Dec. 7, 1954 M. F. HILL 2,696,170

CIRCULATING PUMP

Filed Oct. 4, 1951

Fig. I

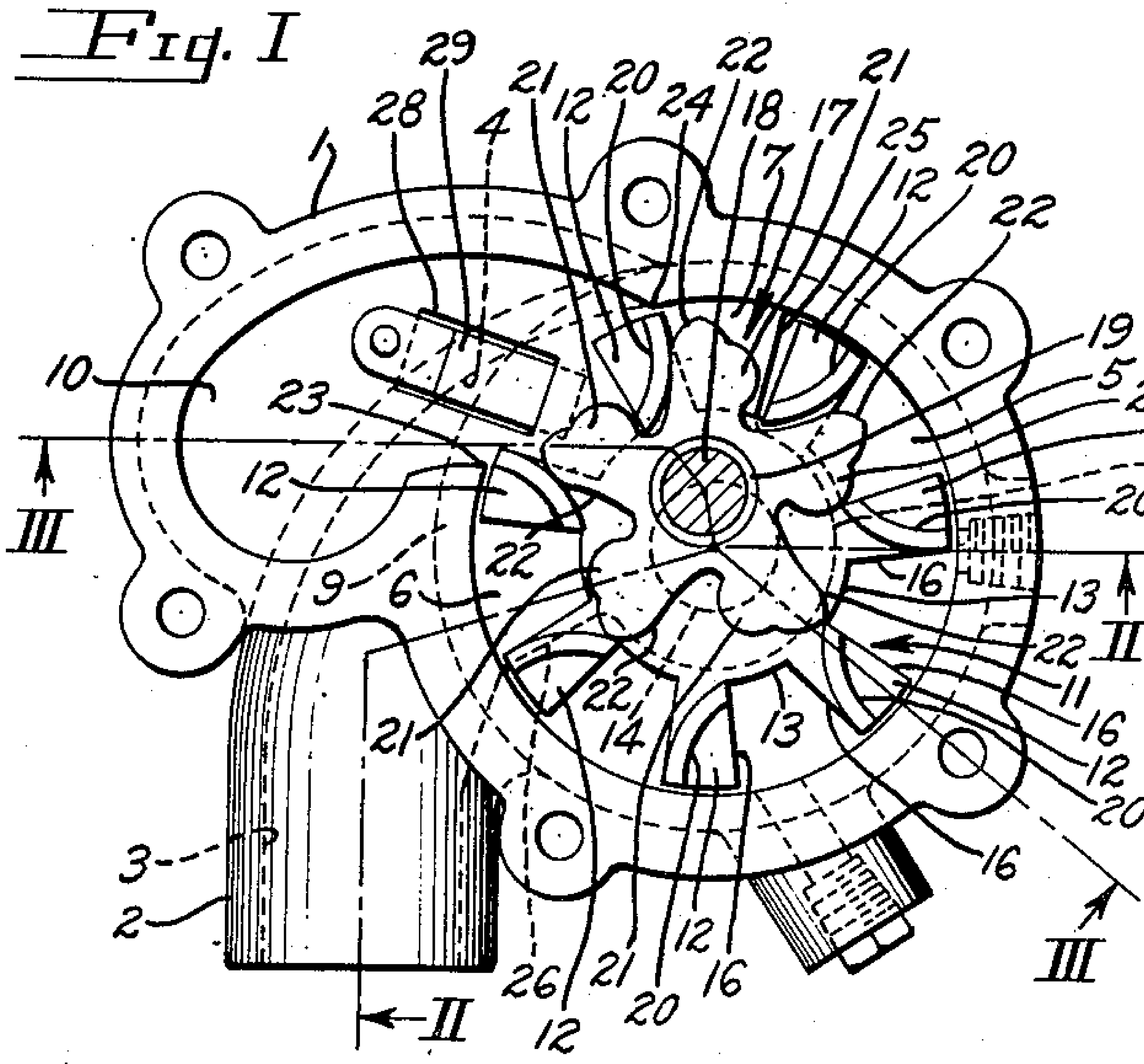


