

June 24, 1952

M. F. HILL

2,601,288

LUBRICATING PUMP FOR ENGINES

Filed July 8, 1947

2 SHEETS—SHEET 1

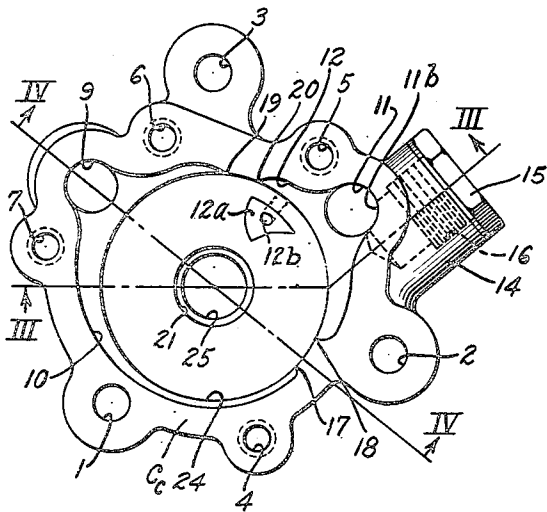


Fig. I

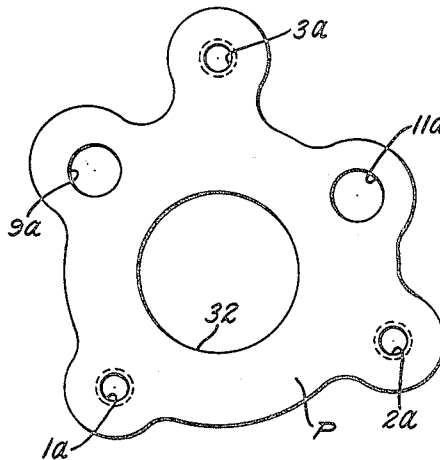


Fig. V

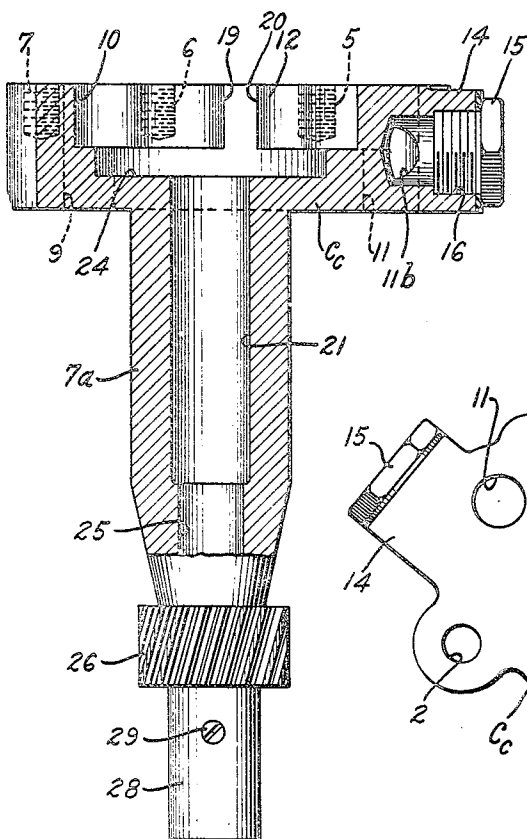


Fig. III

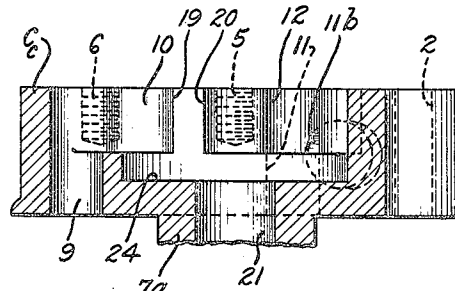


Fig. IV

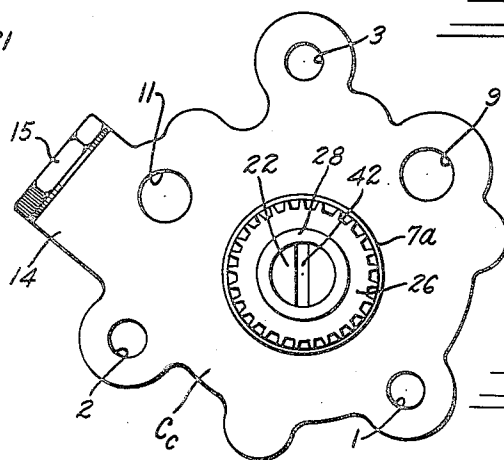


Fig. II

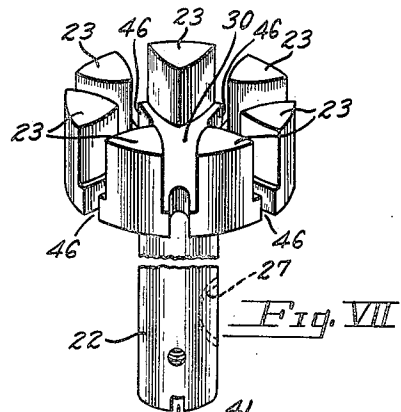
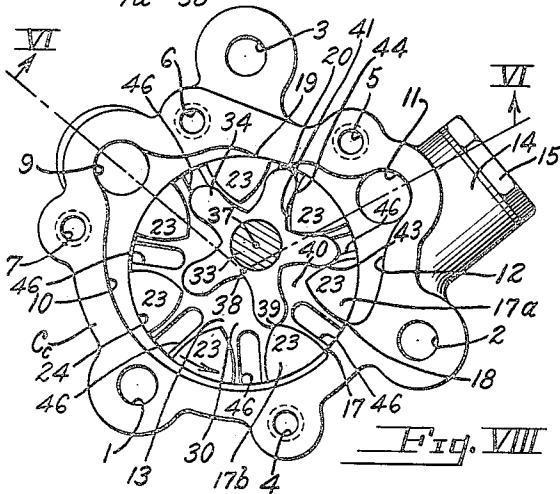
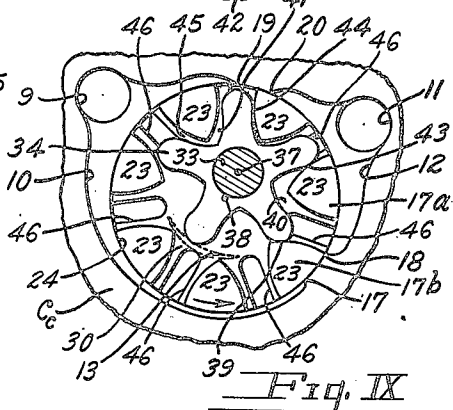
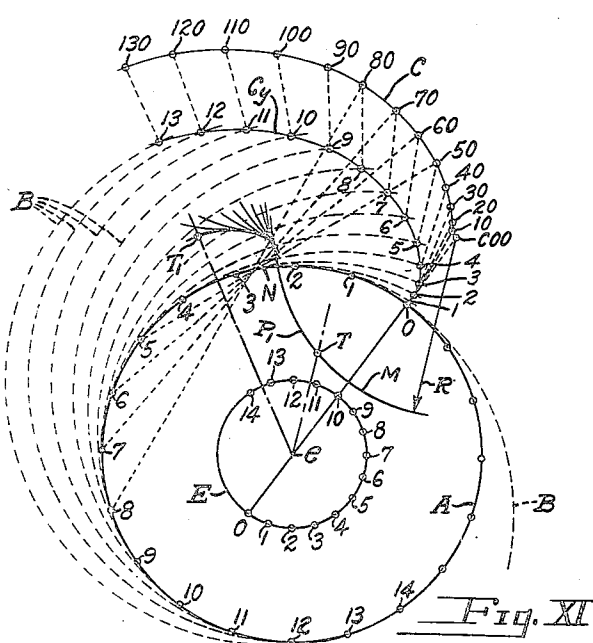
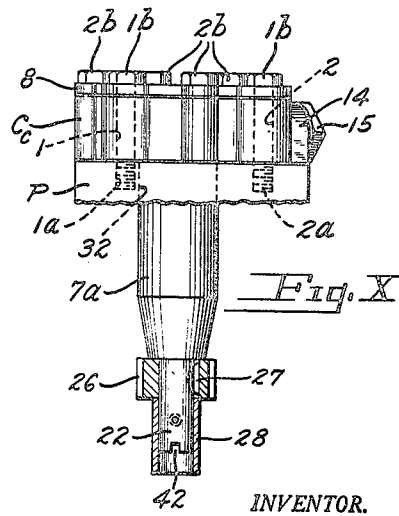
INVENTOR.

Myron Francis Hill

2,601,288

Filed July 8, 1947

2 SHEETS—SHEET 2



INVENTOR.

Myron Francis Hill

## UNITED STATES PATENT OFFICE

2,601,288

## LUBRICATING PUMP FOR ENGINES

Myron F. Hill, Westport, Conn.

Application July 8, 1947, Serial No. 759,551

9 Claims. (Cl. 103—126)

1

My invention relates to automobile engine lubricating pumps, of the type bolted to the side of the engine casing with a shaft-carrying projection passing into the casing carrying the helical driving gears.

A pad or flat machined area on the engine casting has three threaded holes for bolting the pump to the pad, and three other holes. One receives the shank of a pump containing the drive shaft geared to the cam shaft. The other two are oil holes or passageways, one connecting the sump of the engine to the pump inlet, and the other connecting the pump outlet or discharge port to engine bearings and cylinder walls, for lubricating them.

To change the location of even one hole on the pad in machines producing millions of cars is said to cost hundreds of thousands of dollars. No matter how meritorious the pump may be, if it cannot be designed to fit the standard engine pad, its adoption is difficult and usually impossible.

The pad was first designed for a double spur gear pump. That pump had two shafts one in a shank passing thru the pad, three bolt holes and two oil holes. The spur gears were upon opposite sides of a line from the center of one oil hole to the center of the other oil hole. Next a Gerotor pump (see Reissue Patent 21,316) now in general use, was designed to fit the pad.

