June 12, 1923. 1,458,565

W. S. WESTON

HYDRAULIC CLUTCH

Filed May 7, 1920 5 Sheets-Sheet 1

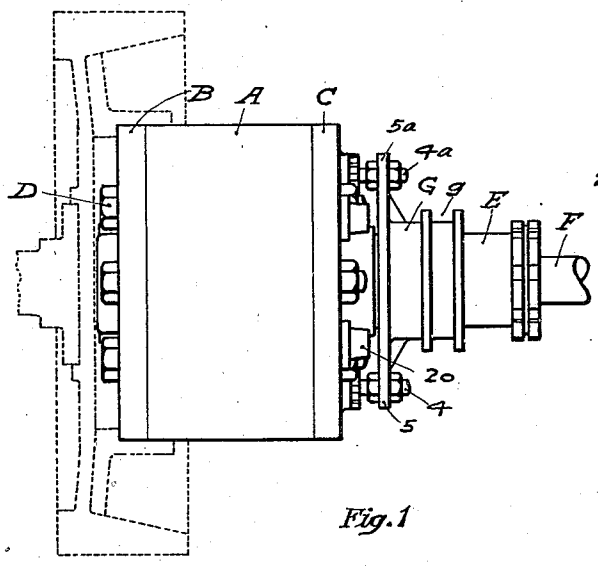


Fig. 1

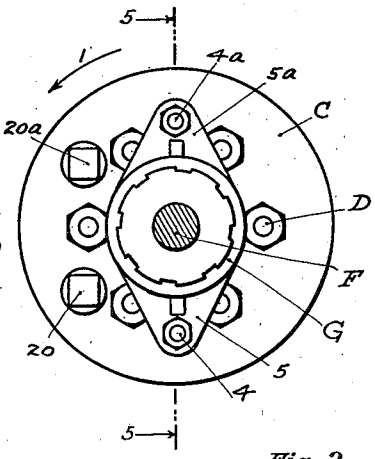


Fig. 2

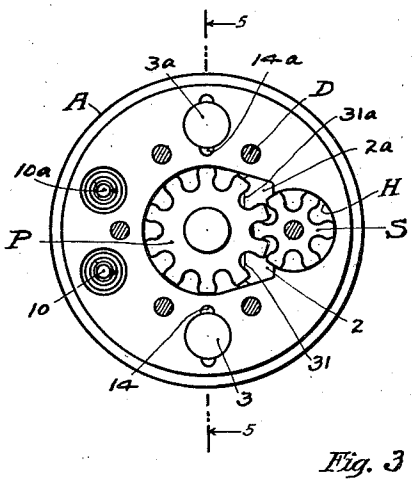


Fig. 3

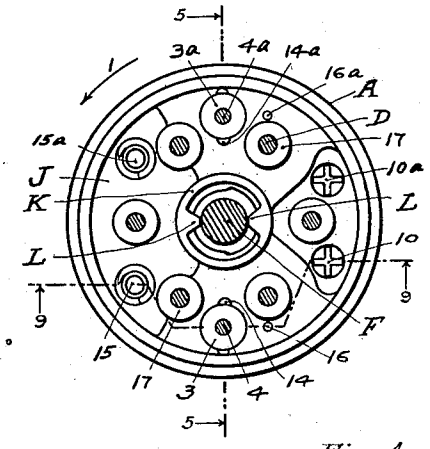


Fig. 4

Witnesses:-
E. M. Giles
M. Spaulding

Inventor:-
William S. Weston
by Fisher, Fowle, Clapp & Soame
Atty's.

June 12, 1923. 1,458,565

W. S. WESTON

HYDRAULIC CLUTCH

Filed May 7, 1920 5 Sheets-Sheet 2

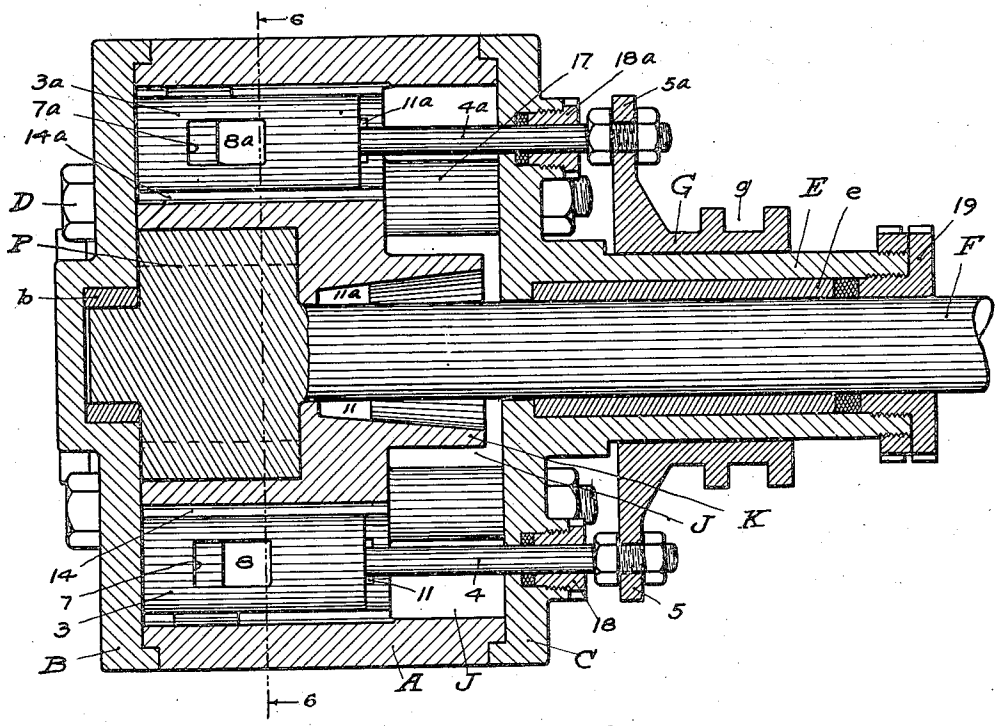


Fig. 5

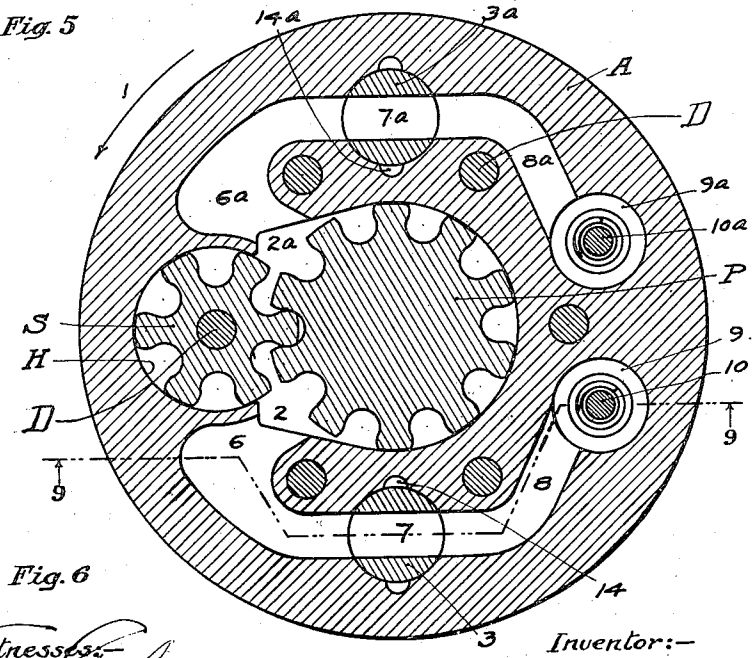


Fig. 6

Witnesses:—
E. M. Giles.
M. Spaulding

Inventor:—
William S. Weston
by Fisher, Towle, Clapp & Soane
Atty's.

June 12, 1923.

