Aug. 28, 1928.

1,682,565

M. F. HILL

ROTARY COMPRESSOR

Filed Nov. 5, 1921

2 Sheets-Sheet 1

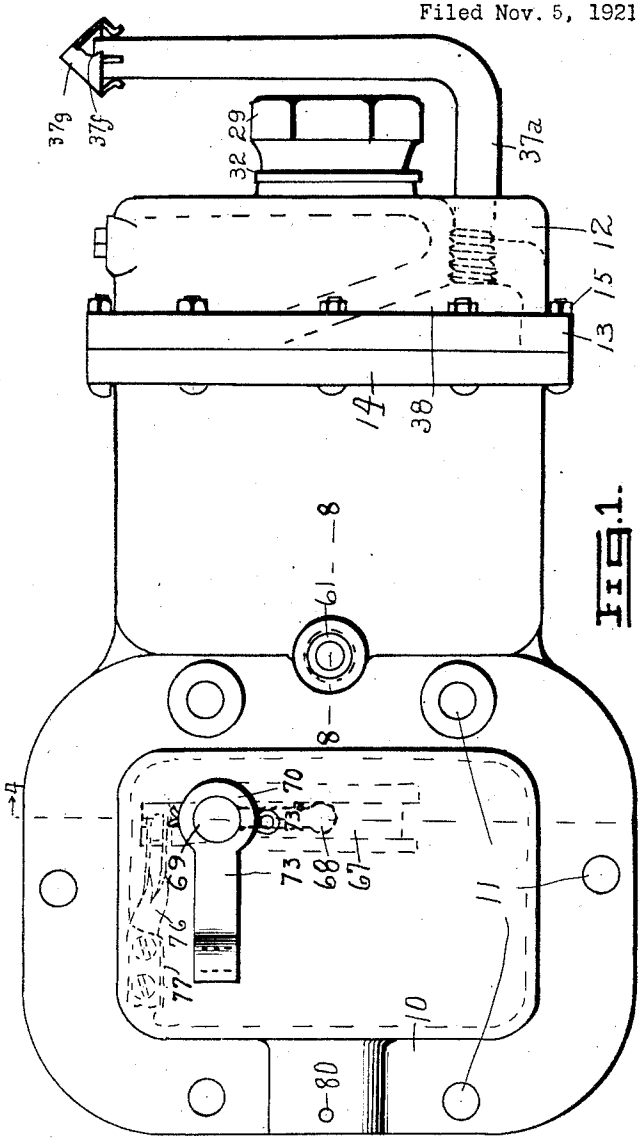


Fig. 1.

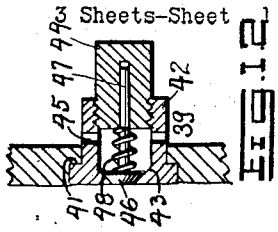


Fig. 12.

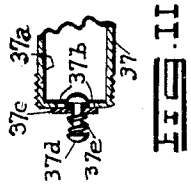


Fig. 11.

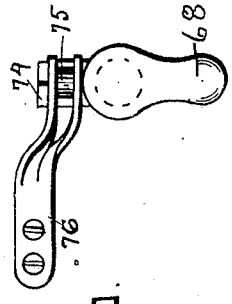


Fig. 10.

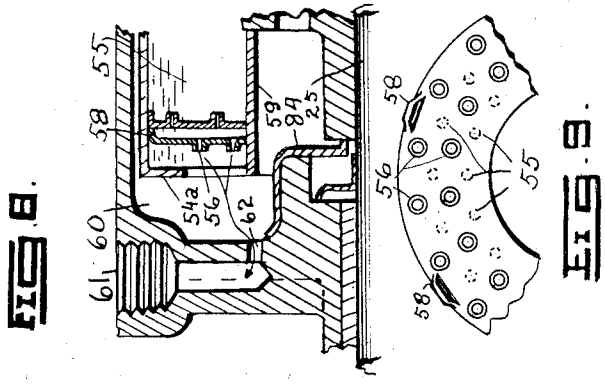


Fig. 8.

Fig. 9.

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3 Sheets-Sheet 2

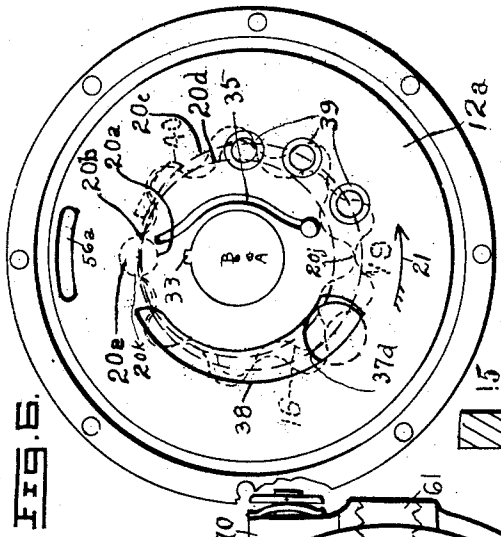


Fig. 6.

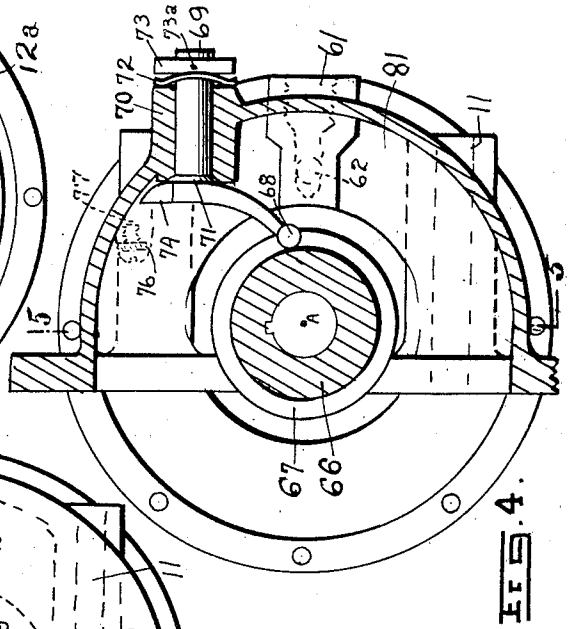


Fig. 4.