With the standard bolt holes, it was a difficult task to locate a Gerotor pump upon the pad. It had two gears or rotors, one rotor within the other and having teeth differing in number by one, with their liquid tight continuous tooth contacts and oil passageways, invented by applicant. The oil passageways had to be tortuous and required a thick base.

These gears or rotors have now been greatly improved as shown in other pending applications. This application is for its combination with the aforesaid standard engine pad. Owing to bolt locations in the pad this pump, with its increased eccentricity, required radial ports, old in themselves, but novel in lubricating pumps on a pad.

This new pump, to which the trade-mark "Rotoid" has been applied, preferably has a difference of two teeth.

The earlier pump had the inner rotor shaft driven. The new pump has an outer rotor drive.

If the new pump had the inner rotor shaft driven its displacement with a given outside rotor diameter (for the same engine) would be

2

but slightly more than the present standard Gerotor pump. But the outer rotor drive adds two more chambers full of oil to its delivery per revolution, an increase of 40%. This enables the rotors to be cut from  $\frac{3}{4}$ " thick to .45" thick, reducing the height of the cup portion of the pump casing correspondingly. Rotors may however vary in thickness to suit different engines. The standard pump has ports in the bottom of the cup portion. These ports required tortuous passageways to connect with the oil holes in the engine pad, requiring a thick base portion under the cup, and special drilling operations.

My new pump eliminates the tortuous passageways and the thickness of the base portion of the casing and the drilling of ports is no longer needed.

The greatest obstacle was the outer rotor diameter. The maximum rotor diameter possible within the bolt holes is  $1\frac{3}{4}$ ". There were spaces between the bolt locations for radial ports and passageways. A driving plate to connect the outer rotor to the driving shaft was a necessity. The outer rotor has to be mounted on it. If the teeth are on the driving plate, the outer diameter of the rotor can be reduced to the outer ends of the tooth spaces. This solved the difficulty as to diameters, for a rotor having  $1\frac{3}{4}$ " diameter and  $\frac{1}{2}$ " thick enabled the cup to be of less depth, the bottom thickness reduced to  $\frac{5}{16}$ ", and radial ports connected by straight holes, plumb to the pad, to the engine oil passageways.

The outer rotor teeth, the plate and the shaft may be a single member as set forth more in detail below. Slots in the plate between the teeth admit oil pressures in front of the plate to its back side, balancing hydraulic end pressures. A dummy port connected to high pressure assists the balance.

In the drawings:

Fig. I is a top view of the cup portion of the pump with cover and rotors removed.

Fig. II is a bottom view with helical driving pinion removed.

Fig. III is a partial section on line III—III, Fig. I.

Fig. IV is a partial section on line IV—IV, Fig. I.

Fig. V is a plan view of the engine pad.

Fig. VI is a partial section on line VI—VI, Fig. VIII.

Fig. VII is a perspective view of the outer rotor and driving shaft.

Figs. VIII and IX show rotors in critical positions with relation to the ports.

Fig. X shows the pump in elevation mounted on the engine pad P with driving gear in section.

Fig. XI shows a method for designing rotor curves.

In Fig. I the pump casing Cc has the bolt holes 1, 2, 3 for fastening the casing to the engine pad P, bolts 1b (Fig. X) being threaded into the holes 1a, 2a and 3a of the pad (Fig. V) respectively.

The threaded holes 4, 5, 6, 7 are to receive cap or other screws to bolt the cover 8 to the casing Cc.

The hole 9 connects the port 10 to the hole 9a in the engine pad. This port, preferably, is the intake port of the pump.

The hole 11 connects the port 12 to the hole 11a in the engine pad. This port preferably is the discharge port of the pump. It is connected to a dummy high pressure port area 12a by passageway 12b shown in broken lines to hydraulically balance the rotary pump elements endwise. The suction port 10, it may be noted, connects with crescent range 13 (Fig. VIII) at open mesh thru the tooth spaces of the outer rotor as far as 17. A more complete filling of the tooth chambers with oil results favoring higher speed or wider rotors of greater capacity.

Boss 14 is provided with a plug 15 closing it. The threaded hole 16 in the boss connects with the port 12 through the holes 11 and 11b so that by removing the plug 15 and attaching a pipe line the discharge of the pump may be connected to other devices about an automobile. As the rotor chambers are filled before the end of the port is reached, the closing teeth exert pressure upon the oil to close vacuum cavities and reduce cavitation. The discharge port may begin at the point 18. The distance between the points 17 and 18 may be exactly the same as the distance between two rotor teeth, indicated at 17a and 17b in Fig. VIII, except for a small overlap to soften any possible noises. The other ends of the ports 19 and 20 have the same relation. A further reference to these ports will be made in connection with Figs. VIII and XI. The hole 21 (Fig. I) is to permit the passage of a shaft 22 in Fig. VII. This hole may fit the shaft to act as a bearing, or it may be larger than the shaft and not act as a bearing, with the outside diameter of the teeth 23 and plate 30 in Fig. VII having a bearing fit in the cup or recess 24 (Figs. I and III). With the latter arrangement, the edges of the teeth upon the outside diameters may be slightly rounded to facilitate the entrance of oil between the teeth and their bearings. The other end of the shaft fits a bearing at 25, Fig. III, and carries driving gear 26, keyed to it at 27 (Figs. VII and X). The sleeve 28 may be secured to the shaft by the screw 29, to hold the gear in place upon the shaft. The rotor teeth 23, form a part of the driving plate 30, Fig. VII. They may be formed integrally with or joined to the plate. The plate is fixed to, or is integral with, the upper end of the shaft 22, Fig. VII, and prevents the shaft from getting out of position inwardly. The gear 26 prevents it from getting out of position outwardly, with enough freedom of motion between the two to accommodate expansion and contraction under the conditions of use. The shank 7a projecting from the pump casing fits freely in the hole 32,

