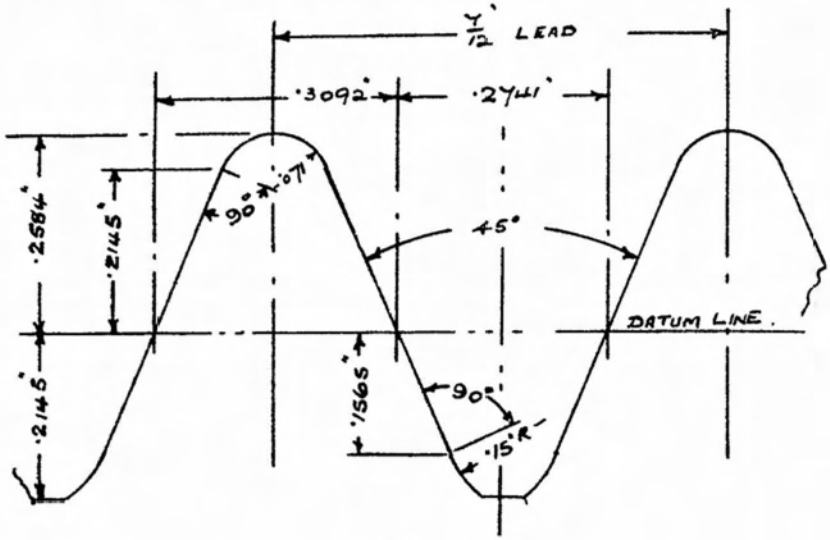

FIG. 2.

Referring to Fig. 3 the radius of curvature of an involute described on a base circle of radius r_1 at any point distance r from the centre of the base circle is given by $\sqrt{r^2 - r_1^2}$ and it will be seen that this increases as the distance from the base circle grows greater,

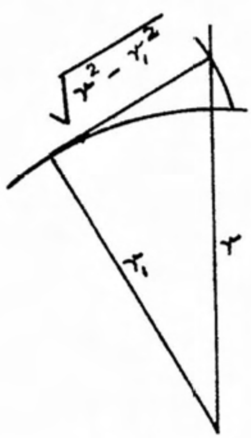

FIG. 3.

being infinitely small at the base circle. At the pitch point of a tooth of pitch circle radius r_0 and angle of obliquity ϕ the radius of curvature is $r_0 \sin \phi$ (*see* Fig. IIIA).

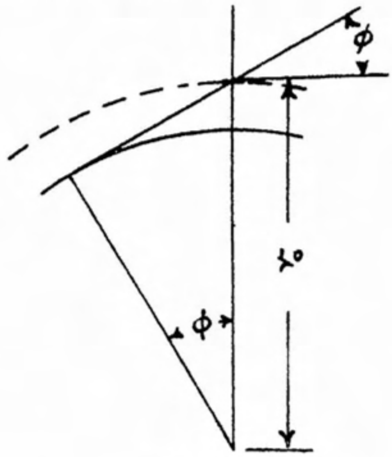

FIG. 3A.

It will be seen that the addendum portion of the tooth will have a greater radius of curvature than the dedendum, and also that increasing the angle of obliquity results in larger radius of curvature.

Increasing the angle of obliquity is not, however, wholly advantageous, as it leads firstly to a greater force being set up tending to part the gear wheel and pinion, which must be taken by the bearings, and secondly, it reduces the length of contact.

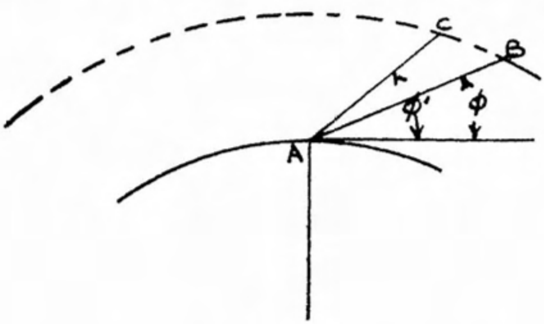

FIG. 4.

Referring to Fig. 4 the length of contact of the spur "all addendum" tooth of angle of obliquity ϕ is the distance along the contact line from the pitch point A until the contact line cuts the addendum circle at B, but if the angle of obliquity be increased to

ϕ^1 the length of contact is now only AC. The contact lengths of helical gear teeth would be correspondingly reduced by increase of angle of obliquity.

Increase of angle of obliquity has another effect—referring again to Fig. 4, if the speed of rotation of the pinion be the same in each case it is obvious that the time taken for the contact point in the spur tooth to travel from root to tip is less in the case of the larger angle of obliquity, and similarly in the case of the helical tooth the contact line moves across the tooth face at a greater speed. It appears probable that this effect is beneficial from a lubrication aspect, as the oil will have less time to extrude under the applied pressure.

With this new " all addendum " tooth form, an improvement in technique of manufacture has also been adopted. Tests carried out on gears under heavy loads showed that the gear wheel teeth were liable to develop pits early in the life of the gear, but if new gear wheels were meshed with pinions that had been " run in " and become slightly worn, these pits did not develop. The pitting was ascribed to the effects of high local loading due to the " high spots " which are inevitably present even in the best cut pinions. It was noticeable that the pinions themselves never pitted.