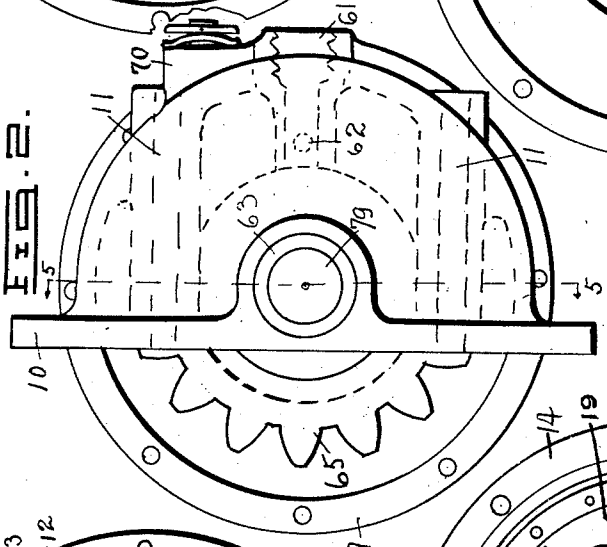


Fig. 2.

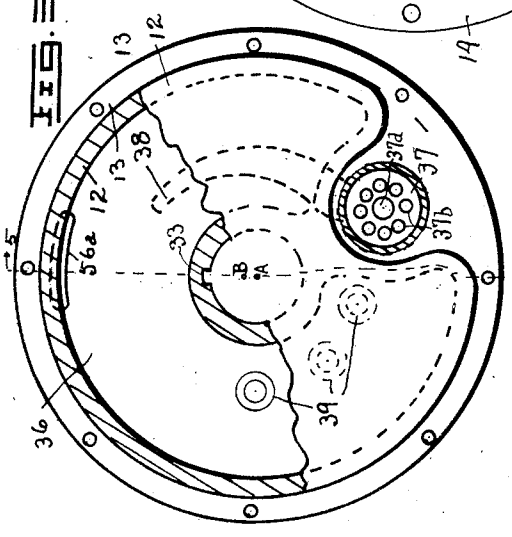


Fig. 3.

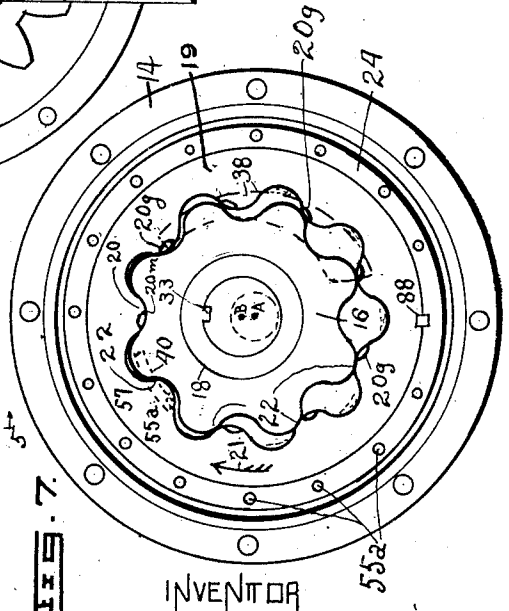


Fig. 7.

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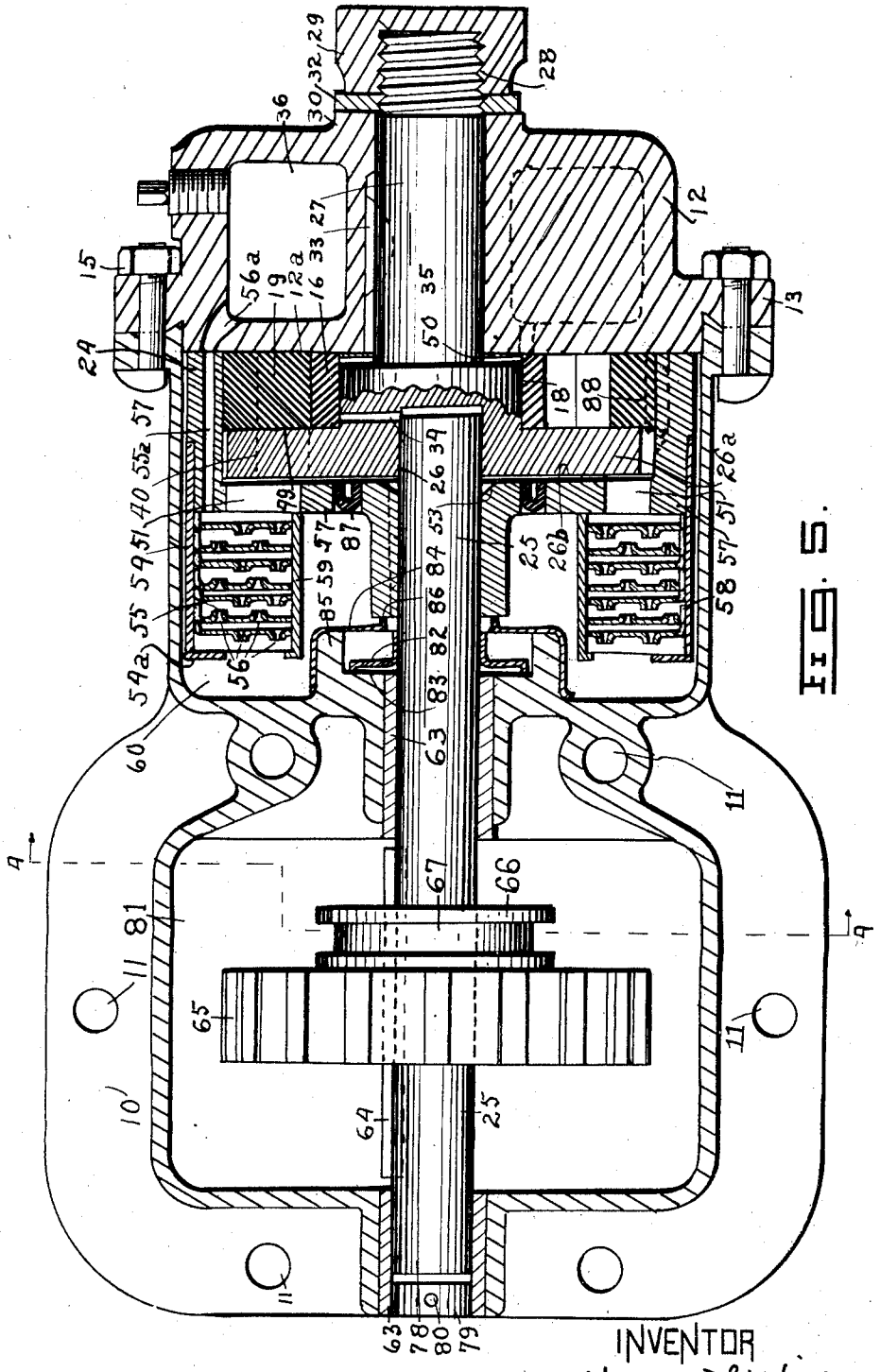