Fig. V, in the engine pad. In Fig. IV is shown a section of the upper portion of the pump casing upon the line IV—IV showing the port 10 connected with the hole 9. The recess 24 is also shown and one of the abutments between the points 19 and 20, Fig. I. Fig. II shows the bottom view of the pump casing in Fig. III. In Fig. VI is shown an assembly of the pump's cup and cover plate 8. It has holes to fit the various holes 1, 2, 4, 5, 6 and 7 (Fig. I). The cover plate has fastened to it a fixed stub shaft 33 carrying the pinion 34 freely journaled upon it. A snap ring and groove 35 in a recess 36 at the end of the pinion holds the pinion from dropping off when removing the cover plate from the pump. This stub shaft 33 is upon the eccentric pinion axis 37, Fig. VIII. This figure also shows the various teeth 23 and the ports 10 and 12. It will be noted that a pinion tooth and an outer rotor tooth engage at the point 39 when the pump is driven anti-clockwise. As these teeth come into engagement the chamber 40 between these two teeth and the next two teeth is disconnected from the suction port 10 and connected to the discharge port 12, so that, as the teeth close the chambers, they drive the fluid out of the chambers into the port 12 whence it traverses the hole 11 to the opening 11a in the pad Fig. V. The teeth on the other side of the centers 37 and 38 open the chambers between them for drawing in oil from the hole 9 in port 10. The chamber at 41 (Fig. IX) on the center line through the axis 37 and 38, has disconnected itself from the discharge port 12 and connected itself with the intake port 10. At full mesh no overlap is needed, although sometimes used to avoid excessive oil pressures due to irregularities in manufacture.

The advantage of these radial ports is that the passageways leading to the hole in the engine pad instead of being tortuous and requiring special drilling operations, require only holes 9 and 11 parallel to the driving shaft. These holes, if they are to be finished by drilling, may be drilled at the same setting as the hole 21 for the shaft and as the bearing recess 24 for the drive plate, eliminating the operations upon two different settings as required by Gerotor lubricating pumps. The outer rotor construction while requiring a driving plate nevertheless makes possible die casting the whole unit, including shaft, driving plate and teeth in one moulding operation. The cover plate 8 is so located with relation to the body Cc of the pump that the pinion shaft 33 is centered upon the axis 37 in Fig. VIII while the drive shaft 22 is centered upon the axis 38 as above described. Separately formed teeth and plate may be brazed, welded or otherwise attached to the shaft.

The cover may be applied to the pump as described by screws or bolts in the holes 4, 5, 6 and 7. When the pump is applied to the engine casing bolts in holes 1, 2, and 3 may then be applied. Bolts in holes 1 and 2 assist also in holding the cover to the pump casing.

It is possible to remove the pump from the engine by withdrawing bolts from the holes 1, 2 and 3 after which the pump is released from the engine pad and completely withdrawn without disturbing the pumping chamber. Or the cover plate may be removed by removing bolts from the holes 1, 2, 4, 5, 6 and 7, leaving the pump casing centered in the engine pad by means of the hole 32 (Fig. V) and bolted to the engine pad by a bolt through the hole 3 (Fig. I).

5

In Fig. II is shown a slot 42 to receive a driving connection from some other source of power if desired, as for shop testing. In operation the gear 26 is driven by a mating gear in an engine, causing the shaft 22 to rotate the teeth 23, which act as the upper bearing for the shaft and drive the pinion on the stub shaft 33, thereby opening and closing chambers, drawing in fluid from passageway 9 and port 10, and discharging it through port 12 and passageway 11.

In Fig. VIII, after the teeth engage (rotating anti-clockwise) at 39, they maintain their engagement as shown at 43 and 44, and in the position shown in Fig. IX, at 45, where they part company. Between the points 44 and 45 the work of one driving the other is effected. In this range at full mesh there is almost a pure roll over wide tooth areas between a convex curve and a concave curve of not very different radii; an ideal relation for economy of driving power, smooth action, and long life. Elsewhere the teeth wear to a polish and maintain the polish indefinitely. Between the teeth at open mesh the rotor chambers are merged, connected together by a crescent range between the teeth of one rotor relative to the teeth of the other rotor. It is in this region that a pinion tooth leaves an outer tooth, jumps over the next tooth and makes continuous contact with the next tooth ahead around to the point 45. After that, with backlash between the teeth, the tooth in question skirts along the periphery of a following pinion tooth but without touching it, eliminating the possibility of one rotor riding on the other. With no backlash, the teeth, except at open mesh along the crescent range, are tight to start with and are run in before released for use. With such an arrangement a shaft driven pinion may drive the outer rotor, and the long intake port retained.