The pinion surfaces are now polished by being run in mesh with a mating wheel under a light load of about 10 lb./in., with an abrasive (emery powder and oil). The process is continued until about 70 per cent. of the surface is polished, the remainder being left as a witness, to ensure that the tooth form is not destroyed. Experiment shows that these polished pinions can withstand much higher loads without signs of pitting of the gear wheels.

In the second tooth form to be described, a departure from the involute contour has been made. It can be shown by geometry that the sole criterion necessary for teeth to mesh correctly and maintain a constant velocity ratio between pinion and gear wheel is that the common normal to the surfaces at the point of contact must pass through the pitch point for every position of contact. This condition will be fulfilled if the mating tooth contours are traced out by a point on a curve which is rolled first on the outside of the pitch circle of one wheel, and secondly on the inside of the pitch circle of the other. Any curve may be chosen provided that its radius of curvature is always less than the radius of the pitch circle on the inside of which it is rolled. Cycloidal teeth are a special example of this, the curve chosen in this case being a circle.

In involute gearing the curvature of the mating surfaces are opposite in direction as indicated in Fig. 5 (*a*), but when the gear wheel is of large diameter the curvature of the tooth profile is small, and it can be assumed that the tooth is substantailly flat-sided, as in the case of an involute rack tooth (Fig. 5 (*b*)). In the case of tooth contours generated by rolling curves in the manner

Fig. 5.

described the curvature of the mating surfaces is in the same direction as shown in Fig. 5 (*c*). It can be shown mathematically that the corresponding radii of curvatures in the three cases (*a*) (*b*) and (*c*) are respectively given by the expressions $\frac{r_1 r_2}{r_1 + r_2}$, r, $\frac{r_1 r_2}{r_1 - r_2}$ where r_1 and r_2 are the radii of curvature of the two surfaces in cases (*a*) and (*c*) and r the radius of curvature in case (*b*).

In the special tooth form being described, a generating curve has been selected which makes the mean value of $\frac{r_1 r_2}{r_1 - r_2}$ averaged over the length of the path of contact considerably larger than the radius of curvature which can be obtained by the involute construction. It may be anticipated, therefore, that a correspondingly higher load can be safely carried. This tooth form is sometimes known as the " enveloping tooth " owing to the curvature of the teeth being in the same direction, but is more often referred to as the " Vickers-Bostock & Bramley " or V.B.B. tooth, as Messrs. Vickers hold the British patent rights and Messrs. Bostock & Bramley are the inventors.

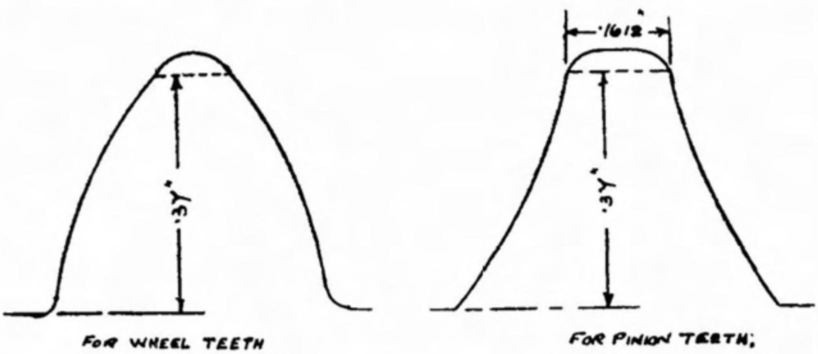

Fig. 6.

The hob tooth form is shown in Fig. 6 ; the tooth is of the " all addendum " type, the pinion tooth being formed wholly above the pitch line and the gear wheel tooth below it.

As the involute form has been departed from, the angle of obliquity is not constant, but varies from zero at the pitch line to some 36° at the tip of the pinion tooth, the point when engagement ceases. In spur gearing this change in angle of obliquity might be considered a disadvantage, as the load on the bearings would vary from point to point of the contact, but in helical tooth gearing where all except the ends of the teeth are in engagement from root to tip, the variation in load is small.

Theoretically, as opposed to involute gears, the gear will not work correctly unless the gear wheel centres are spaced exactly the right distance apart, experiment shows, however, that an error of as much as 0·02in. in an installation of normal size, does not sensibly affect the working.

The hobs to cut the mating gears must, of course, have different shaped teeth, and this is clearly indicated in Fig. 6. As the teeth are not flat sided the manufacture of a hob is more difficult than in the case of the involute tooth. Once manufactured the hob can be checked for accuracy in the same manner as the involute hobs—the method used to check the accuracy of the tooth profile being the " shadow graph " method by which a shadow of the hob tooth is projected on to a screen on which a much enlarged outline of the true tooth form has been previously drawn out ; errors of form can be readily seen and measured by observing the differences between the tooth shadow and the drawing.

Once the hobs have been obtained, the methods of manufacture of the V.B.B. gears is precisely similar to that of involute gears.