Fig. IV

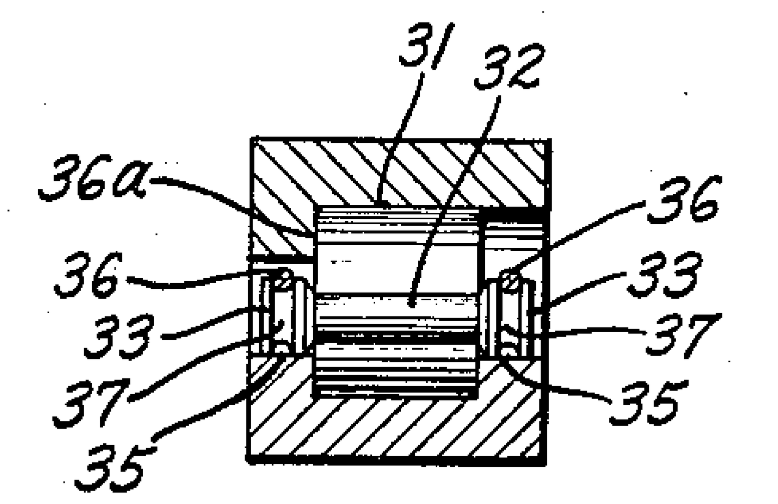


Fig. V

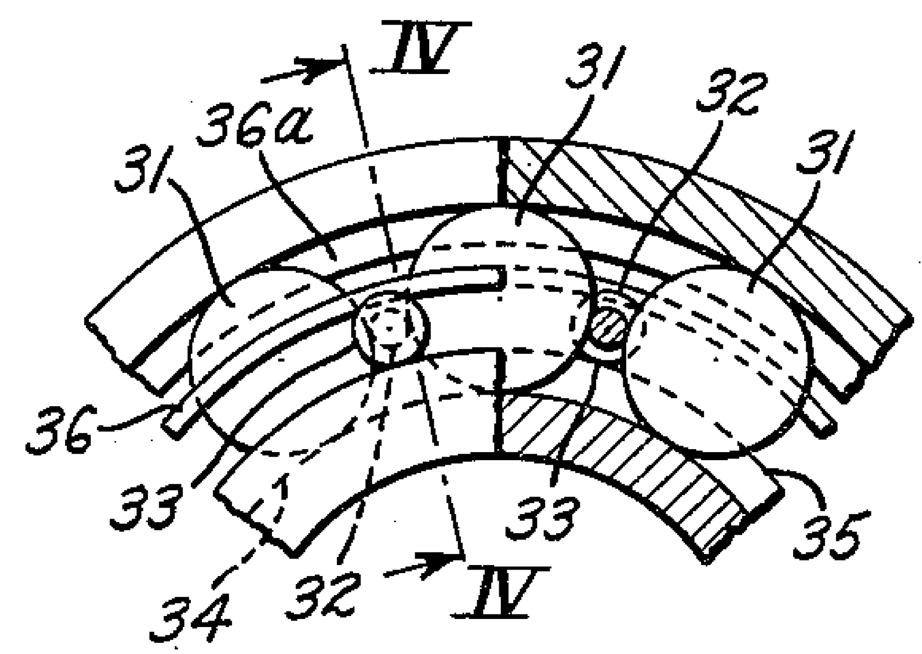