The engine pad upon which lubricating pumps are mounted is illustrated in Fig. V. It is a machined surface upon an engine casting with bolt holes 1a, 2a, and 3a for clamping a pump casing tightly to the pad, and with oil holes or passageways for intake and discharge of oil for the pump. It has also a hole 32 for the shank of the pump containing the drive shaft geared usually to the engine cam shaft. The pad was designed for double gear pumps, that is pumps comprising two spur gears each in a separate pump bore in its cup fitting the outside of the gear teeth. The bores overlapped to bring the teeth of one into mesh with the teeth of the other, oil in the tooth spaces of one being displaced by the teeth of the other. The oil holes 9a and 11a had to be on opposite sides of the meshing area. Therefore while one gear was centered over the hole for the shank and driving shaft, the other gear had to be over the bolt hole 3a.

Gerotor pumps had to be designed to fit this pad and these bolt, shaft and oil holes. No change could be allowed in the pad on account of the very great expense of changing over production machinery correspondingly. And in a plant making millions of pumps a year the expense was said to run into hundreds of thousands of dollars. Instead of ports on opposite sides of a line between the pad's oil holes as in the double gear pump, the ports of the Gerotor pump are on the same side. Gerotor oil passageways followed a curved path in the base requiring some thickness. But ports in both pumps are in the bottoms of their cups.

6

The outer rotor of the Gerotor pump is a ring with teeth upon its inside. The ring has a journal bearing in the cup. These factors are of the prior art.

In the constant effort to reduce cost of manufacture and weight ideas occur which end the use of one form of pump and begin that of another. Reducing the two bores of the double gear pump to one, with its concomitant features is believed to be responsible for the general introduction of the Gerotor pump.

My present invention, the so-called Rotoid type of pump, still has but one bore, but the ports in the side wall of the pump eliminates most of the cost of machining it. The short abutments 17 to 18 and 19 to 20 which have to be fairly accurate are within the limits of modern casting methods. The driving plate in the bottoms of the cup would interfere with reliance upon a steady flow of oil at the intake. But peripheral ports, in the side wall of the cup provide for steady flow. The outer rotor teeth being on the driving plate, the outer ring of the Gerotor pump is eliminated, reducing the diameters of both rotors and pump casing.

Displacement of the Rotoid teeth of the same diameters is greater than with Gerotors and the outer rotor drive of the Rotoid pump increases the volume handled as 5 to 7. The thickness of the base is reduced as compared to either of the older pumps. Driving angles between the teeth also are improved, by more than 50%.

To plot tooth contours I first design one half of the pinion tooth. The other half of the tooth is its replica in reverse. The five teeth of the pinion are duplicates of such a tooth. The outer rotor has seven teeth with circular crowns, the dimensions of which are described below.

In Fig. XI the pinion ratio circle A is divided into any convenient number of divisions. Twenty are indicated, fourteen between the points 0 to 14 and six beyond, not necessary for my purposes. The advantage of 20 divisions is that one division has 18 degrees which is one-quarter of a pinion tooth division, so that a half of a pinion tooth will extend from the point 0 to the point 2 in the ratio circle A. Except for this convenience there might be any number of divisions desired, so long as the circle of eccentricity E, referred to later, has the same number of divisions.

The whole of the ratio circle B is not shown, but on the right hand side of the figure a part of the circle B is shown in broken lines tangent to the circle A at 0. Rolling positions of the ratio circle are shown at B on the left side of the figure also in broken lines. A radius of the ratio circle B extends from 0 in A where B is tangent to it, to 0 in the circle E. This radius is part of the rolling system, being fixed to the circle B and from the point of tangency of the circle B at 0 on A (referred to as A0), it swings to the left around the circle A through all the positions shown, 1, 2, 3, etc., and thru every point in between. The points 1, 2, 3, etc., are noted only for the purpose of illustration. When the circle B has rolled so that its point of tangency is A1 the end of the radius at the center 0 which is the center of the ratio circle B, has travelled around the circle of eccentricity E to the point E1. At the same time the point 0 in B has reached the point 1 of the cycloid (referred to as Cy1), thus beginning to outline this epicycloid. When the rolling circle B has reached the tangent point A2, one end, the center 0 of the

radius of B in E has reached the point 2 in the circle of eccentricity E (referred to as E2) and the other end, the point 0 in B, has reached the point Cy2 in the cycloid. As B continues to roll its tangency reaches the point A3, the center reaches the point E3, and the point 0 reaches the position Cy3. As the ratio circle B continues to roll thru all the successive points 4, 5, etc., the center 0 in E continues to travel thru the points E4, E5, etc., and its point 0 continues to travel thru the points Cy4, Cy5, etc.