1,458,565

W. S. WESTON

HYDRAULIC CLUTCH

Filed May 7, 1920

5 Sheets-Sheet 3

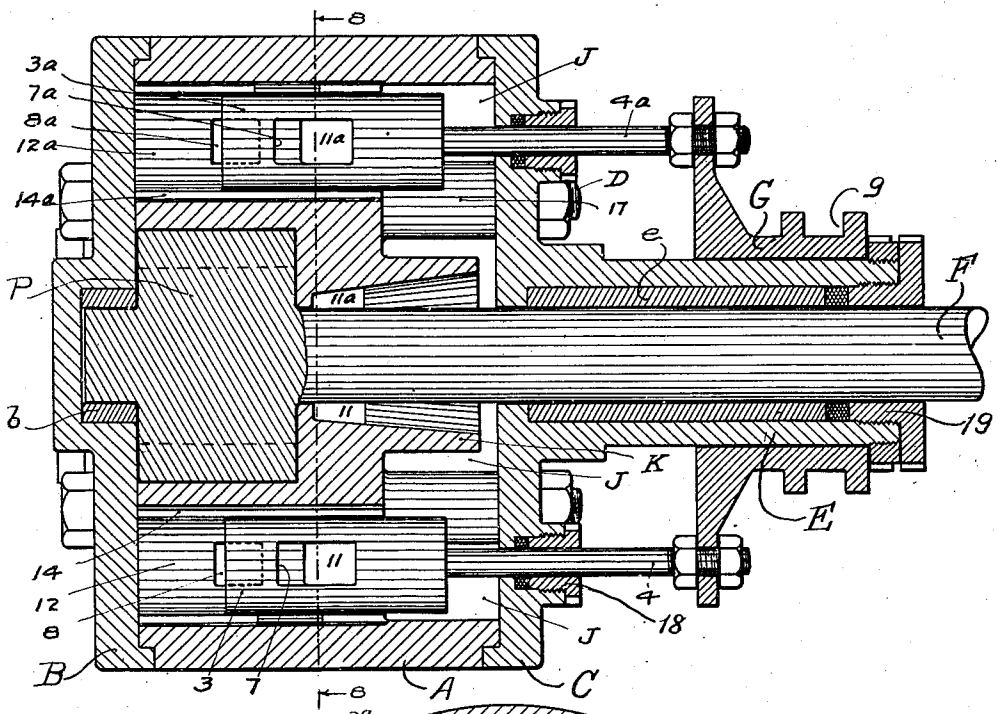


Fig. 7

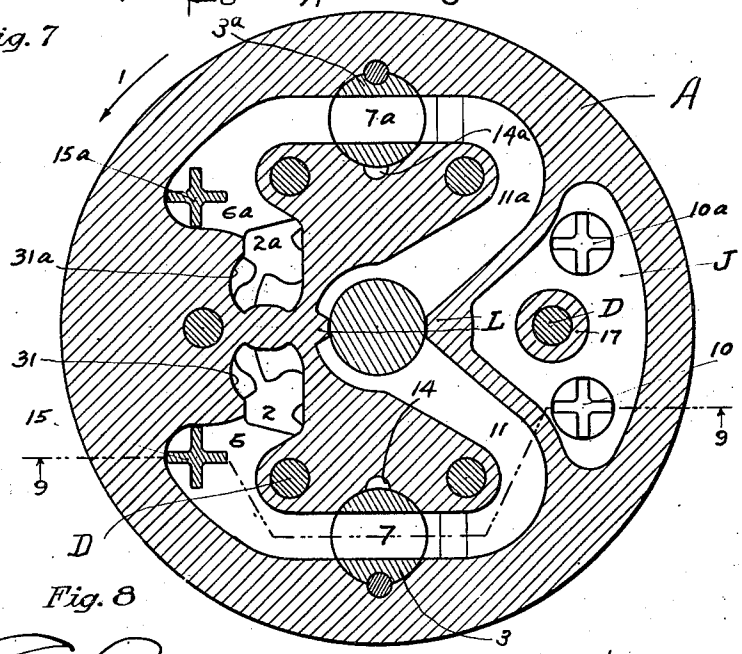


Fig. 8

Witnesses:-

M. Giles

M. Spaulding

Inventor:-

William S. Weston

by Fisher, Poole, Clapp & Sawyer

Atty's.

June 12, 1923.

1,458,565

W. S. WESTON

HYDRAULIC CLUTCH

Filed May 7, 1920

5 Sheets-Sheet 4

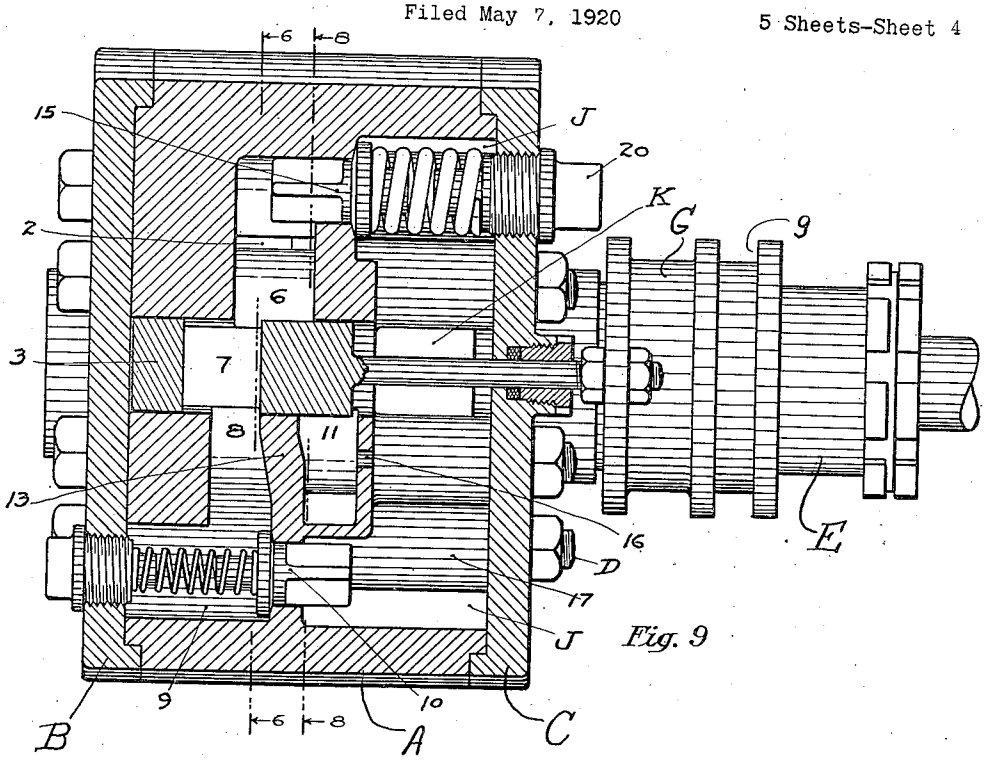


Fig. 9

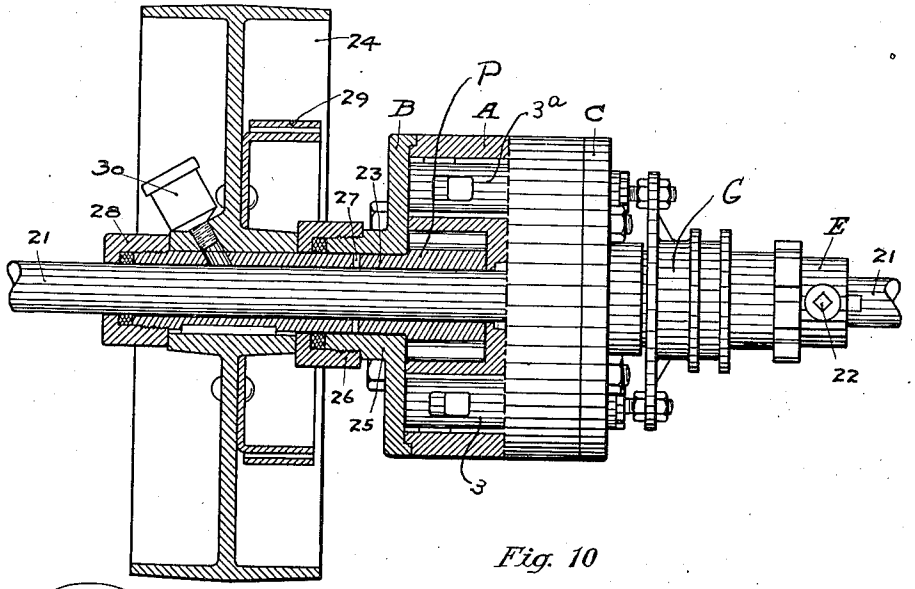


Fig. 10

Witnesses:-

E. M. Giles

M. T. Spaulding

Inventor:-

William S. Weston

by Fisher, Fowle, Clapp & Soame

Atty's.