Fig. II

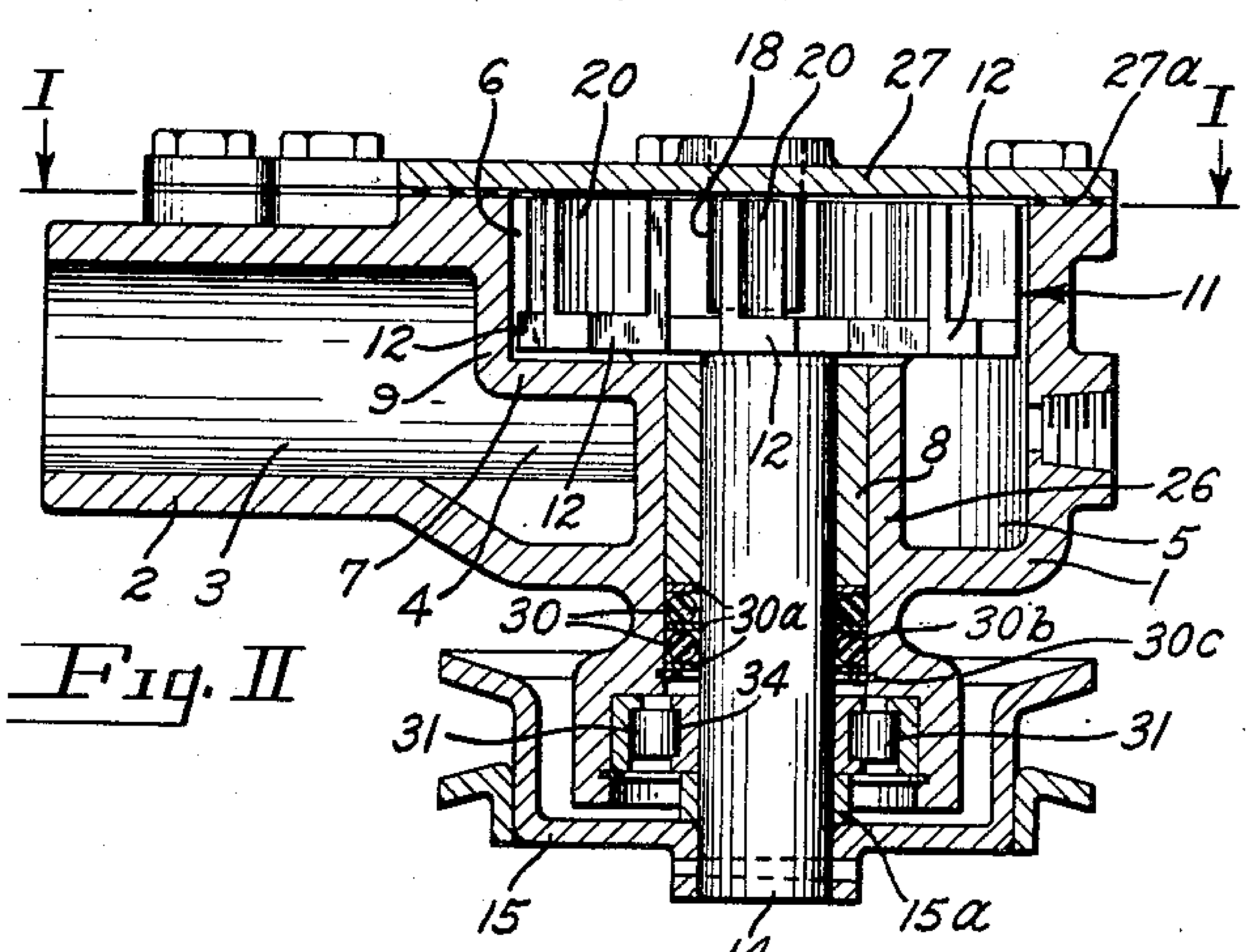


Fig. VI

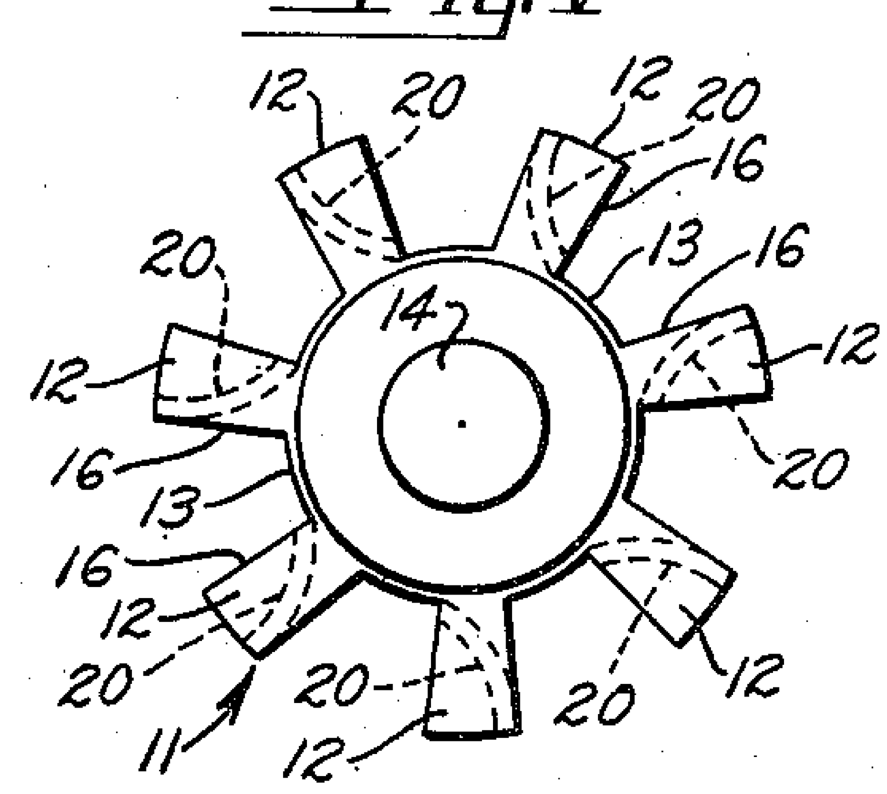