Thus it is seen that this radius of the outer ratio circle B has one end travelling thru the points 1, 2, 3, 4, etc., in E and the other end thru the points 1, 2, 3, 4, etc., in the cycloid. Its successive positions are therefore positively determined. In plotting positions on a chart, circular lengths of arcs as A1 to A0 are measured off on each arc of B from its point of tangency with A. A curve thru these B points is the cycloid Cy.

Next we study the circroidal addition. This actually is the extension of the radius of the ratio circle B from B0 to some outside point. The point selected is C00. This point C00 at the end of the circroidal addition outlines the circroid C. For convenience of reference the radius and the circroidal addition together are called radicroid, a condensation of the phrase "radius of a circroid."

Theoretically, the circroidal addition may be from almost nothing up to infinity. The radicroid shown is within practical limits, aimed towards rotors having the greatest displacement for their diameter consistent with the differences of 5 to 7 teeth; and also with good pressure angles.

As the center E0 of the radius, or now of the radicroid, has travelled to the point E1 in the circle E and the point B0 in the circle B has travelled to the point Cy1 in the cycloid, the end C00 has travelled to the point C10 in the circroid. As the center E0 in the circle E continues to travel thru its successive points 2, 3, 4, 5, 6 and 7, its point B0 in the ratio circle B and in the radicroid travels thru the successive points 2, 3, 4, 5, 6 and 7 of the cycloid causing the end C00 to describe the circroid thru the points C20, C30, C40, C50, C60 and C70.

To find half a tooth division of pinions having other numbers of teeth, such as 3, 4, 6, 7, 8 etc., divide 360 degrees by twice the number of teeth.

The circles A and E for convenience should have corresponding divisions.

In the Epi system the end of the radicroid on the circle E always moves with the ratio circle of the outer rotor.

The next step is to determine the size of teeth, that is the radius of curvature of the teeth of the outer rotor. We do not yet know how large a radius R for the outer rotor teeth will work with the radicroid already drawn. It is this radius that develops the curve P1 of the pinion contour. The instant radius of a circroid at any point of course extends to that point on the ratio circle A upon which the ratio circle B is at that instant tangent. For example, the instant radius of the circroid, obviously is from C00 to 0 in the circle A. The instant radius from the point C10 of the circroid, however, is from that point to the point A1 of the ratio circle A, because when the point C10 has been reached by the point C00 of the radicroid, the ratio circle B, which carried the radicroid, is tangent to the point A1 of the ratio circle A.

The next instant radius might be from the point 20 in the circroid C to the point 2 in the ratio circle A, but as this instant radius is not involved in the final intersection it is not shown. The same is true of instant radii from 30 and 40. The instant radius from the point 50, extending to the point 5 in the ratio circle A is shown. Likewise instant radii from the points 60, 70 and 80 to the points 6, 7, 8 respectively in the ratio circle A are shown. It appears that the instant radii from the points 60 and 70 intersect nearer the circroid. It is well to examine other instant radii around points 60 and 70. As a matter of fact, in this particular figure and with the proportions shown, no other intersection appears substantially closer to the circroid.

A radius R, of the master generating curve is next determined. It may extend from the nearest point in the circroid to the intersection point N of the instant radii of the circroid. Actually the radius R is usually a trifle less than this distance from the intersection point N to the circroid. If it wasn't, the corner near N of the tooth curve P1, of the envelope outlined by arcs drawn from successive points of the circroid, is sharp. The radius is therefore reduced somewhat, the gap between the theoretical curve and the actual one rounding off this corner as well as accounting for drafting and manufacturing inaccuracies.

In actual practice it is well to use a large diagram for rotors of the size shown in Fig. VIII. Rigid beam compasses, sharp ink lines and precision layout result in sufficient accuracy for rotor designing. The first rotor actually generated is used to check the design.

A number of changes may be made in this layout for purposes of comparison. First a shorter radius R increases the radius of the corner or shoulder near the point N. Pressure angles of tooth contact also are slightly increased. Increasing the radicroid will produce curves that maintain continuous contact though the pressure angles are increased. If, however, the radicroid is decreased so that the same radius R outlines a curve P1 upon the other side of the point N, constant contact theoretically would be destroyed. But the radicroid might be decreased with a corresponding decrease in the radius R and the resulting curve would be a good operating curve. A radius R that is too long cuts away the driving contacts making the rotors leaky and noisy.

The diagram in Fig. XI shows the epi system. It is possible to describe upon the same diagram different circroids and different curves P1 for purposes of comparison. The selection of curves needs judgment assisted by experience.

If the circular arc for generation is modified, the curve P1 is correspondingly changed. For example if elliptical in form, and secured to the radicroid, a different shaped curve P1 is generated by it. Such an ellipse may be mounted as if the radicroid passed through both centers of the ellipse theoretically or the elliptical curve may be mounted at different angles. Pressure angles may be improved. With symmetrical teeth, the rotor contours are reversible, but unsymmetrical rotors can be put together in only the correct way. Instead of an elliptical curve the form M may have any other known or mongrel curve. Continuity of contacts depends upon the locations of points N within the envelopes outlined by the particular tooth forms employed, that is, locations upon the side of the envelopes away from the circroids.