Fig. 5.

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Patented Aug. 28, 1928. 1,682,565

UNITED STATES PATENT OFFICE.

MYRON F. HILL, OF NEW YORK, N. Y.

ROTARY COMPRESSOR.

Application filed November 5, 1921. Serial No. 513,075.

My invention relates to air compressors, and particularly to air pumps for inflating tires of automobiles, tho some of its features have wider utility.

It is based upon two rotors or gears, one within and eccentric to the other, having tooth divisions, with contours on each outlined or generated by the tooth form of the other at regular angular motion—i. e., steady angular motor as the two rotors turn at the speeds inversely proportional to the numbers of teeth—to provide rotor chambers separated by travelling tooth contacts for performing pressure functions.

It relates preferably to a gerotor pump requiring for best efficiency a liquid seal, tho some of its features are useful in other connections. And the seal is arranged to circulate in order to be used over and over.

Another object is the cooling of the air as it is compressed, the liquid absorbing the heat.

Another object is means to cool the liquid after it has been warmed in service, the liquid being carried in a reservoir exposed to the air.

In the pump the liquid is intimately mixed with the air being compressed, and another object of the invention is its removal by means of a separator which utilizes centrifugal force in extracting the liquid from the air.

And another object is the prevention of damage to rubber tires by using as a seal a liquid that does not damage rubber, so that its vapor if condensed in the tire being pumped will not injure it.

In the drawings;

Fig. 1 is a side elevation of my pump.

Fig. 2 is a left hand end view of Fig. 1 with transmission case removed.

Fig. 3 is a right hand end view of Fig. 1 with parts broken away.

Fig. 4 is a section of Fig. 1 on line 4—4.

Fig. 5 is a vertical sectional view of the pump on line 5—5, Fig. 4.

Fig. 6 is an elevation of the inside of the cover plate showing the intake port and relief valves with relative positions of the rotor contours shown in broken lines.

Fig. 7 shows the right hand end of Fig. 1 with the cover plate removed.

Fig. 8 is a section of parts upon the line 8—8 Fig. 1.

Figs. 9, 10, 11, and 12 show details.

In Fig. 1 the casing of my tire pump is provided with a flange 10 and bolt holes 11 by means of which it may be bolted to the transmission case of an automobile. The casing may be varied for the purpose of fitting it to other mechanisms for driving it.

The casing is provided with the cover 12 adapted to be bolted by means of the flange 13 to the flange 14 of the casing by means of bolts and nuts 15.

Fig. 7 shows a view of my pump with the cover removed. This pump mechanism is provided with two rotors; the pinion rotor 16 having teeth 17 and journalled at 18, eccentric to, inside of, and driven by the outer, annular or ring rotor 19, having teeth 20. The pinion and annular rotors are provided with contours of a new type, such that the teeth of each in theory wipe or make continuous travelling contact over those of the other during rotation.

The contours have been described in theory. In practice, contours conform as closely thereto as practical conditions of manufacture and use permit. While errors in mechanical operations have a tendency to depart slightly from the theoretical exact curves described, the actual use of the rotors, particularly if such errors favor the operation, tend to wear together to create actual contacts between the tooth divisions, or their teeth, which maintain a polish on their contours where employed for the performance of pressure functions. In any event my invention makes it possible to make rotors of such definite curves that the teeth of one slide over the teeth of the other and into and out of tooth spaces, in such close proximity in spite of their relative shifting of angular positions, that very high efficiency, both volumetric and mechanical, is attained,—a result heretofore unknown with rotors of this type.

In Fig. 6, in broken lines the rotors are shown in a difference position from that shown in Fig. 7. In Fig. 6 the circular curvature of the pinion tooth and of the annular rotor space is indicated, for they coincide substantially with the circle 20^a which may represent the cutting periphery of the milling cutter. The axis of the circle 20^a follows cycloi-

dal paths, or variations of them called prolate trochoids. These contours show contacts at full mesh at the top of the figure where a pinion tooth contacts with the contour of a tooth space from 20^b to 20^k, a relation possible only with the specific curve system described in detail in this case. The contours are also shown in contact at open mesh at the bottom of this figure at 20^j.

During a complete rotation of the pinion about its axis, the tooth of a pinion travels over the surface of one annular tooth. Each complete rotation of the annular rotor, then, causes the pinion to advance relatively to the annular rotor the distance of one tooth. This is relatively eccentric rotation. In this movement the pinion tooth does not leave the contour of the annular tooth (except where the annular curve, not necessary for tightness between rotor chambers, may be undercut). It is well known that a crevice the size of a hair will permit air pressure to escape. It is therefore essential that the teeth maintain continuously, wherever tightness is necessary, a fluid tight pressure holding engagement with each other. There are countless numbers of types of curves possible for rotors. Haphazard curves are useless for this purpose. Without some system of geometry as a foundation for the curves no logical curves would appear feasible. My invention supplies this requirement.