Fig. VII

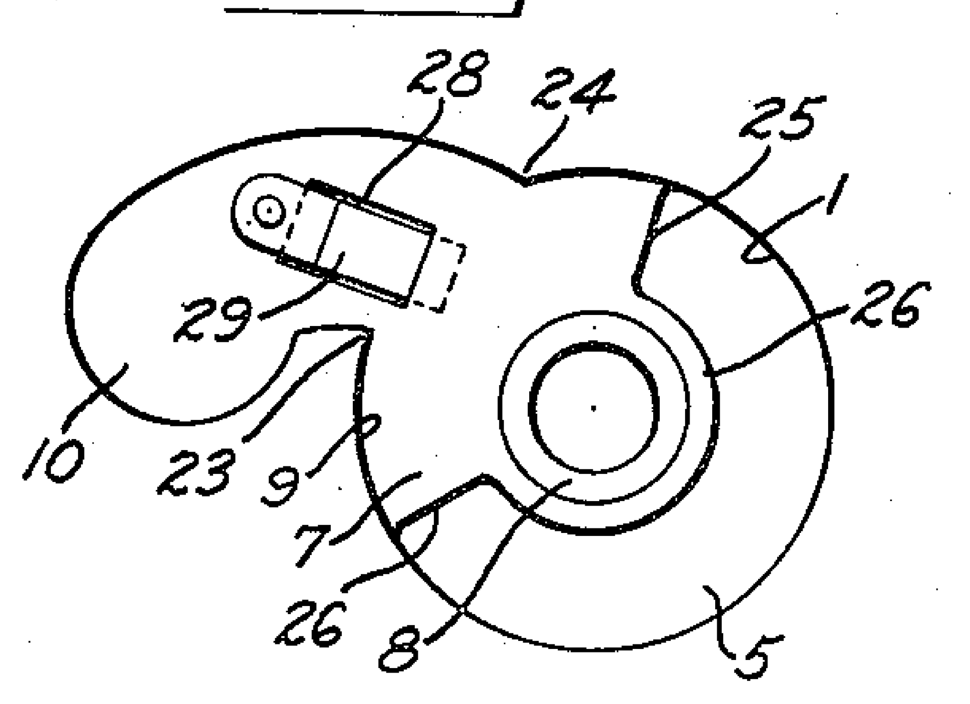
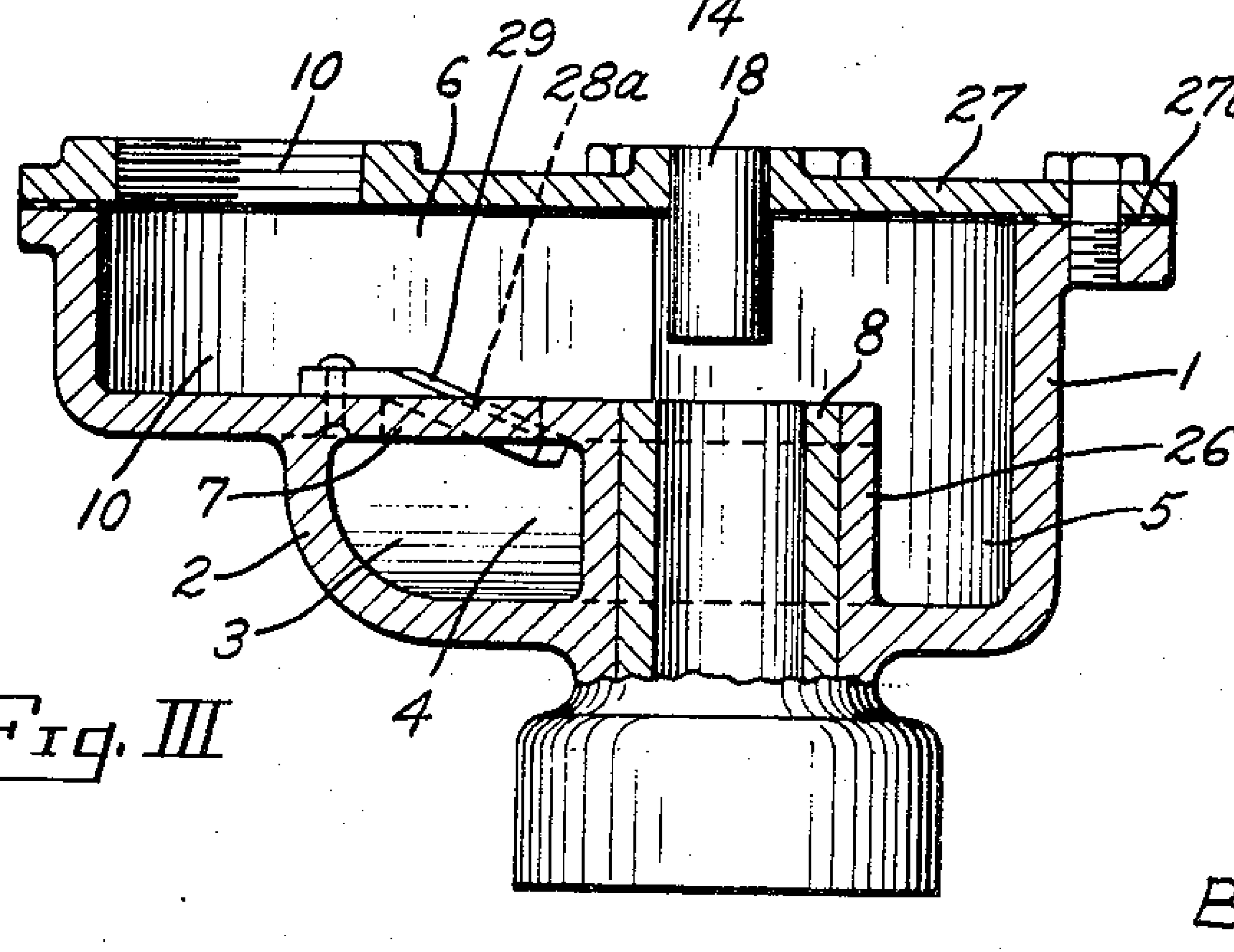