June 12, 1923. 1,458,565

W. S. WESTON

HYDRAULIC CLUTCH

Filed May 7, 1920 5 Sheets-Sheet 5

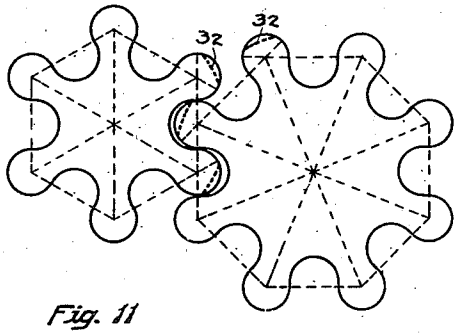


Fig. 11

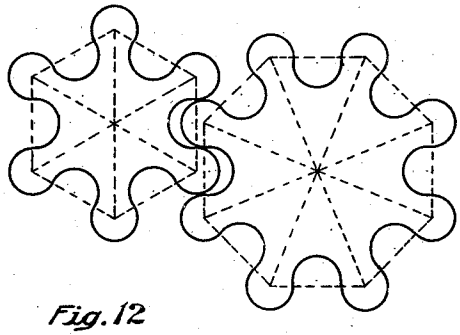


Fig. 12

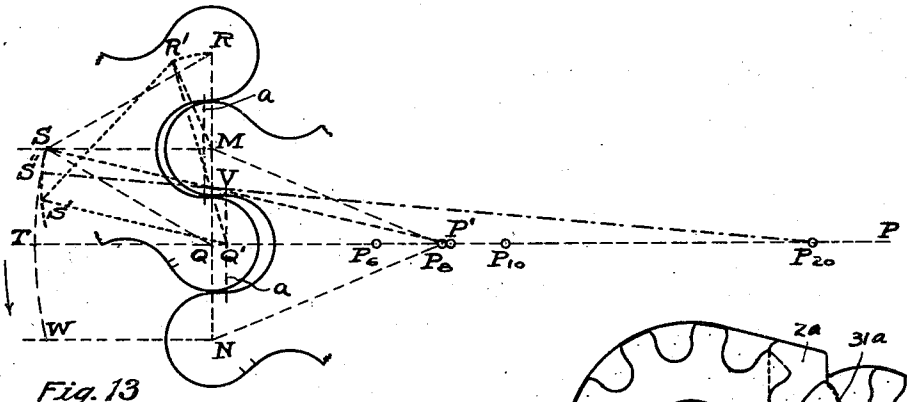


Fig. 13

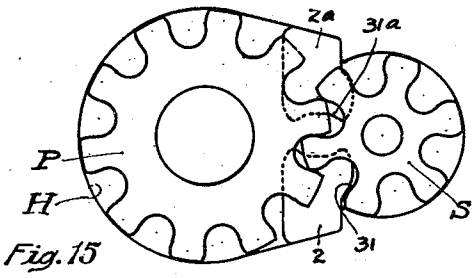


Fig. 15

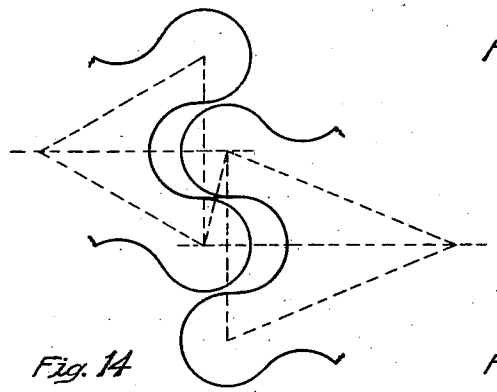


Fig. 14

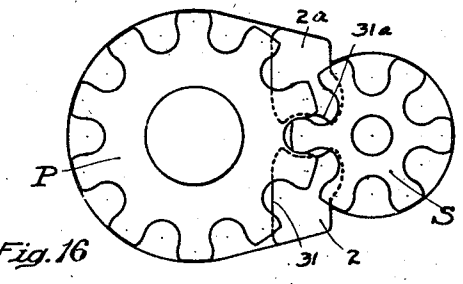


Fig. 16

Witnesses:-

E. M. Gilles

M. O. Spaulding

Inventor:-

William S. Weston

by Fisher, Powle, Clapp & Soame

Attys.

Patented June 12, 1923. 1,458,565

UNITED STATES PATENT OFFICE.

WILLIAM S. WESTON, OF DALLAS, TEXAS.

HYDRAULIC CLUTCH.

Application filed May 7, 1920. Serial No. 379,654.

To all whom it may concern:

Be it known that I, WILLIAM S. WESTON, a citizen of the United States, residing in the city of Dallas, county of Dallas, and State of Texas, have invented certain new and useful Improvements in a Hydraulic Clutch, of which the following is a full, clear, and exact description.

My invention relates to improvements in that class of clutches in which the opening and closing operation is obtained by a manipulation of the valves governing the flow of a liquid medium through the elements of a hydraulic pump, and in which the pump casing, with the elements normally stationary considered as a pump, constitute one member of the clutch, and the pump operating shaft with its attached parts constitute the other member. Many types of such a clutch have been devised. The primary improvements of my present invention apply preferably to that type in which the pump elements acting directly on the liquid medium are a pair of intermeshing spur gears.

The object of my invention is the attainment in a clutch of this character of a high rate of efficiency in both the open and closed position together with greater simplicity in construction and a more perfect control in operation.

All the essential parts in the construction of a hydraulic clutch of the class described embodying the new and improved features whereby I attain my object, together with other novel features of auxiliary value are hereinafter fully described and clearly illustrated in the accompanying drawings, which form a part of these specifications. The elements of novelty are particularly pointed out in the claims.

Fig. 1 is a side elevation of the clutch illustrating how it may be attached to the motor fly wheel and used in the place of the cone or friction disc clutch of an automobile or motor driven vehicle. The fly-wheel and means of attachment thereto not being an essential element in this invention, are shown in dotted outline in this figure only. Fig. 2 is an end elevation, as seen from the right of Fig. 1. Fig. 3 is an end elevation of the main clutch casing as seen from the left of Fig. 1, with the cover plate on the end removed. Fig. 4 is a similar view, as seen from the right of Fig. 1, with the cover plate on that end removed. In Figs. 3 and 4 the bolts and parts that pass through the cover plates are shown in section. Fig. 5 is a vertical longitudinal section of Fig. 1, drawn to a larger scale as viewed on the line 5—5 of Figs. 2, 3, and 4, in the direction indicated by the arrows. Fig. 6 is a transverse section of the main clutch casing on the line 6—6 of Figs. 5 and 9. Fig. 7 is the same as Fig. 5 with the controlling valves shifted to the open position. Fig. 8 is a transverse section of the main clutch casing on the line 8—8 of Fig. 7, and on the same line of Fig. 9, assuming the controlling valve shown in that figure to be shifted to open position. Fig. 9 is a longitudinal section through one side of the main casing and cover plates on the line 9—9 of Figs. 4 and 6, showing one controlling valve in the closed position. The Fig. 9 section is also indicated by the line 9—9 of Fig. 8, which, as stated above, is a transverse section with the controlling valves open. Fig. 10 is a side elevation similar to Fig. 1 with the left hand half of the main casing shown in vertical longitudinal section, illustrating how the clutch may be utilized to connect a pulley with a driving line shaft. Figs. 11, 12, 13 and 14 are mathematical diagrams used to illustrate the intermeshing relation of spur gears having teeth with cylindrical contact surfaces. Figs. 15 and 16 are enlarged views of the gear recess taken from the same point of view as Fig. 3 and used to illustrate more clearly a detail essential to the practical operation of the clutch when constructed with that type of gear. The same letters and numerals of reference marked on the drawings refer to the same parts in the several views.

My improved hydraulic clutch, first described in general terms, comprises in its two fundamental elements a rotating casing constituting the driving member and a shaft or its equivalent, concentric with the casing and constituting the driven member.

Within the driving casing immediately under the left hand cover plate, as illustrated, is a close fitting figure 8 recess containing a pair of intermeshing spur gear pump gears. One of these gears, called the primary gear, is concentric with the casing and is attached to or made integral with the driven shaft. The other gear, called the secondary gear, is journaled in the casing, preferably on one of the construction bolts, and is free to be rotated in a planetary movement about the primary gear when such ro-

tation is not prevented by a liquid medium which may be confined in the interdental spaces of the gears and within the gear recess. In operation a free planetary movement is the condition for the open clutch while an inter-lock of the gears with the liquid medium is the condition for the closed clutch.

Within the casing immediately beneath the right hand cover plate, as illustrated, is a relatively large storage chamber containing a liquid medium, preferably a moderately viscous lubricating oil.

Within the cast body of the casing and leading between the figure **8** gear recess on one end and the storage chamber on the other are a series of passages and valves providing for and governing the flow of the liquid medium to and from the gear recess. The passages are so arranged and the valves may be so controlled that one or the other of the two operating conditions above referred to may be obtained. First, for the operating closed position of the clutch the gear recess is maintained full of the liquid medium and the passages are so connected together and closed against a return of the liquid to the storage chamber, that the gears will be interlocked for all normal loads for which the clutch is designed, or second, for the operating open position of the clutch the passages are so connected and opened that the gear recess is maintained empty of any appreciable amount of the liquid medium, thereby permitting a free planetary movement of the secondary gear about the primary.