It will be evident that each rotor has a contour outlined by the tooth form of the other as they rotate at steady angular velocity with relation to each other.

In other words the contours have geometrical outlines produced by three elements, namely two rolling circles and a master curve, assisted by a fourth element, a mating curve derived from the first three.

Let two circles be located upon a plane, one within the other and tangent to it. Let their diameters be in proportion to the numbers of tooth divisions selected for the two rotors which should differ by one. A master curve to represent the desired tooth form of a rotor (either rotor) is selected to start curve generation with and located upon the radius of the circle representing that rotor whose tooth form is so selected.

If the master form is located upon the radius of the inner circle it should have convexity upon its outer side. If located upon the radius of the outer circle it should have convexity upon its inner side. One circle is then rolled with the other, without slip at the point of tangency, so that there is steady angular motion of one with relation to the other, at speeds determined by their relative diameters that is, in the inverse ratio of the numbers of tooth divisions selected. This curved form is traced in all the successive positions it assumes with relation to the other circle, and a curve then drawn along the crests of the curves so traced as a curve of envelopment which is the contour sought for one rotor element. A portion of this contour, which may be called the mating curve, is then carried by the other circle, on its radius, as the rolling action is continued in the same way, and its form in every successive position with relation to the first circle is traced; and a curve of envelopment is drawn along the crests of the curves so traced. This latter curve is the contour of the other rotor element.

Applying this description now to the specific curve forms in which my invention is embodied, the master curve selected is a small circle or an arc of it, representing the addendum or crown of a tooth of the inner rotor. It is located upon the circle, or upon its radius with the center of the arc on or outside of the periphery, depending on its size. The two circles, with the inner carrying this arc, are then rolled one with the other and the form of this arc traced in each successive position which it assumes with relation to the other circle. This inner circle may rotate nine times while the outer circle rotates eight times, which speeds are in inverse ratio to the numbers of tooth divisions of the two rotors selected. This relative speed, not being varied, has relatively steady angular motion. The contour or outline of the outer or nine tooth rotor is the curve of envelopment traced along the crests of these traced curves.

A portion of this curve of envelopment, representing the convex crown of a tooth division of the outer rotor, and which may be called a "mating curve", is then located upon the outer circle (or upon its radius) in its generated position and the two circles rolled again in the same way, and the form of the mating curve is traced in all the successive positions it assumes with relation to the inner circle, during the rotation of the outer circle eight times and of the inner circle nine times,—being the inverse ratio of the numbers of tooth divisions of the two rotors. A curve of envelopment is then drawn along the crests of the traced curves (on the inside of course) to represent the contour of the inner rotating element which cooperates with the outer rotor element in the way specified.

A convenient method of laying out these curves is to mount the describing or outlining curve—either the master or mating curve—upon an arm which is swung around the center of the circle to which it belongs and rotate a blank around the center of the other circle, and trace upon the blank the describing curve in the successive positions which it assumes, the arm and the blank rotating upon their centers at speeds inversely proportional to, or in the inverse ratio of, the numbers of tooth divisions.

Instead of rolling both circles, one circle may remain stationary, and the other circle rolled in or on it as the case may be, always

without slip at the point of tangency. This is equivalent to mounting the circles upon a plane, and rotating the plane backward as fast as one or the other circles rotates forward, thus neutralizing it actual motion. Nevertheless with relation to such a plane their speeds still vary inversely proportional to, or in the inverse ratio of, the numbers of tooth divisions of the two rotor elements. Such a plane might be rotated at various speeds correspondingly affecting the actual speeds of the rolling circles, but not affecting their relative speeds. Rotating the plane causes the center of one circle to travel around the center of the other circle.

While I have mentioned tooth divisions, and convex curves, I do not want it understood that my invention is limited to the specific curves described or even to simple as distinguished from compound curves.

In the mechanical operation of making my rotors, the two rolling circles mentioned are the base or pitch circles 20^c and 20^d of the actual rotors, which vary in diameter in proportion to the numbers of tooth divisions. The master curve selected is the form 20^a of a milling cutter having cutting teeth upon its outer diameter. The mating curve has the characteristic of the convex portion of the tooth of the outer rotor, and its generated position on the base circle. The two base circles and the master milling cutter determine the contour of the outer rotor and therefore of the mating curve. That is, the mating curve is derived from those three elements.

A blank for the outer rotor is selected with a hole in it to clear the tooth positions, so that there is material from which to form the teeth. It may be mounted in a milling machine in which the milling cutter is carried upon a fixed axis,—which corresponds to holding the inner base circle in a fixed position, as above described. The mechanic is supplied with a table of figures to shift the table vertically and horizontally between cuts so that with relation to the blank the milling cutter follows the successive positions of a tooth of the inner rotor. If the rotors are of the size shown in the drawings, twenty successive cuts from the top of a tooth to the bottom of the next tooth space, repeated for all the other teeth and tooth spaces; and twenty cuts from this bottom position of a tooth space to the top of the next tooth similarly repeated; will form the contour of the outer rotor. The surfaces have minute serrations which are removed by wearing the rotor into its mate.

The mating rotor, the pinion, is then to be formed. A shaping cutter or "mating" tool is then made having the contour of the convex portion of the tooth form of the outer rotor and mounted in a so called "shaper," and a blank slightly larger than the outside diameter of the pinion or inner rotor is mounted on the table of the shaper. The mechanic is supplied with figures for setting the table in a series of horizontal and vertical positions and in each position the mating tool cuts the blank. Such positions are the positions of the tooth form of the outer rotor with relation to the contour of the inner rotor. Twenty such positions from the top of a tooth of the inner rotor to the bottom of the next tooth space and twenty more from that point to the top of the next tooth, repeated for all the teeth and tooth spaces, provide the outline or contour of the inner rotor. This method corresponds to the geometrical description above noted in which the inner circle is held stationary and the outer circle rolled upon it without slip at the point of tangency.