Fig. III



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CIRCULATING PUMP

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7 Claims. (Cl. 103—42)

My invention relates to circulating pumps for such water circulating systems, for example, as that in gas engined autos with radiators to cause water heated in the engine cylinder jackets to circulate through the radiators and back again through the said jackets for cooling said cylinders.

Centrifugal pumps alone are now used for this service. When engines are idling, however, adequate circulation fails, and the water in the radiator becomes cold. As the engine is started up again, the cold water from the radiator so chills the cylinder that cracks occur, ruining them.

The object of my invention is a combination centrifugal and displacement pump for a gas engine and its radiator to keep the engine cool in both running and idling. The centrifugal pump at operating engine speeds pumps volumes of fluid without developing excessive pressures in so far as the radiator is concerned, and the displacement gear at idling engine speeds causes enough circulation through the radiator to prevent it from getting cold.

High pressure pumps of the type used in hydraulic pressure or lubricating pumps would burst the radiators which are of thin metal sections. Instead my rotors pump only at low pressures. They are mounted loosely in the pump casing, to permit enough leakage around them to prevent damaging pressures. Loosely is used in a relative sense. Hydraulic pumping gears have from .0003" to .005" clearance between the side walls and between their teeth. In my invention, the gears or rotors or impeller and gear may have ten times this amount of clearance and sometimes more.

My rotors will pump water at idling engine speeds when centrifugal pumps fail to circulate sufficient cooling water. On the other hand, at higher engine speeds they will not develop damaging pressures.

Centrifugal auto pumps are usually cast with no finish machining of the rotor. My rotors also are cast, with such tolerances to allow copious leakage. As pressure rises with speed of rotation, due to friction in the water passageways of the circulating system, the leakage of water increases through the tooth and looseness between the rotors and the casing, reducing the displacement per rotation to minor proportions. A by-pass valve also provides an additional pressure limitation.

My pump employs a one-direction outer rotor having blades to fan the water around, something after the order of centrifugal pumps; the blades having rotor contours on one side of each blade for continuous contact with cooperating tooth contours on one side of each tooth of the pinion rotor. Although inclined at an angle not the best for purely centrifugal action, however, by entering the spaces between the outer rotor teeth, the pinion or inner or pinion gear teeth force a discharge.