The arrangement of the passages and the controlling valves for this second or operating open position of the clutch is an important novel feature of my invention. In it I take advantage of the centrifugal action on the liquid medium in the rotating casing to assist in clearing the gear recess and interdental spaces of all liquid. By this construction I am able to use a slow running viscous liquid with a low leakage ratio for the operating closed position, and yet provide for a high rate of speed of the driving casing with only a minimum of drag on the driven shaft when in the operating open position. In this open position the driven shaft would be held by a brake or by its own inertia or that of the machinery attached to it.

In the preferred form of construction of my hydraulic clutch and in combination with the novel feature just described, the primary and secondary gears are made as illustrated, with teeth having cylindrical contact surfaces instead of involute or cycloidal surfaces, as in the standard machine gear. This type of gear enables me to obtain and maintain more readily a liquid tight joint between the gears at the lines of intermesh and between the gears and the peripheral walls of the recess, so that in the closed position of the clutch I may reduce the leakage ratio of a viscous medium to an inappreciable minimum and secure a high percentage of efficiency in the transmission of the rotative movement. The characteristics of gears with cylindrically shaped teeth and the method for determining their dimensions and setting will be described in detail at the end of these specifications. Other novel features of improvement will be mentioned in the following detailed description.

The driving casing A is a single casting in which the recesses, passages, valve seats and storage chamber, partly formed by coring, may be finished in a lathe or boring mill. The left end is fitted with a cover plate B and the right end with a cover plate C. The casing and cover plates are held together with fluid tight joints by means of bolts D. The cover C is provided with a hub E. Extending through the hub and concentric both with the hub and casing is the driven shaft F. The hub is fitted with an interior lining of anti-friction metal *e* surrounding the shaft to provide for wear when the casing is rotating and the shaft is stationary in the operating open position of the clutch. On the exterior of the hub is a sleeve G, adapted to have a longitudinal movement on the hub and thereby be utilized to change the clutch through valve mechanism yet to be described from the opened to the closed position, or vice versa. The movement of the sleeve may be obtained by means of a shifter fork, fitted to the groove *g*. The shifter fork is a well known mechanism and need not be illustrated or described.

In the driving casing immediately under the cover plate B is the figure **8** recess H, containing in a liquid tight fit on peripheral and face surfaces the primary gear P and the secondary gear S. The fit is such that when the interdental spaces and any small chambers connected to opposite sides of the recess are filled with a liquid and the outlets from these opposite sides are closed, any intermeshing movement of one gear on the other will develop compression on one side of the recess and suction on the other. The primary gear P is mounted on and preferably made integral with the shaft F. The shaft is extended beyond the gear so as to have a short bearing in a socket in the cover plate B, which is preferably lined with antifriction metal *b*. The secondary gear S is mounted to rotate freely on one of the structure bolts D.

In the driving casing under the cover plate C is a chamber J for holding in storage a quantity of liquid sufficient for operation in both the operating opened and

closed positions. The central portion of this chamber around the shaft F is separated from the radially outer portion by a cylindrical ring formation K of the main casing casting which does not quite reach the cover plate C. The two portions of the chamber are therefore connected by a narrow annular opening around the edge of the ring. The interior surface of the ring is conical and increases in diameter toward the edge. The construction is such that when the casing is rotating, any liquid in the central portion of chamber J will flow by centrifugal action toward and around the edge of ring K into the outer portion. The central portion of the chamber is divided into two compartments by oppositely positioned partitions L, L, extending from the inner surface of the ring toward the shaft F and coming sufficiently close to the shaft to form a practically liquid tight joint without taking a bearing on the shaft. The purpose of these partitions will be stated later on.

Before going into the description of the several passages and valves, it should be stated that the preferred form of my invention, as illustrated, is operable with the driving casing rotating in either direction. In this construction the passages and valves are made in duplicate and symmetrically opposite sets. To avoid confusion in the following description, the rotation of the casing will be considered as counter clockwise when viewed from the right of Fig. 1 and as seen in Figs. 2, 4, 6 and 8, the direction being indicated by the arrow 1. Referring to Fig. 6, with rotation counter clock-wise, any liquid or fluid in the gear recess will be subject by the relative gear movement to compression on one side in the region marked by the numeral 2 and to suction in the opposite region marked by the numeral 2 with a suffix small "a". Using this method of distinguishing the similar opposite parts, the passages and valves associated with the compression side of the gear recess are referred to by numerals while those associated with the suction side are referred to by the same numerals with the suffix small "a".

Referring to Figs. 5, 6 and 7, the parts 3 and 3^a are piston valves, having stems 4 and 4^a extending through the cover plate C and attached to wings 5 and 5^a on the shifter sleeve G. Fig. 5 shows these valves in the closed position of the clutch and Fig. 7 in the open position, as will be understood after a little further consideration. Passages 6 and 6^a lead from the gear recess to the piston valves 3 and 3^a respectively. Ducts 7 and 7^a extending through the piston valves connect passages 6 and 6^a in the closed position of Figs. 5 and 6, with passages 8 and 8^a that lead into small chambers 9 and 9^a, which are provided with check valves 10 and 10^a, that close openings into the storage chamber J. The check valves are preferably held to their seats by light springs and are so designed as to open only to a suction action from the gear recess or from pressure of the liquid in the storage chamber under centrifugal action.

A study of Fig. 6 made now following the description given in the preceding paragraph discloses the operation of the device in the closed position. Any tendency of the secondary gear S to move counter clock-wise about the primary P draws in liquid from the storage chamber by suction action through the check valve 10^a and maintains suction regions or passages 2^a, 6^a, 7^a, 8^a and 9^a full while the check valve 10 prevents discharge from the compression regions 2, 6, 7, 8 and 9. It is clear, therefore, that for this position with a practically incompressible liquid the secondary gear cannot have a planetary movement about the primary and that the primary with the shaft F must rotate with the casing. It should be noted that the check valves 10, 10^a, the storage chamber ends of which are shown in Figs. 4 and 8, are located near the periphery of the driving casing which is always rotating when in operation and will therefore always be submerged by the liquid in the storage chamber. Any liquid that may under compression leak back around the piston valve 3 or check valve 10, or around the shaft F will be replenished immediately and continuously through the check valve 10^a. It is also well to note in this connection that, with the suction and compression regions just described full of an incompressible liquid, the direction of rotation of the driving casing may be reversed without any material amount of lost motion between the casing and the driven shaft.

Referring now to Figs. 7 and 8, with the piston valves in the open clutch position, the ducts 7 and 7^a still register with passages 6 and 6^a from the gear recess but lead into another set of passages 11 and 11^a that open into the bottom of the two halves of the central portion of the chamber J. It can be seen in Fig. 7 that the ducts 7, 7^a are entirely out of register with the passages 8, 8^a, shown partly in dotted outline, which lead to the check valves. The suction of the gears therefore cannot draw liquid through the check valves and can only be replenished by drawing liquid from the central portion of chamber J by way of the passage 11^a. As already stated, this central portion during operation is kept practically empty due to the centrifugal action on the liquid. By this novel construction I am able in the operating open clutch position of the piston valves to maintain the gear recess and the adjoining passages practically free of the liquid medium, securing a free planetary movement of the

secondary gear about the primary and permitting the full rotary movement of the driving casing at its normal speed while the shaft to be driven may be standing still or be locked against rotative movement. Any liquid that may be on the compression side of the gears has a free passage into the central portion of the storage chamber from which it will be discharged by centrifugal force. It will now be apparent that the function of the radial walls or partitions L, L, dividing the central portion of chamber J into two halves, is to prevent the liquid discharged from the compression side going back by centrifugal action as well as suction action into the suction side.

The approximate maximum quantity of liquid that may be used in the device will be just enough to fill the outer portion of the chamber J when the gear recess, the connected passages on the compression and suction sides, and the central portion of the chamber are substantially empty, as described in the preceding paragraph.

It is to be noted that normally the spaces empty of liquid will contain air or gases from the liquid at atmospheric pressure. When the clutch is operating in the open position, the planetary movement of the secondary gear about the primary causes the device to act like a positive blower, driving a current of air or gas from the compression side through the two halves of the central chamber and back into the suction side. This current is depended upon to blow out the excess of any liquid that may accumulate under centrifugal action in the vicinity of the duct 7 in the piston valve on the compression side, and to suck in any that may accumulate in the vicinity of duct 7^a on the suction side. This air current action is utilized in a provision hereinafter described for keeping the gears lubricated when running in the open position for a long period of time.