Such tables are supplied with micrometer divisions on the screws that adjust them, so that accurate settings are possible.

It is apparent to a mechanic that rotors so made fit so tightly one within the other that rotation is difficult, and one rotor has been worn into the other by an operation which may be called "burnishing" which wears off the minute serrations between the cuts so that the rotors word freely and easily and maintain the contacts between their contours in the region of tangency of the base circles which is usually (in gear parlance) termed "full mesh"; and in the region opposite where the base circles are farthest apart usually termed "open mesh," which are utilized in fluid mechanisms to keep the pressure in one passageway from leaking through the teeth over into the other passageway. In both full mesh and open mesh regions this contact is travelling and continuous during rotation so that as the teeth shift in their relative positions with each other and the tooth spaces they do not recede from each other at points which would permit a substantial dissipation of pressure from a high pressure passageway over into a low pressure passageway. My contours with their steady angular velocities maintain fluid tightness in these regions so that high mechanical and volumetric efficiency are made possible.

Whatever the master curve, and whatever the contour system it creates, this continuous travelling fluid tight relation between the ports both at full mesh and open mesh regions is maintained, tho the port locations shown apply specifically to the specific type of contour shown in which the master curve is circular, and represents a tooth form of the inner rotor.

If the diameter of the milling cutter is $\frac{7}{8}$ths (of any unit of measurement) its center should lie .023. more or less, outside of the base or pitch circle, when there are eight tooth divisions of the form shown on the inner

rotor, and nine tooth divisions on the outer rotor. The base or pitch circles vary as eight to nine, the inner circle having a radius of eight units and the outer a radius of nine units, their diameters being sixteen and eighteen units respectively, which is proportional to the numbers of tooth divisions selected for the rotors.

When one rotor is "burnished" into the other as described, it is apparent that a tooth of one rotor makes such a close engagement with the contour of the tooth space of the other rotor in the full mesh region that the travelling engagement substantially preventing leakage is realized, and with the ports in the positions shown there is always at least one point of contact or engagement of the nature specified between the rotor contours substantially preventing the leakage in this region.

Some slight degree of lost motion between a pinion rotor and the outer rotor is needed if they are to work freely together. Portions of the curves which have no functions for purposes of pressure or angular motion, may be varied to provide such lost motion. Whether or not the teeth make contact upon the intake side of a compressor in which the outer rotor is driven, for example, along the intake port where the chambers are opening from full mesh to open mesh, is unimportant since the rotor chambers are connected together through the intake port anyway. In such a form of mechanism the adjacent tooth faces may be varied to provide the amount of lost motion desired. Such a variation in no way affects the other sides of the teeth where they perform pressure functions or provide for steady angular speeds, for these sides of the teeth still cooperate in the manner specified. The curves may be varied by burnishing, or the sides of the teeth referred to that come into proximity along the intake port (in the example cited) may be varied, that is, relieved.

In forming such rotors, machinery is incapable of making contours theoretically perfect. They vary due to lost motion in the journals, lack of rigidity, and other causes. Moreover journals may vary due to necessary freedom for cool running, so that the theoretically perfect eccentric distance may not in practise be attained. When rotors are put together they have high points causing friction at points where no friction should exist. And they may ride on each other and not fully ride in their journals. Running them in, which may be termed burnishing, causes the high points to be rubbed off and causes the mutually generative relation to be perfected by this action. In this way the rotors come to have the tightness of a valve to its valve seat. Then the rotors settle down into their journals fully and run easily and with high volumetric efficiency.

In practice after once determining correct contours, the rotors may be made by any suitable machinery, milling or otherwise.

The rotors are preferably ground flat on both sides to the same thickness thruout. The back plate is also ground flat to make an easy sliding surface for the rotors to run against. If the rotors have their theoretically correct forms, one would have to be driven into the other, and couldn't turn in it. In machine practice lost motion is necessary for easy running. This is true of rotors. On one side of the center line, the left side in Fig. 7, the teeth make continuous travelling contact. On the right hand side they part company particularly as the teeth wear. So that along the intake port the chambers in practice are joined into one. Portions of the contours which have no function in pressure holding engagements, as at 20^5, may be cut away for various purposes.

As the rotors compress any gaseous fluid, they travel in the direction of the arrow 21 with relation to the inlet and outlet ports as located. In travelling in the direction indicated the teeth tend to engage each other at the points 22 on the compression side of the rotors. This is due to the fact that the annular rotor drives the pinion, and also to the fact that the pinion is resisting the forward drive thru the pressure of the air being compressed in the rotor chambers which tends to rotate the rotors in a direction opposite to arrow 21. The driving power for a compressor of gas is applied to the outer rotor preferably, the teeth of which drive the teeth of the inner rotor forward, clockwise in Fig. 7. This creates fluid tight pressure holding travelling contacts between them and forms rotor chambers, and prevents leakage thru the tooth contacts, particularly when assisted by the liquid seal. Moreover, during air compression there is a sort of staged or stepped up pressure in the series of rotor chambers between the inlet port and the discharge port. The contacts between the teeth of the rotors in this region, as the rotors rotate, is continuous, due to the mutually generative contours.

In practice the amount of wear between the teeth depends upon the pressure, speed, and relative curvature. The nearer the contacts are to open mesh, at the bottom of Figs. 6 and 7, the greater the wear. The nearer full mesh—at the top of the rotors in these figures, the less the wear. The durability is therefore determined by the wear at full mesh for the length of a tooth division. And elsewhere the teeth wear until pressure is about eliminated, and further wear there practically ceases unless the teeth wear at full mesh, in which case other contacts wear correspondingly. It is obvious that such rotors "wear tight" instead of wear loose, that is, maintain

friction free travelling contacts regardless of wear.

When first assembled one rotor is apt to ride on the teeth of the other. After they have worn in, the rotors settle down on their journals. Whether or not the teeth engage on the intake side in a pump or compressor is of no consequence since there is no pressure there to leak.