A perfect cycloid is not subject to the laws of the circroidal addition. Such a curve has a radius reducing to zero at the ratio circle. All of its instant radii end at the ratio circle. But any curve with an instant radius which has a center of curvature outside of the ratio circle appears to be subject to the principles which have been laid down. If an involute curve is applied this diagram will show how far away from the ratio circle B it might have to be located to make continuous contact with a generated tooth of the pinion. While in this new art it is never safe to make an absolute statement, nevertheless certain truths appear and they have to be stated in such language as expresses what seems to be true. The original rule of continuous contact requiring a difference of one tooth, for example, had to be changed to a difference of one in the ratio of the number of the teeth of the respective rotors. So the rule about the location of the centers of curvature outside of the ratio circle might conceivably have to be modified by future disclosures. But whatever such curvature may be, having centers outside of the ratio circle, they must lie, it seems, between the circroid selected and the nearest intersection to the circroid of its instantaneous radii.

One rotor drives the other. In a pump, if a shaft drives the pinion, it drives the outer rotor as the chambers between the teeth open. If the shaft drives the outer rotor, it drives the pinion where the chambers are closing.

The outer rotor drive increases displacement by two chambers per revolution, but the shaft has to be connected to the outer rotor by the driving plate 30. Fluid pressures in rotor chambers are balanced by means of the slots 46 cut in the driving plate between the teeth as shown in Figs. VII, VIII and IX. Whatever the fluid pressure is between the teeth it is communicated thru these slots to the back of the driving plate 30, balancing the plate hydraulically. In designing these slots care should be taken that they do not connect together a rotor chamber having high pressure to a rotor chamber containing low pressure, during rotation.

As one rotor drives the other teeth that rub on each other wear until mechanical pressure between them ceases, leaving a wiping contact without mechanical pressure. The result is polished tooth surfaces. The mechanical pressure between the teeth due to driving is at this time limited to the length of a tooth division at full mesh. Any wear at full mesh is reflected in wear between the other engaging teeth to match. The drive at full mesh however is between a convex curve rolling in a concave curve of slightly different radius (due to generation) providing a wide contact area thru a sizeable oil cushion.

A master form having a radius of R carrying a generating arc M will generate the curve  $P_1$  for the pinion contour between the intersection N and the circroid C. It will generate more of the contour than is needed. This curve M— $P_1$  where it makes a sudden turn around the intersection N, has small radii of curvature. The portion of the tooth in the neighborhood of the letter  $P_1$  is narrower than the outer end of the tooth, which will provide a true running continuous contact.

We notice that with 5 and 7 teeth the tooth curves are radially taller than with 6 and 7 teeth.

Each tooth occupies one fifth of 360 degrees, namely 72 degrees, and one half of a pinion tooth will occupy one half of 72 degrees or 36 degrees. If now a protractor is used with an opening of 36

degrees and the intersections of the sides of the protractor coincide with the center  $e$  of the circle of eccentricity E, they may be swung on that center back and forth to select the best portion T to T' of the curve M— $P_1$ . The arms T—T' of the protractor centered at  $e$  enclose 36 degrees of the curve M— $P_1$  which is the complete curve of one side of a tooth. It lies between the points T and T'. That portion of the master form having the radius R, which generated this portion of the pinion tooth is the part of the master form M that is retained in the outer rotor for one side of a tooth. The other side of that tooth may be its replica in reverse.

An exact contour of a corresponding half of a tooth space of the outer rotor might be generated by a tool having the form of the pinion between points T' and T, but a space deeper than such a generated curve is preferable.

It will be seen in this figure that the contours of the teeth in the driving portion near the ratio circle A have pressure angles very close to zero since a tangent to the curve at the pitch circle might indeed almost pass through the center  $e$  as the drive occurs, not only at this point, but at other points outside of and inside of the ratio circle A, until the next tooth takes up the load. The angle increases and might run up to 20 or more degrees before the next outer rotor tooth engages the next pinion tooth. The 5 to 7 teeth of the exhibit illustrate how the teeth engage each other only after they have passed open mesh where an open crescent space between the teeth is left.

It might be noted that varied pressure angles are natural to rolling members. A fixed pressure angle appears to bar continuous contacts.

Rotoids do not need to have a preliminary run before they are ready for service.

Rotoids may have curves more crudely made particularly if certain portions of the curves are high so that they engage before other portions. The result of such an arrangement is that they wear in, in service, and become more and more efficient.

Curiously the displacement of rotors having a difference of two teeth is greater than that of rotors having a difference of one tooth, while the displacement of rotors having a difference of three or more teeth is less than of either. A difference of two teeth has greater eccentricity, therefore radially deeper chambers. Rotors having a difference of three or more teeth have longer crescent spaces between the teeth at open mesh where no displacement is effected. This crescent space in rotors added to the intake port in pumps tends to reduce noise of cavitation.

While I have shown and described my invention in its preferred form, its scope contemplates variations without departing from its essence.