An understanding of the important action that takes place during the period of shifting the piston valves from the open to the closed position, or vice versa, may be obtained from the study of Fig. 9 in conjunction with Fig. 7. Fig. 9 is a longitudinal section through the piston valve 3 and the associated passages on the compression side of the gears, as indicated by the line 9—9 in Figs. 6 and 8. In this figure the piston valve is in the closed position and a clear opening is shown from region 2 of the gear recess through passage 6, duct 7, passage 8 small chamber 9, to the check valve 10. When the piston valve is pulled out for the open position, as shown in Fig. 7, it is moved just enough to close the opening from passage 6 to passage 8 without making an opening from the passage 6 into the space 12 under the piston. In this position from which the section of Fig. 8 is taken, a clear opening will be had from region 2 through passage 6, duct 7, and passage 11 to the central portion of chamber J.

The especial feature now to be noticed is that the duct 7, through the valve, is wider than the partition wall 13 between passage 8 and passage 11 and that for a period in the movement of the valve from one to the other of its two operating positions the passage 6 from the gear recess will be opened into both passages 8 and 11. The same condition also develops simultaneously in the passages on the suction side of the system. The result during this intermediate stage will be that both air and liquid will be drawn in on the suction side and discharged into the central chamber from the compression side. As the movement for closing the clutch goes on through this stage, the inward flow of liquid on the suction side will be facilitated while the discharge from the compression side will be retarded. The final operation, more or less gradual at the will of the operator, of closing off the compression side from the central chamber, is a feature of paramount importance, distinguishing the hydraulic clutch from all other types. The manipulation of the piston valves through this middle position constitutes the means, entirely free of uncertain or variable friction strains, whereby the full rotating movement of the driving casing may be very gradually developed in the driven shaft in the process of overcoming the inertia of the parts or the resistance of the load.

Provision is made to avoid the sudden closing of the clutch by an inexperienced operator. When in the open position the space 12 at the bottom of the piston socket fills with liquid coming finally through the check valve and the passage 8, see Fig. 7, or gradually down through a longitudinal groove 14 in the wall of the valve socket, leading into the storage chamber. When the valve is started toward the closed position, the liquid in space 12 must all go out through the groove 14, as none of it even in the beginning of the movement can go back into passage 8 against the check valve. The retardation of movement, due to the necessity of forcing a comparatively large quantity of liquid through the small groove 14 is the well known dash pot effect and is well understood. On the other hand, with the development of a momentary vacuum in the dash pot space, the piston valves may be moved to the open clutch position suddenly in case of emergency.

A number of additional features of improvement, all of auxiliary value to my new hydraulic clutch, will now be described. Referring to Fig. 9, 15 is a spring retained relief valve, closing an opening from passage 6 direct into the storage chamber. The in-

ner ends of relief valve 15 and its duplicate 15ª are shown in cross section in Fig. 8 projecting into passages 6 and 6ª respectively. A very heavy spring is used and its tension is so adjusted that it will allow the valve to open only when the hydraulic pressure developed on the compression side, by the transmission of the power load through the clutch, is increased by an excess load or otherwise to a point when strains in the parts of the mechanism would be greater than those for which it was designed. This relief valve would be of especial value in the application of my clutch to the driving of machinery subject to intermittent loading and sometimes dangerous overloading. My improved clutch may be used to advantage with constant speed motors with the relief valve so adjusted that the motor will continue running at its normal speed, delivering its maximum power during moments of overloading. This use is especially applicable in driving machines having heavy fly wheels designed to overcome sudden overloads, as with the relief valve properly adjusted the full power of the motor is still back of the fly wheel during the period the latter may be giving up some of its momentum.

In the application of my clutch to the driving of various kinds of machinery, conditions may frequently exist where the device will be run for a long continuous period in the open position. For such a condition provision is made for insuring the lubrication of the gears by placing small leak holes 16 and 16ª, leading from the storage chamber into passages 11, 11ª. The liquid that leaks into passage 11ª will be sucked through the gears and with the portion that leaks into passage 11 will eventually be blown back again into the storage chamber. The amount of liquid provided by these leak holes will be relatively small and will not in any way interfere with the free planetary movement of the secondary gear.

Lubrication of the bearing between the shaft F and the anti-friction metal lining of the hub E for conditions of long operation in the open position may be obtained by providing the usual journal oil grooves leading from the storage chamber.

The structural joints of the device, such as those between the main casing and the cover plates and around the bolts, are made liquid and air tight by means of a paint or other joint cement. The structural bolts D, where they pass through the storage chamber, are preferably surrounded by metal columns 17, which are cast with the main casing and help to support the cover C. Stuffing boxes 18, 18ª, are provided where the piston valve stems pass through to the exterior, and a stuffing box 19 is provided for the joint between the end of the hub E and the driven shaft F. It should be noted that the stuffing box joints are not subject to the hydraulic pressure developed within the mechanism and that leakage to the exterior will be a negligible quantity. The liquid medium may be renewed from time to time by removing the plugs 20 and 20ª located over the relief valves, 15 and 15ª.

My hydraulic clutch, as illustrated, and thus far described, is universal in respect that it is operable when the driving member is rotating in either direction and that for the closed position it constitutes a positive union between the members. For instance, this construction would prevent the driven member from racing under any exterior impulse at a greater speed of rotation than that of the driving casing. As an example, in the casing of an automobile running down grade under the action of gravity, the clutch would still be in operation after the gas supply to the engine had been cut off, with the automobile driving the dead engine against the compression in its cylinders. It should be noted, however, that for driving a shaft always in one direction under conditions where provision against racing is unnecessary, a clutch embodying the fundamental feature of my device, whereby I maintain a gear recess liquid full for the closed position and substantially liquid empty for the open position can be made with fewer parts. The check valves may be omitted. That is, the check valve on the compression side may be permanently closed while that on the suction side may be permanently opened. In that case, the operation would be dependent entirely on the two piston valves. In the operating closed position of the clutch the suction side of the gears would then have an opening direct into the storage chamber always submerged by liquid, while the compression side would be closed to the storage chamber, thereby maintaining the gear recess full of liquid and locking the gears against intermeshing movement. On the other hand in the operating open position of the clutch, both the suction and the compression sides would have an opening into the central portion of the chamber which discharges by centrifugal action into the outer portion thereby resulting in the substantial clearance of all liquid from the gear recess and permitting the free intermeshing movement of the gears. For the one way construction an auxiliary relief valve, if used, would be needed only on the compression side of the gears.

It is obvious that many changes in the form and adaptation of my invention may be made without any modification of the fundamental novel features on which its operation is based. Such a change is illustrated in Fig. 10. In this particular adaptation the driving casing is keyed to a line shaft 21 and held in position by a set screw

22 in the end of the hub E, while the driven member carrying the primary gear is made in the form of a sleeve 23, running on the line shaft and extending through the left hand cover plate B. A pulley 24, or any other means for connecting up to the machinery to be driven when the clutch is closed, may be keyed to the outer end of the sleeve. The cover plate B at the opening for the sleeve is provided with a hub 25 and a stuffing box cap 26 to prevent leakage to the exterior. Hydraulic pressure against this stuffing box is relieved by means of oil holes 27 leading through the sleeve to the usual oil grooves that may be cut in its interior surface so as to lead back toward the central chamber. A stuffing box 28 is also provided for the outer end of the sleeve covering the exterior joint between the sleeve and the line shaft 21. A brake band 29 is adapted to hold the pulley stationary when the clutch is in the open position. Means are provided, such as the grease cup 30, for lubricating the interior of the sleeve when the line shaft is running with the clutch in open position and the sleeve, together with the pulley, are held by the brake. The valve construction and all other details pertaining to the manipulation and operation are in no way changed.

As stated in the preliminary description of my invention, a primary object in the choice of the special gears having cylindrical surfaces of contact on opposite sides of the teeth as shown, instead of cycloidal or involute surfaces is to provide a construction that will give a minimum of leakage between the gears for a maximum of liquid to be displaced. It is also my object to hold the driving casing to as small a diameter as practicable for the power to be transmitted. These objects are best obtained by the use of gears with a small number of large teeth.

The cylindrical contact surfaces may be defined as such that all points in these surfaces on both sides of any tooth are a constant radial distance from a line parallel with the axis of the gear, constituting the axis of the tooth and hereinafter called the center of the tooth. In the intermeshing movement the opposite cylindrical surfaces of a tooth have a rolling sliding contact with short parallel faces of the interdental space of the companion gear. As one tooth comes out of such double contact the incoming tooth of the companion gear comes into double contact.