It will be noted that an annular tooth 20 rolls upon a tooth space of the pinion as they cross the center line at full mesh. This travelling contact between the ports prevents leakage. A tooth driving contact may also cooperate in this function as it nears the center line at full mesh.

The pinion rotor 16 is mounted upon the fixed shaft 18, Figs. 5 and 7, and the annular rotor 19 is shown mounted in a driving member or cylinder 24 which is fixed to a rotatable driving plate 57 and shaft 25, the right hand member of which is journalled at 26 in the casing member which carries the journal 18 f the pinion. This fixed eccentric casing member, carrying the eccentric journals of the two rotor members, is provided with a shaft extension 27 and screw threads 28 by means of which it may be bolted against the cover plate to prevent it from being pushed by rotor pressure away from the rotors thus permitting the pressure in a rotor chamber leaking out over the rotor face to the low pressure side. A packing washer 32 and hollow nut 29 prevent leakage from around the shaft. It will be noted that pressure from the discharge port reaches the back side of plate 26^a at 26^b. Tire pumps employ from one hundred to one hundred and fifty pounds pressure, and such pressure on the surface 26^b forces it to the right towards the rotors, but in starting unless it is held sufficiently by the packing washer and nut to confine the pressure in the rotor chambers, no pressure could be created there.

The shaft is prevented from turning in the cover plate 12 by means of the key 33, lying in slots in the shaft and cover plate.

The journal 26 is lubricated by means of the oil passage 34 extending from the journal of the pinion, which in turn is supplied with oil from the port 35, Fig. 6, connected to the oil reservoir 36.

An air intake check valve 37 may be employed, see Figs. 3 and 11. This check valve has a pipe end 37^a, with a flat end perforated at 37^b and provided with a valve plate 37^c secured by means of the pin 37^d. The spring 37^e causes the valve to close until the suction of the compressor pulls it open. When the pump stops the check valve prevents the issue of pressure from the intake port. The valve during operation is open all the time, admitting air to the suction or intake port 38, Fig. 6. This port is also indicated in Figs. 1, 3 and 7. In Fig. 7 its relation to the rotor chambers is shown, the port itself being in the member removed,—the cover 12.

The intake pipe 37^a extends upward to 37^f, Fig. 1, above the level of the oil in the reservoir, to prevent oil from seeping out while the pump is idle. Upon starting, oil in this pipe is sucked back into the rotors providing them with the liquid seal, much of it being driven thru the valves 39 back into the reservoir. The valves thus become relief valves for excess pressure in the rotor chambers tending to choke their action. A tiny roof 37^g to shed rain or foreign matter may be provided to shield the open end of the pipe.

After the chambers are filled with air at their point of fullest opening, at full open mesh, they begin to close to compress air. After the rotors have been idle sealing liquid is apt to fill these chambers and prevent easy starting of the rotors. The check valves 39 act as relief valves to permit this oil to escape into the reservoir. When starting, there is no pressure in the reservoir, and even air compressed in the rotor chambers open these valves and enters the reservoir, so that starting is made easy. This is a means to "unload" the compressor for starting purposes. The valves are located so that including the port 40 there is at all times a means of egress from the rotor chambers for fluid under excess pressure. Before as the other chamber passes from one valve 39 it connects with the next one. And before it leaves the last one, it connects with the discharge port 40.

In Fig. 12 is shown a check valve 39 in section. In a counter bored hole 41 in the cover plate 12 may be driven the valve casing 42 having the valve seat 43 and the plug 44 threaded into it—the casing. The plug is provided with perforations 45 to provide an exit for the fluid. The valve 46 has a stem 47 to guide it and a light spring 48 to set the valve against the seat 43 normally. During normal operation the pressure in the reservoir behind the valves keeps them closed, unless an excess amount of liquid gets into them to force them open.

The port 40 is shown in Figs. 5, 6 and 7. It is indicated in Fig. 6 for the purpose of showing its circular position relative to the rotor chambers. It is not located in any member shown in Fig. 6. Instead, it is located on the other side of the rotors in the back plate 26^a, the front surface, 49, of which has a sliding engagement with the rotors, and closes this end of the rotor chambers. This plate does not touch the cylinder 24, preferably. From the discharge port compressed air containing liquid issues into the cavity 60 thru a liquid separator of a centrifugal type. This pressure is conveyed to the rear side of the plate 26^a between it and the driving plate 57 and exerts a pressure on the stationary plate to press it towards the rotors to prevent the pressure in the rotor

chambers from pushing it to the left, Fig. 5, away from the rotor chambers which would have the effect of reducing the discharge pressure at the port 40, by permitting leakage over the sides of the rotors. While the nut 29 bolts this unit to the cover plate 12, the packing washer 32 prevents an unyielding joint from being attained. If it yields the thickness of a hair, no pressure will be developed. The plate 26ª, if it has a diameter of three inches or more—as it was built—and a pressure of 100 lbs. is maintained, is given a thrust toward the rotors of 600 lbs. or more. Such a braking action would wear the rotors badly and develop great heat vaporizing the oil and preventing air from being sucked in since the chambers would fill with hot vapor trying to escape. To prevent these evil effects I provide two remedies; one, enough liquid seal to quench the heat, and second, one or more thin shims 50 to just lift the plate 26ª off from the rotors to remove the braking action. If the shim is not thick enough to remove the braking action at first the wear of the plate against the rotors in time permits easy running. The nut 29 and resilient packing washer 32 oppose the plate 26ª from being pushed away from the rotor chambers when starting by pressure developed in the chambers. If the plate 26ª could be so pushed away from the rotors, they would be incapable alone of developing the pressure necessary to move it back again. The nut is therefore a means to confine its endwise motion, to maintain a starting pressure-holding film between the rotors and the front and back plates. The 100 lbs. or so of air in one rotor chamber, with lesser pressures in succeeding rotor chambers on one side of the center line thru the axes, is incapable of course of overcoming the 600 lbs. or more on the other or back side of the plate, so that in normal running there is always a resultant pressure on the back of the plate to press it forcibly toward the rotors.