The term "continuous engagement" includes tooth contours which, owing to tolerances of manufacture and other departures from theoretical contours, do not have actual contact but travel along, one with respect to the other, in such close proximity, being based upon the circroidal addition, that they provide volumetric efficiency suitable for the fluid being pumped. Even such tooth contours may be undercut in regions where the lack of continuous contact is not needed for such efficiency. Nor are the numbers of teeth limited to five and seven or to a difference of two teeth so long as the tooth ratio of the number of teeth of the respective rotors has a difference of one.

While I have described tooth curves of a gen-

erated rotor and of its mating rotor as being arrived at by starting generation with a circular arc as a master form from which all the tooth curves of both rotors are derived, it should be clear that if any one of the convex or concave curves so arrived at are used as master forms, the remaining contours are also derived from such master forms. This is true of rotors having a difference of one tooth. In rotors having a difference of more than one tooth, it is true as to one side of the teeth of the respective rotors, the other sides being duplicates in reverse.

It is also possible within the limits of my invention to change any one of such contours used as master forms, to elliptical, spiral, circroidal or other curves known to geometry, or mongrel curves, or to contours composed of parts of two or more such curves; in which case any such master form impresses its characteristics upon all the other generated curves of the teeth.

Nor is my invention limited to symmetrical teeth, for the contours on one side of all the teeth where they engage, may be derived from one of the aforesaid master curves, and the other side from any other of said master curves; provided however that whatever change is made, the resulting tooth contours, where they engage to create or maintain pressure functions of fluids, observe the principle of the circroidal addition.

Nor is my invention limited to the functions of a pump, since operating in reverse, it becomes a fluid motor, facilitated in operation by the low pressure angles between the teeth.

Furthermore, my invention is not limited to the particular shaft arrangement shown or described, so long as its outer bearing and means to prevent inward displacement are maintained.

What I claim is:

1. The combination for a pad or clamping area on an engine for a lubricating pump, said pad having a central bore for a pump driving shaft in a sleeve or shank, three threaded bolt holes in said pad around said central bore, for bolting said pump to said pad, and two oil holes each located between two of said bolt holes, one oil hole for connection to the sump of said engine, and the other to bearings inside of said engine; comprising an oil pump having a cup containing three bolt holes located to register with the bolt holes in said pad, a shank projecting from said cup and located to enter said central bore in said pad, a cup on said base containing two toothed rotors, one within and eccentric to the other, and having rotor chambers between them which open and close during rotation to receive and discharge oil, a driving shaft in said shank entering said cup, a drive plate in said cup having a drive connection to said driving shaft, the teeth of said outer rotor being mounted on said drive plate and spaced apart angularly to allow radial flow of oil between them, inlet and outlet ports in the wall of said cup surrounding the teeth of said outer rotor, oil holes in said cup, passageways from said ports to said oil holes in said cup, said oil holes in said cup located to register with the oil holes in said pad.

2. The combination claimed in claim 1 having

a cover plate on said cup and the inner rotor rotatably mounted on said cover plate.

3. The combination claimed in claim 1 having the ratio of the number of the teeth of the respective rotors differing by one.

4. The combination claimed in claim 1 having the ratio of the number of the teeth of the respective rotors differing by one, and the numbers of said teeth of the respective rotors differing by two.

5. The combination claimed in claim 1 having the basic ratio of the number of the teeth of the respective rotors composed of fractional numbers differing by one.

6. In combination, an oil lubricating pump having a cup, a cover for said cup, said cup containing two toothed members, one within and eccentric to the other and having a lesser number of teeth, rotor chambers between said teeth which open and close during rotation, the teeth of the outer rotor driving the teeth of the inner rotor and maintaining continuous fluid tight contacts between the teeth of said rotor chambers as they close during the performance of pressure functions, said teeth having contours providing a crescent range between the teeth where no tooth engagement occurs, a driving shaft in said pump projecting into said cup, a drive plate inside of said cup having a driving connection with said driving shaft, the teeth of the outer rotor being mounted on said drive plate and spaced apart angularly to allow radial flow of oil between them, intake and discharge passageways in said drive plate providing balance of oil pressure on the front and back side of said drive plate inside of said cup, and inlet and outlet ports in the wall of said cup cooperating with the spaces between the teeth and the passageways in said drive plate.

7. The combination claimed in claim 6 having the ratio of the number of the teeth of the respective rotors based upon fractional numbers differing by one.

8. The combination claimed in claim 6 having the ratio of the number of the teeth of the respective rotors differing by one.

9. The combination claimed in claim 6 having the ratio of the number of the teeth of the respective rotors differing by one, and the actual numbers of teeth differing by two.

MYRON F. HILL.

#### REFERENCES CITED

The following references are of record in the file of this patent:

#### UNITED STATES PATENTS

| Number    | Name   | Date          |
|-----------|--------|---------------|
| 1,672,257 | Heitz  | June 5, 1928  |
| 1,732,871 | Wilsey | Oct. 22, 1929 |
| 1,739,139 | Haight | Dec. 10, 1929 |
| 1,773,211 | Wilsey | Aug. 19, 1930 |
| 1,993,721 | Pigott | Mar. 5, 1935  |

#### OTHER REFERENCES

The Horseless Age, December 20, 1905, page 792.