I am aware that cylindrical teeth have been used before, but in only one of a pair of gears, as in the squirrel cage pinion members of many clocks. I am also aware that engineers and draftsmen have illustrated intermeshing gears with more or less cylindrically shaped teeth on both gears, as for instance in the patent to E. Neff, et al., No. 275844, April 17, 1883, for a rotary double piston pump. I am not aware, however, that intermeshing gears with teeth in each gear, each provided on opposite sides with true cylindrical surfaces adapted to have a rolling sliding contact between short parallel faces in the interdental space of the companion gear have ever been put into any practical use, or have ever before been used in the construction of a hydraulic clutch. Neither am I aware that the intermeshing relation of such gears and their limitations have ever been formulated and put into such shape as to enable the mechanic to predetermine the correct practical spacing for the gear centers after having determined on the size of the teeth and the number in each gear.

In the beginning of this description it is to be noted that a pair of gears with cylindrical teeth cannot both have a relative uniform rotative movement about their axes, a prime essential governing the manufacture of gears with cycloidal or involute contact surfaces. If one such gear with cylindrical teeth has a uniform rotation movement the companion gear will have a motion accelerated and retarded alternately at the passing of every tooth, setting up a rhythmic vibratory action. This feature makes the use of the cylindrical tooth in both gears practically prohibitive in all construction where power is to be transmitted or work done by the actual intermeshing movement of the gears, as in all machine gears and in spur gear pump gears operating as pumps. The rhythmic acceleration and retardation would necessarily be transmitted to the load, or forward or backward in the machinery, unless absorbed in some sort of cushion couplings.

The distinguishing characteristic in my use of gears with cylindrical teeth is found in the fact that when transmitting power or doing work the gears are locked against intermeshing movement. Only when the clutch is operating in the open position is there an intermeshing movement, and for many uses, as in automobiles, that will be for only a small per cent of the total time of operation. Referring to Fig. 6, when the driving casing is rotating at a uniform speed with the primary gear P held stationary, the secondary gear in its planetary movement has the revolution on its own axis retarded as a tooth enters an interdental space of the primary and accelerated as it comes out. In this operation only the inertia of the secondary gear has to be overcome, and as its mass is very small relative to that of the balance of the machinery, both ways from the clutch, the only effect

except for a rhythmic impulse will be a slight increase in the wear between the teeth. The statement that the gears are locked against intermeshing movement when the clutch is closed and doing work is to be understood in the practical interpretation of the expression. There will necessarily be a perceptible movement sufficient to follow up the leakage from the compression side of the gears. This follow up movement constitutes the only cause for mechanical wear between the teeth under load conditions and insures a uniform distribution of that wear.

In order that the mechanical engineer may prepare the necessary shop drawings and give dimensions for this type of gear, the following theoretical discussion is given and illustrated by diagrams, Figs. 11 to 14. In the construction and in this discussion I have considered a combination having only six teeth in the smaller secondary gear. It will be noted, however, that the discussion would be the same in principle if the secondary had any other numbers of teeth.

Fig. 11 is a diagram of an eight tooth primary and a six tooth secondary gear. The centers of the teeth in each gear are connected together and to the center of the gear by lines forming a polygon made of a number of isosceles triangles. It can be demonstrated that two gears so constructed with six and eight teeth may be set to a theoretical minimum distance center to center such that half the base of a triangle of one gear will coincide with half the base of a triangle of the other gear when rotated to the position shown in Fig. 11. The method for computing the actual distance center to center will be demonstrated and derived from this theoretical setting.

Fig. 12 shows the gears of Fig. 11, in the same relative position but with their centers moved further apart in a direction perpendicular to the line that was common to both polygons. The object of this illustration is to show the small amount of loosening in the fit that would attend a very perceptible increase in the gear center distance. Stated in trigonometrical terms, the amount of the opening at each of the two lines of contact on opposite sides of any tooth is equal to two times the external secant of an angle whose tangent is one-half the distance the polygons are moved apart in the direction described and whose radius is one-half the diameter of the tooth. To illustrate by concrete figures, if the polygons are separated as much as one tenth the diameter of the tooth, which may be considered an excessive amount, the theoretical opening on each side will amount to the decimal quantity .005, the tooth diameter being unity. That will give a total play between the gears of one hundredth (.01) part of the tooth diameter. Fig. 14, drawn to a larger scale, is given to illustrate graphically the above trigonometrical statement.

If the mechanical engineer has already determined the maximum of looseness between adjacent teeth which can be allowed without a wasteful leakage of a moderately viscous liquid, he may now use the above formula in determining the leeway he has in the accurate setting of the gear centers.

I take advantage of this possible leeway in the setting of the gears to meet theoretical conditions where the number of teeth in the secondary may be held at six and the number in the primary increased to more than eight. From a theoretical point of view, when the number in the primary is more than eight the polygons have to be separated, as shown in Fig. 12, a very small amount in order that the teeth shall clear each other in the planetary movement of one about the other. The separation, however, is so small that when as much as three or four thousandths (.003 to .004) part of the tooth diameter is allowed for looseness in the journal bearing fit between the teeth the necessity for departing from the setting, shown in Fig. 11, may be ignored. To illustrate by an exact statement, the theoretical separation, as illustrated in Fig. 12, necessary for ten teeth in the primary amounts to the decimal .00126 part of the tooth diameter, and its effect in loosening the teeth from the position of Fig. 11 is too small to be figured with a table of seven place logarithms. Diagram Fig. 11 may be used therefore by the mechanic in determining the gear centers when constructing the clutch as illustrated.

The mathematics involved in the relation of the teeth is given in connection with diagram, Fig. 13. Two teeth of a primary gear have their centers M and N connected by a line. A sequence of points P_6, P_8, etc., located on the extension of a line T P_8 drawn perpendicular to the center of the line MN represent the centers of primary gears having the number of teeth indicated by the sub-figure. The triangle of the points P_8 MN would be, therefore, an isosceles triangle in the polygon of the primary gear with eight teeth. The triangle SRQ is the isosceles triangle of a six tooth secondary gear with the gear center at S and the centers of two teeth R, Q, located on the line MN extended. This is the setting illustrated in Fig. 11.

For a counter-clockwise motion the center S of the secondary would have in actual construction a circular movement in the direction indicated by the arrow around the center P_8 at a constant distance, following a line ST, while the center of tooth

Q would remain on the line TP until the center S had passed an equal distance beyond the line to the point W. The solution of the problem is based on the conditions governing the exact distance from the center S to the center P_8 which will allow tooth R to clear tooth M in this movement. Assuming for a moment that the rotative movement of the triangle SRQ was governed by the center of tooth R being held at a constant distance, RM, from the center of tooth M while the center Q moves along the line TP. Under that assumption the triangle would take the position S'R'Q', the center S having followed the curve SS'. This curve can be figured trigonometrically and is found to be, for all practicable purposes, an arc of a circle with its center located on the line MN. The true curve is slightly flat near the central portion of the length SS' an amount approximately one thousandth (.001) part of a tooth diameter, and as will be recognized later the error in assuming it to be a circle is on the side of clearance and may be ignored.

The engineer will now appreciate that the arc of true movement ST must not encroach on the assumed arc SS' but may be tangent to it. The distance between the point of tangency of the two curves and the center P_8 is the required distance between the gear centers. It can be shown by the differential calculus that when the two arcs are tangent at the point S the radius of the arc ST will center at a point P' just beyond the center P_8 of the eight toothed primary when the assumed arc SS' is that of a six tooth secondary. For centers at P_8, P_7 or P_6, the arc ST will therefore clear curve SS', and the setting shown on Fig. 11 is theoretically correct. The center for the arc SS' is necessarily on the line SP', since the arcs are tangent at S, and is found within finite practical limits to be located at the point V where the line SP' crosses the base line MN. The solution by the differential calculus also discloses that the radial line SP' always intersects the line MN at the same point V, irrespective of the number of teeth in the secondary gear. With more than six teeth the center S and the assumed arc SS' would be located farther to the left of the base line MN while the center P' for the actual curve tangent at S would be farther to right.

Consider next that the arc ST for the six tooth secondary is actually tangent to the curve SS' at a point S'' for a primary gear having twenty teeth, and draw the line S'' P_{20} connecting S'' with the center P_{20}. This line necessarily passes through the center of the arc SS' located, as stated above, at point V. Its length is equal to the radius S'' V or SV of the arc SS' plus the distance VP_{20}, both of which can be computed from the dimensions of the gears and the location of the point V. If in this computation the diameter of the tooth is taken as unity the distance VQ=.5857861 and VM=.4142139. The distance S'' P_{20} then found represents the correct setting of the gear centers, on the assumption that the arc SS' is a true arc of a circle as already stated. If now the line S'' P_{20} is swung back clockwise on the center P_{20} the end S'' in its actual movement will fall on the line MS at a point slightly to the left of S, an amount too small to represent on the diagram. This amount is the separation of the polygons required and as shown in Fig. 12 when the ratio of the teeth is greater than eight to six with six in the secondary, or greater than the ratio of .5857861 to .4142139 for any other number in the secondary.