The outer rotor or impeller teeth are relatively thin in cross section to provide large tooth spaces between them. The back faces of the pinion teeth are made as large as possible by adding to the back face to better fill these spaces, thus increasing displacement per revolution. The path traced by the inner ends of the impeller teeth as they pass around the gear teeth during tooth engagement determine the back face thickness of the gear teeth relative to its rotation.

Circulating pumps operate on water in autos and trucks. Separate chambers for oil have been needed for plain bearings capable of running at several thousand

revolutions per minute. The roller bearing shown eliminates the oil chamber, since its rollers do not rub against a cage, but roll against separating rollers which in turn are mounted in a rolling support. Bearings of this type have had long tests under dry conditions. Sand even has been rolled into powder and blown out by air circulation. They are self-cleaning. 40,000 R. P. M. have been attained without injurious heat, many times the maximum speed of circulating pumps in auto service.

In the drawings:

Fig. I is a plan view taken on line I—I of Fig. II in the direction of the arrows;

Fig. II is a section taken on line II—II of Fig. I in the direction of the arrows, the pinion being omitted, and the impeller being shown in full lines;

Fig. III is a section of the casing taken on line III—III of Fig. I in the direction of the arrows, the by-pass valve being shifted for better showing;

Fig. IV is a section taken on line IV—IV of Fig. V in the direction of the arrows;

Fig. V is a part plan and part sectional view showing the bearing details;

Fig. VI is a bottom plan view of the impeller of Fig. II on a reduced scale looking upward from the bottom of the pump; and

Fig. VII is a plan view of the port and shelf system in Fig. I on a reduced scale.

In Fig. I is shown a casing **1** having a hose connection **2** for the intake side of the pump. Its passage or passageway **3** extends, as shown at **4**, to the pump cavity **5**. This cavity has two portions, the intake passage or passageway **3** and the discharge region **6** (Fig. III) separated from the intake passageway by the wall **7** extending from the bushing **8** to the outer wall of the casing at **9** (Fig. I) and beyond it to the discharge outlet **10**.

The outer rotor consists of a back plate or impeller **11** having arms **12** (Figs. I, II, and VI) with driving teeth **20** with convex driving faces. The impeller **11** is driven by the shaft **14** which is driven by the pulley **15** or by such other power as may be desired. The impeller has enlarged spaces between the blades **12** at **13** and between the arms of the impeller at **16**, and at substantial speeds can act like a centrifugal pump.

The pinion rotor **17**, the displacement rotor, is freely mounted on the shaft **18**, its teeth driven by the outer rotor. It may have a bushing **19** of Graphitar, which with the super-finish now attainable on shafts is durable in water for the relatively light load upon it. Between the teeth **20** on the impeller **11** there are wide tooth spaces containing liquid driven around in the casing so that considerable liquid centrifugal force develops at substantial rotary speed for displacement purposes. Spaces **13** between the arms **12** of the impeller **11** connect the tooth spaces to the intake port or suction area **5**.

The impeller teeth may be thin as shown at **20**. This provides for wide teeth spaces between the impeller teeth. Because these spaces are so wide, the pinion or inner gear teeth may be thick as indicated at **21**. The driving faces of the teeth of the impeller and the driven faces of the inner gear have contours for maintaining continuous fluid tight engagements as disclosed in M. F. Hill et al. Patent No. 2,601,397. The non-driven or opposite faces of the inner gear may have any shape. They may be flat or concave or convex. But they must be of such shape as not to clash with the inner ends of the teeth of the impeller. The thick teeth shown in the drawings are preferable because they displace more liquid during tooth engagement and particularly at full mesh. This displacement by the gear teeth more than offsets loss of efficiency due to the tilting of the impeller teeth in the wrong direction for the best centrifugal action. Each rotor chamber in the discharge section **6** extends from one tooth contact to the next one. These rotors are for clockwise rotation as a pump and discharge radially outward between the points of the casing at **23** and **24**. The discharge takes place over the wall **7** which is outlined by the solid line **25** and the dotted line **26**. This wall is illustrated more clearly in solid lines in Fig. VII on a reduced scale. The outlet **10** may be a hole in a cover plate **27** (Fig. III) to fit existing engine casings or it may be any pipe connection of usual form for convenience of flow. To prevent pressure rising to injurious proportions, a by-pass may be used. A convenient location is in the wall **7**, with an aperture communicating with the intake passageway **4**, as at **28** Fig. I or **28A**, Fig. III. A neoprene valve **29** may be riveted in the wall **7** as indicated, so that the rivet may be engaged by tools outside and inside of the casing as the rivet is headed over. The valve is passed thru the aperture. Pressure of fluid in the discharge area above the wall opens the valve to allow escape. The stiffness of the neoprene determines the maximum pressure and should be suited to the desired action. Any form of valve may be used of course, that shown being a convenient one for mass production at low cost.