After the air charged with more or less oil or liquid seal—as hereinafter described—has issued thru the outlet or discharge port 40, it passes through the holes 51 in the driving plate or member 57 welded to the shaft 25 at 53. This driving plate is preferably integral with the cylinder 24 carrying the annular or outer or ring rotor 19. Mounted on this unit also is a thin cylinder 54 containing a plurality of perforated plates 55 capable of acting as a centrifugal separator to separate the liquid from the air after compression. Liquid fed to the rotors enters in a comminuted state due to the pressure behind it—the discharge pressure—and as the air is compressed and it starts to rise in temperature due to compression, the mist absorbs the heat and as no high temperature is created, and as it is at a substantial pressure, little or no vapor is in the air,—certainly less than at atmospheric pressure. The centrifugal action tends therefore to send the liquid outward and the air inward. As the air has to pass thru the perforated plates 55 before reaching any exit, it is giving up its liquid to each baffle plate that it comes in contact with. Enough baffles remove the liquid from the air substantially.

The perforations in the baffles are staggered as indicated in Figs. 5 and 9. The plates themselves are separated by lugs 58, or clips, loosely fitting the inside of cylinder 54. The liquid thrown outward by rotation passes over the edges of the plates between the lugs 58 to the holes 55ª, Figs. 5 and 7, whence it passes thru the port 56ª, Figs. 3 and 7, into the reservoir 36. It is fed back to the rotors thru the oil port 35 and as it enters the rotor chambers along the intake port and during the lower stages of compression it is shredded off the rotor edges into the chambers, driven by the pressure in the reservoir. It thus becomes a mist. The reservoir 36 is large enough to hold liquid for cooling enough air to fill one or more tires according to design. It has time to cool off when not in use.

In order that the baffling operation of the plates may be more complete, the perforations 56 in one plate may be staggered with relation to those in the next plate. This is easily done by locating holes in alternate plates on different circles from those of the intermediate plates. A stream of air issuing from one hole strikes the flat of the next plate, spreads out, and is distributed among a number of holes in the next plate. Each stream of air following such tortuous baffling courses gives up practically all its liquid to the baffles. In order to prevent air from escaping the baffling action it is confined between the outer cylinder 54 and an inner cylinder 59 mounted on the edges of the separator plates. Pure air entering the chamber 60 passes thru the hole 62 into the vent 61 provided with screw threads for attaching the usual hose which connects with a tire. The air is cool and dry. Any accidental mishandling which sends liquid into the tire does no harm since the oil does not disintegrate rubber as does ordinary lubricating oil.

The hardened shaft 25 is journalled in cast iron bushings, by preference, cast into the casing, which may be of steel, or other suitable material.

A key 64 in the shaft provides the driving action for the gear 65 slidably mounted on the shaft, adapted to be driven by another gear inside the transmission case of an automobile for example. The gear 65 is provided with a boss 66 and trunnion groove 67, the latter being shown also in Fig. 4. In this groove there works the finger 68 adapted to slide the gear 65 laterally upon the shaft 25 and on the key 64. This finger 68 is inte-

gral with the pivot 69 journalled in the boss 70 of the gear case, as shown in Figs. 1 and 4. The pivot has a conical seat 71 to which it is held by the spring washer 72 and lever 73 attached to the pivot by means of a pin 73ª to hold the spring tense. This is to prevent the escape of oil from the transmission case through the pivot journal. The lever 73 may be operated by any suitable means to throw the gear 65 into and out of mesh to start and stop the pump. Means are provided to hold the shaft in either position securely. The finger 68 has at its rear end 71 a wedge shape upon which rides a roller detent 75, Fig. 10, pressed down by the spring 76 mounted upon a boss 77, Fig. 1. This boss is shown in side elevation in broken lines in Fig. 4, but it is not in this end of the casing. Fig. 10 construction is shown in broken lines in Fig. 1. In Fig. 10 the roller is shown in transit from one side of the wedge 74 to the other side. In shifting the gear 65 from one position to the other the pressure of this roller has to be overpowered. The outer bearing of the shaft is closed with a plug 79 held in place by a pin 80. This plug prevents the escape of oil and takes the thrust of the shaft in one direction.

The chamber 81 open to the transmission case, is normally filled with grease, and to prevent it from leaking into the chamber 60 when the pump is idle and all pressure has leaked out from chamber 60. Furthermore it is desired to prevent leakage from that chamber when the pump is active and it contains air pressure. A two way stuffing box is provided, consisting of the spring diaphragm 82 having a cylindrical hub driven onto the shaft 25 and a turned over edge running against the surface 83 of the casing; and a second spring diaphragm turned in the opposite direction having a hub driven on the casing boss 85 and having an edge resting against the surface 86 of the driving member on the shaft 25. During idle periods, whatever oil tends to ooze into the space between the two diaphragms, tends to be driven back into chamber 81 where pressure is low, by the high air pressure in chamber 60. The rotation of the diaphragm tends to throw the grease outward to the joint between the diaphragm 82 and the surface 83 against which it runs; and the air pressure leaking by diaphragm 84 drives the grease thru the joint and thru the journal into chamber 81 again.

In assembly the member 84 is loosely mounted on the shaft before the diaphragm 82 is driven on. The shaft 25, carrying the driving member 24 is then inserted into the case and the diaphragm 84 driven onto the boss 85 by means of proper tools inserted thru the holes 87, after which the holes may be closed by plugs.

The driving member 24 drives the annular rotor by means of keys 88. The driving member has an easy sliding fit over the annular rotor in order that it may assume a balanced position between the side walls, regardless of the influence of the shaft 25.

I find the liquid seal of great importance in securing efficient action. Lubricating oil makes a tight seal but if it by chance gets into a rubber tire it destroys the rubber. Glycerine is greatly superior due to its lack of rapid corrosive action on rubber and is efficient to maintain tightness between moving parts. And not having the capillary adhesion to metals like oil, provides a more efficient film for easy running.