The exact location of the tangent point S'' on the actual curve SS' and the absolute distance S''P for any primary (the smaller of the two gears being considered as the secondary) may be determined by means of trial solutions of a long trigonometrical equation; but as that refinement is of theoretical rather than practical importance the statement of the equation and its explanation will not be given.

Referring to diagram, Fig. 13, at the time tooth Q moves inward on the line TP it will have a journal tight fit between short parallel faces in the interdental space between primary teeth M and N. During this movement tooth Q will act like a piston and provision must be made for the discharge of any liquid that may be confined in the bottom of the space. In my present invention I have provided for this clearance by extending the gear recess spaces 2 and 2^a a short distance into the body of the main casing back of the interior faces of the gears. The extension of these recesses is shown in the transverse section of Fig. 8, which is taken on the line 8—8 of Fig. 7, a little back of the primary gear. The extensions are also shown from the opposite direction, partly in dotted outline, in Figs. 15 and 16.

The conformation of the extensions, which I will call ports, are marked 31 and 31^a, and are such that an interdental space is always open to one or the other except for a fraction of movement as the center of the space passes the medial line of the gears at a moment when the piston action of the tooth in the space is practically nil. Fig. 15 illustrates all the interdental spaces adjacent the medial line as opening into a port. Fig. 16 shows one space on the medial line to be cut off. The dotted outlines of the ports show the relative short arc through which the tooth and space have to move in either direction for the latter to again con-

nect with a port. It will thus be noted that the adjacent edges of the ports 31 and 31ª are so located that liquid cannot leak from one to the other as a space and tooth pass the medial line of the gears, but these edges are sufficiently close to the medial line to provide clearance for the liquid in each of the interdental spaces as a tooth of the companion gear moves into it in piston-like fashion.

Referring to Fig. 11, only that portion of the tooth surfaces subtended between the base of one isosceles triangle and the line of the base of the adjoining triangle extended need to be cylindrical. In the illustrations of the clutch I have shown the bottoms of the spaces below the short parallel faces to be circular in outline to conform to teeth with full cylindrical ends. The ends of the teeth are truncated, however, somewhat outside the limit of the external angles of the polygon as indicated by a dotted line 32 in Fig. 11, to conform to the peripheral wall of the gear recess and insure a liquid tight fit of the gear therein. There is, therefore, a portion of the bottom of the interdental space that need not be emptied during the intermeshing movement, and this surplus portion facilitates the flow of the liquid up to the limit when the space is cut off from either port. When the clutch is running in the open position and during the initial stage when the clutch is being closed centrifugal force will more or less clear these spaces before the intermesh takes place.

The use of the special spur gears, each with cylindrical shaped teeth that have a rolling sliding double contact between parallel faces in interdental spaces of the companion gear, in a power transmission device in which the transmission is attained through the interlock of the gears by means of a liquid medium, and in which the intermeshing movement of the gears when so transmitting power is relatively small and slow as compared with the total rotative movement of the device is a subject of U. S. Letters Patent No. 1,420,798 issued to me June 27, 1922, and is not claimed broadly herein.

What I claim as new in my present invention and desire to secure by Letters Patent is:

I claim:

1. A hydraulic clutch comprising a rotating driving casing; a driven shaft concentric with the casing; a pair of intermeshing spur gear pump gears carried in a close fitting recess in the casing, one of said gears being attached to said shaft, said casing having a storage chamber and passages between the storage chamber and said gear recess, means for controlling said passages to open and close the clutch, a liquid medium partly filling the storage chamber adapted by an intermeshing movement of the said gears to have a circulatory flow into or from said recess, and structural means whereby the liquid medium may flow into but not from said recess with the clutch closed and flow from but not into said recess with the clutch open.

2. A hydraulic clutch comprising a rotating driving casing, a driven shaft concentric therewith, a pair of intermeshing spur gear pump gears carried in a close fitting recess in the casing, one of said gears being attached to said shaft, said casing having a storage chamber and passages between the storage chamber and said gear recess, valve means for controlling said passages to open and close the clutch, a liquid medium partly filling the storage chamber adapted by an intermeshing movement of the said gears to have a circulatory flow into or from said recess, and structural means whereby said liquid medium may flow into but not from said recess with the clutch closed and flow from but not into said recess, with the clutch open; said structural means being symmetrical and operative with the clutch rotating in either direction.

3. A hydraulic clutch comprising a rotating driving casing, a driven shaft concentric therewith, a pair of intermeshing spur gear pump gears carried in a close fitting recess in the casing, one of said gears being attached to said shaft, said casing having a storage chamber and passages between the storage chamber and said gear recess, valve means for controlling said passages, and a liquid medium confined in and filling said recess when the controlling valve means is in one position and cleared and excluded from said recess when the valve means is in another position.

4. A hydraulic clutch comprising a rotating driving casing, a driven shaft concentric therewith, a pair of intermeshing spur gear pump gears carried in a close fitting recess in the casing, one of said gears being attached to said shaft, said casing having a storage chamber and passages between the storage chamber and the said gear recess, valve means for controlling said passages, and a liquid medium confined in and filling said recess when the controlling valve means is in one position and cleared and excluded from said recess when the valve means is in another position, the construction being symmetrical and operable with the driving casing rotating in either direction.

5. In a hydraulic clutch of the spur gear pump type wherein, of the pump elements, the pump casing is rotatable to carry the secondary gear in a planetary movement about the primary gear and constitutes the driving member, and the shaft attached to the primary gear concentric with the casing constitutes the driven member, and wherein

a relative movement between said driving and driven members establishes a suction and discharge action with the elements acting as a pump, the combination of said pump elements: said casing having a storage chamber and two series of passages, a liquid medium in said storage chamber held away from the center of rotation by centrifugal action, and two valves cooperating respectively with said two series of passages to control the suction and discharge action of said pump elements on said liquid medium and adapted in one position to close the discharge side of the pump and connect the suction side to the storage chamber at a point submerged by the liquid medium, and, in another position, to connect both discharge and suction sides to the storage chamber at points not submerged by the liquid medium.

6. A hydraulic clutch of the spur gear pump type comprising a rotating casing for the main driving member, a shaft concentric with the casing for the driven member, a pair of intermeshing spur gears set in a liquid tight fit in a gear recess in the casing, one of said gears being attached to the shaft, said casing having a storage chamber and two series of passages between said chamber and the gear recess and arranged on opposite sides of said recess, a liquid medium in the storage chamber adapted to be held to the radially outer portions of said chamber by the centrifugal force generated by the rotating casing, and two operable valves for governing the flow of the liquid through the passages to and from the gear recess and adapted in one position, to confine a portion of the liquid medium in the gear recess and prevent intermeshing movement of the gears, and, in another position, to permit the clearance of the liquid medium from the gear recess and the free intermeshing movement of the gears.

7. A hydraulic clutch of the spur gear pump type adapted to be equally effective under a reversal of strain between the interlocked parts when in the closed position or under a reversal of relative movement of the unlocked parts when in the open position, and comprising a rotating casing for the main driving member, a shaft concentric with the casing for the driven member; a pair of intermeshing spur gears set in a liquid tight fit in a gear recess in the casing, one of said gears being attached to the shaft, said casing having a storage chamber and two series of passages between said chamber and the gear recess and arranged on opposite sides of said recess, a liquid medium in the storage chamber adapted to be held to the radially outer portions of said chamber by the centrifugal force generated by the rotating casing; two operable valves for governing the flow of the liquid through the passages to and from the gear recess, and two automatic check valves located in said passages on opposite sides of the gear recess and adapted to open only for a flow of the liquid toward said recess.

8. In a clutch of the hydraulic pump type, a pump mechanism comprising rotatably mounted pump elements constituting the driving and driven clutch members, a storage chamber rotatable with the driving clutch member, separate passages connecting the radial inner and outer portions respectively of said storage chamber with said pump mechanism, valve means interposed in each of said passages for controlling the flow therethrough, and a fluid medium, part liquid and part gaseous in said storage chamber, the liquid portion, in operation, being held in the radially outer portion of the chamber by centrifugal action.

9. A hydraulic clutch of the spur gear pump type comprising a main rotating casing A with cover plates B and C for the driving member, a shaft F concentric with the casing for the driven member, intermeshing gears P and S set in a liquid tight fit in a gear recess in the casing, the gear P being attached to the shaft F, said casing having a storage chamber J, and two series of passages between the storage chamber and the opposite sides of said gear recess, two piston valves 3 and 3^a for governing said passages and means for shifting the piston valves jointly from one to the other of two operative positions and thereby either permit the independent rotative movements of the driving and driven members or effect the interlocking thereof by a liquid medium confined in the gear recess.