The reservoir 36 provides storage space to hold enough glycerine for a long period of use. The efficiency of my centrifugal separator prevents it from being wasted.

If ordinary oil is used, the separator is also most efficient particularly as oil at high pressure and low temperature contains a negligible amount of vapor. In the form of mist it is easily separated from air by baffling.

In action the air is whirled by the baffle plates around in the cylinder 54 and deposits its liquid on the baffles as it issues thru the perforations of one baffle plate and strikes the flat portion of the next one. The deposited liquid is whirled outward to the outer diameter of the cylinder 54 while the air tends inward, being lighter than oil. The liquid passes over the outer edges of the baffle plates and thru the holes 55ª and port 56ª into the reservoir 36. It returns to the rotors thru the passageway and oil port 35.

The air of course is sucked in thru the intake port 38, is whirled around in the rotors as the chambers first open and then close, is compressed while the chambers are closing, and discharged out of the port 40 into the baffle plate separator, and the separated air drawn off thru the exit pipe connection 61.

While I have described my invention in the way I prefer to make it I do not wish to be understood as limiting it to the specific construction shown since it may be used in many other forms of construction. The many features of novelty may be used alone or in combinations of various degrees with each other.

In this specification and in the claims I have used the expression or made the statement that the curves of envelopment upon or by which the teeth of the rotors are formed are generated by the tooth-forms during "relative angular motions" inversely proportional to the number of teeth. I mean to indicate by this expression that although the two peripheries travel at the same circumferential speed at the place where the teeth engage, they necessarily vary in relative angular displacement, moving as they do on different radii and one having one less tooth than the other; in the ratio of 8:9, for example, the larger rotor would not have com-

pleted its revolution by 40° when the smaller has made a complete turn. Necessarily this makes the one set of teeth slide upon the other; and it is one of the main objects of my invention so to form the curves of development of the two sets of teeth that during the working range (which may be of either using or of delivering power) the two sets of teeth shall continuously maintain sliding contact due to the relative angular displacement specified; on the one hand not permitting any opening or "relieving" between the teeth of the two rotors through which anything approaching commercial pressures would be immediately dissipated; and on the other maintaining the contact substantially as continuously and with substantially as slight friction as between plane surfaces, so that the teeth burnish one another as they engage and part, and can be efficiently lubricated to form a film which is not removed either by the contact of the metal or by the fluid pressure. In practice, after the teeth have become burnished no substantial lubrication between them is required.

What I claim is:—

1. In a rotary mechanism dealing with gas pressures having intake and outlet passageways, two rotor members having tooth divisions forming between them chambers which open and close by relative rotation to perform pressure functions between said passageways, one rotor within and eccentric to the other and having one less tooth division, contours on said tooth divisions providing said pressure functions and relative angular motion, said contours of each rotor conforming to curves of envelopment outlined by the successive positions of a tooth form of the other rotor during relative angular motions which motions are those of two circles tangent internally, each in turn carrying the appropriate tooth form, and the two circles having diameters proportional to the numbers of tooth divisions of the said rotors that are to conform thereto, the angular velocity of the outer circle bearing to that of the inner circle the inverse ratio of the numbers of tooth divisions of said rotors, said rotor contours having continuous travelling engagements in the full mesh region between said passageways, and having a plurality of continuous travelling engagements between said contours on the open mesh side between said passageways, means for providing a liquid seal, end walls on opposite ends of said rotors fixed to each other directly engaging the ends of said rotors and enclosing said rotor chambers at said ends, to balance said rotors and end walls with relation to each other with a floating relation free of endwise pressure, and a driving connection to the outer rotor.

2. The combination claimed in claim 1 having the driving connection to the outer rotor comprising a driving shaft engaging the outer rotor and centering it on its axis.

3. The combination claimed in claim 1 having said end walls fixed, and said driving means comprising a shaft extended over one of said end walls, and centering and driving the outer rotor, one of said fixed end walls being rigidly attached to the other by a shaft through the inner rotor.

4. The combination claimed in claim 1 having said end walls connected together through the inner rotor, one wall having the outlet passageway and the other wall having the inlet passageway.

5. The combination claimed in claim 1 combined with a centrifugal separator connected to the outlet passageway, said rotary mechanism being in a casing, with a mechanical connection to one of said rotors to rotate said separator to separate said liquid seal from said gas.

6. The combination claimed in claim 1 combined with a centrifugal separator connected to the outlet passageway, said rotary mechanism being in a casing with mechanical connection to one of said rotors to separate centrifugally said liquid seal from said gas, and perforated baffle plates in said separator to help separate liquid seal from gas.

7. The combination claimed in claim 1 combined with a centrifugal separator mechanically connected to said rotary mechanism to separate said liquid seal from said gas, said separator being connected in the outlet passageway.

8. The combination claimed in claim 1 in which the driving means comprises a shaft extended over one of said end walls and centering and driving the outer rotor, one of said fixed end walls being rigidly attached to the other by a shaft through the inner rotor, and a passageway to said rotor chambers opened through said rotor centering shaft.

9. In a rotary mechanism dealing with fluid pressures and having intake and outlet passageways, two rotor members having tooth divisions forming between them chambers which open and close by relative rotation to perform pressure functions in connection with said passageways, one rotor within and eccentric to the other and having one less tooth division, contours on said tooth divisions providing said pressure functions and providing relative angular motions, said contours of each rotor conforming to curves of envelopment outlined by the successive positions of the tooth form of the other rotor during relative angular motions which motions are those of two circles tangent internally, each in turn carrying the appropriate tooth form, and the two circles having diameters proportional to the numbers of tooth divisions of the said rotors that are to conform thereto, the angular velocity of the outer circle

bearing to that of the inner the inverse ratio of the numbers of tooth divisions of the said rotors, said rotor contours having continuous travelling engagements between the tooth divisions separating said passageways both in the full mesh region and open mesh region during said rotation, and having portions of said contours not providing said pressure functions and relative angular motions varied or relieved.

Signed at New York, in the county of New York and State of New York, this 1st day of November A. D. 1921.

MYRON F. HILL.