10. In a clutch of the hydraulic pump type wherein a rotating casing containing the pump elements, the liquid medium and the operating valves, constitutes the driving member, and a shaft concentric with the casing and attached to one of the pump elements constitutes the driven member; the combination of said rotating casing, pump elements, liquid medium and valves, said casing having a storage chamber and a passage from a radially outer portion of the storage chamber adapted to be opened to the suction side of said pump elements when the operating valves are in the closed clutch position, and said casing having two passages from a radially inner portion of the storage chamber adapted to open respectively one to the suction side and one to the discharge side of the pump elements when the operating valves are in the open clutch position.

11. In a clutch of the hydraulic pump type wherein a rotating casing, containing the pump elements, the liquid medium and the operating valves, constitutes the driving member, and a shaft concentric with the casing and attached to one of the pump elements

constitutes the driven member; the combination in and with said rotating casing and the said parts contained therein, said casing having a storage chamber and two passages leading from radially outer portions of the storage chamber, one to the suction side and one to the discharge side of said pump elements, automatic check valves normally closing said passages and one or the other of which is opened by suction action of the pump elements when said operating valves are in closed clutch position, and two other passages leading from a radially inner portion of the storage chamber and adapted to be opened respectively one to the suction side and one to the discharge side of the pump elements when said operating valves are in the open clutch position the arrangement permitting the operation of the clutch for a rotation of the driving member in either direction.

12. A hydraulic clutch comprising a rotating casing, a shaft concentric with the axis of rotation of said casing, a pump mechanism actuated by any relative rotary movement of said casing and said shaft, said casing having a storage chamber and separate passages connecting said storage chamber and said pump mechanism and constituting suction and discharge passages for the latter, a fluid medium separated into liquid and gas portions in said storage chamber and said passages, and controlling valve means interposed in each of said passages for opening and closing the clutch, said valve means cooperating with said passages to cut off the discharge of fluid medium from said pump mechanism and limit the suction thereof to liquid when the clutch is closed, and permit the discharge of fluid medium from said pump mechanism and limit the suction thereof substantially to gas when the clutch is open.

13. A hydraulic clutch comprising a rotating casing having a number of recesses and passages, a shaft concentric with said casing, a fluid medium, part liquid and part gaseous, filling said recesses and passages, a pump mechanism actuated by any relative rotary movement of said casing and shaft to cause a flow of the fluid medium through the recesses and passages, and valve means for controlling said passages adapted to cut off the discharge of fluid medium from said pump mechanism and limit the suction action thereof to liquid when the clutch is closed, and permit the discharge of fluid medium from said pump mechanism and limit the suction action thereof substantially to gas when the clutch is open, the said casing having a contracted duct for admitting a small amount of the liquid to the suction side of the pump mechanism for lubrication when the clutch is open.

14. A hydraulic clutch of the spur gear pump type, wherein for the closed clutch position the spur gears are interlocked by means of a liquid medium confined within the gear recess, and wherein for the open clutch position the spur gears have a free intermeshing movement, and comprising a rotating casing for the driving member; a shaft concentric with the casing for the driven member, said casing having a series of recesses and passages, a fluid medium, part liquid and part gaseous, filling said recesses and passages, a pair of intermeshing spur gear pump gears set within one of said recesses, one gear being attached to the said shaft, and the other being rotatably mounted in the recess, and adapted by their intermeshing movement to produce a circulatory flow of the fluid medium through said recesses and passages, the spur gears having teeth each with cylindrically shaped bearing surfaces on opposite sides adapted to have a rolling sliding fit between short parallel surfaces in the interdental space of the companion gear, and a pair of operable valves for governing the flow of said fluid medium through the recesses and passages and thereby, in open clutch position, maintaining the gear recess substantially empty of the liquid portion of the fluid medium.

15. A hydraulic clutch comprising a rotating casing constituting the driving member, a concentric shaft constituting the driven member, a pump mechanism in said casing adapted to be operated by and during any relative movement of said shaft and casing, a storage chamber, a fluid medium separated into liquid and gas portions in said chamber, separate passages connecting said storage chamber with said pump mechanism and constituting the suction and discharge passages respectively for the latter, a valve for opening and closing said discharge passage to open and close the clutch and a valve cooperating with said suction passage to limit the suction action of the pump to the liquid portion of the medium when the clutch is closed and substantially to the gaseous portion thereof when the clutch is open.

16. In a clutch of the hydraulic pump type, the combination of a pump mechanism wherein the pump casing is rotatable and constitutes the driving member and wherein the pump shaft extending into the pump casing concentric therewith constitutes the driven member, a storage chamber rotatable with said pump casing and passages connecting said storage chamber to the suction and discharge sides respectively of said pump mechanism, a fluid medium, part liquid and part gas, in said chamber, and means for controlling the flow through said passages comprising a valve interposed in one of said passages and cooperating therewith to admit only liquid to the suction side of the pump mechanism in closed clutch po-

sition and only gas in open clutch position and another valve interposed in the other of said passages and cooperating therewith to close and open the discharge side of the pump mechanism in its closed and open clutch positions respectively.

17. In a clutch of the hydraulic pump type, a pump mechanism including a rotatable casing and a concentrically mounted rotatable shaft, said pump mechanism having a discharge passage and separate passages for the supply of liquid and gas respectively to the suction side thereof, and controlling valve means interposed in each of said passages and arranged, in closed clutch position, to close the discharge and gas supply passages and permit flow through the liquid supply passage and, in open clutch position, to close the liquid supply passage and permit flow through the discharge and gas supply passages.

18. In a clutch of the hydraulic pump type, a pump mechanism comprising a rotary casing having a pump chamber, a storage chamber and separate passages connecting said storage chamber to the suction and discharge ports of said pump chamber, a concentrically mounted shaft extending into said casing, a pump element in said pump chamber operatively connected to said shaft, a fluid medium, separated into liquid and gas portions, in said storage chamber, and valve means interposed in each of the passages between said pump and storage chambers and adapted, in one position, to close the discharge from said pump chamber and connect the suction port thereof to the liquid containing portion only of the storage chamber and, in another position, to open the discharge from the pump chamber and connect the suction port thereof to the gas containing portion only of the storage chamber.

19. In a clutch of the hydraulic pump type, a pump mechanism comprising a rotary casing having a pump chamber, a storage chamber and separate passages connecting said storage chamber to the suction and discharge ports of said pump chamber, a concentrically mounted shaft extending into said casing, a pump element in said pump chamber operatively connected to said shaft, a fluid medium, separated into liquid and gas portions, in said storage chamber, and valve means interposed in each of the passages between said pump and storage chambers and adapted, in one position, to close the discharge from said pump chamber and connect the suction port thereof to the liquid containing portion only of the storage chamber and, in another position, to open the discharge from the pump chamber and connect the suction port thereof to the gas containing portion only of the storage chamber, said valve means having an intermediate position for partly closing the discharge from said pump chamber and for connecting the suction port thereof both to the liquid and gas containing portions of the storage chamber.

20. In a clutch of the hydraulic pump type, a rotatable casing comprising a main section recessed in its opposite end faces to form respectively a gear pump chamber and a storage chamber and having separate suction and discharge passages connecting said chambers, cover plates secured in liquid-tight fashion to the ends of said main section and forming the outer walls of said chambers, a shaft extending through the storage chamber and the adjacent cover plate and into said pump chamber, intermeshing spur pump gears in said pump chamber, one of said gears being connected to said shaft, a piston valve for opening and closing said discharge passage, said main casing section having a chamber for said piston valve, the opposite ends of said piston chamber being in communication with said storage chamber, one such communication being restricted to retard the closing movement of said valve and means extending into said storage chamber through the adjacent cover plate for shifting said valve.

21. In a clutch of the hydraulic pump type, a rotatable casing comprising a main section recessed in its opposite end faces to form a gear pump chamber and a storage chamber respectively and having passages connecting said chamber to the suction and discharge sides of said pump chamber, cover plates secured to the ends of said main section and forming the outer walls of said chambers, a shaft extending through the storage chamber and the adjacent cover plate and into said pump chamber, spur pump gears in the latter chamber, one being connected to said shaft, controlling valve mechanism co-operating with said passages and arranged, in closed clutch position, to connect only the suction side of the pump chamber and the radially outer portion of the storage chamber and, in open clutch position, to connect both the suction and discharge sides of the pump chamber to the radially central portion only of the storage chamber, and shifter means for the valve mechanism extending into the storage chamber through the adjacent cover plate.

WILLIAM S. WESTON.