When castings are made shrinkage occurs in cooling so that castings are smaller than the original pattern castings. However, by careful design of patterns and molds, rotor shrinkage is taken into account so as to maintain correct tooth relations. If the patterns are made of correct thickness for service then the castings will also be of the correct thickness for service although thinner than the patterns. These shrinkages are taken into consideration in making patterns and in designing of the pump. As the casing is cast according to the same plan it also will shrink in such a way as to maintain the same depth of chamber for the rotors to work in. If the rotors and casing bore dimensions are made with theoretical sizes and the castings are put together without machining, the joint between the cover plate and the casing body would leak. Also the rotors would be tight and unable to turn. A gasket **27***a* of the requisite thickness provides clearance for the rotors to work in and with additional thickness to provide a by-pass for liquid as pressure rises due to speed, pipe friction and other factors in the cooling system of an auto. The rotors may have plenty of room for operation and too high pressure may be prevented. The same is true if the joint between the cover plate and casing body having machined surfaces. As the gasket is made of soft material on those faces which make contact with the cast surfaces it will seal the joint between the casing and cover. A gasket having a layer of fabric filled in on both sides with rubber or neoprene, or any other similar material, will fill the needs of such a pump, and keep the joint tight.

In order to prevent water in the pump from leaking out around the shaft and into the roller bearing, I prefer to use a form of what is known as the linear seal, which consists of one or more rings **30** of neoprene impregnated, often, with graphite, which hug the shaft **14** and which press against side wall members **30***a* in the casing and are held to their work by the bore **30***b* of the casing where the seal is inserted. Water is unable to leak out. There may be as many of these rings as desired. One ring does great service. Two rings guard against any flaws that might develop in manufacture of one. The first ring may rest against the end of the bushing **8**, the surface of which is smoothed off, then comes an annular disk or washer and a second linear ring and a second annular disk or washer held in position by a snap ring in a groove **30***c*, all of which are of types well known to the art, and are illustrated in Fig. II. The neoprene rings are usually greased before being installed in order to prevent the shaft from sticking to the rings when dry, when first started. Thereafter there is no danger of sticking.

The shaft **14** may be driven by the belt pulley **15** pinned to the shaft. As the pulley drive sometimes exerts a heavier than usual load upon the shaft bearing, a novel type of roller bearing is employed. As this is primarily a water circulating pump without oil bearings my desire is for a roller bearing for indefinite periods of use, without oil or grease. In my Patent 784,002 was shown and described such a bearing tho its oil-less quality was not then recognized. It provided pure rolling motion. Hundreds of thousands of applications however in shafting and auto service proved that lubrication was not needed.

Bearings made under this patent were the first antifriction bearings commercially adopted by autos. The bearing in this pump has substantially a pure rolling action but by different means. In Fig. IV a section shows a main roller **31** and a separating roller **32**.

The separating roller harmonizes the opposing roller surfaces in rolling together. The separating roller has the enlarged ends **33**, preferably engaging the ends of the main rollers to keep them parallel, as in the patent. But in the patent the diameters of the ends and bodies of the separators had a proportion equal to that of outer separator supports or retainer caps to the main inner races. This bearing, instead, omits that ratio, since to keep to it may mean new manufacturing tools throughout industry for races and other bearing parts.

By locating the axis of a separator substantially in a plane thru the axes of the two adjoining main rollers as indicated in Fig. V at **33**, to prevent radial inward or outward thrust as main rollers press against them, and by trimming the diameters of the ends of the separators to lightly touch, or just clear, the shoulders **35** which have diameters determined by manufacturers of standard roller bearings; and then fitting rings **36** around the outsides of such separator ends, a friction free mounting is attained. The separators, under the influence of centrifugal force, roll in the rings; and the rings, instead of running with the shaft as in my early patent, travel differently, it may be forward or rearward, according to separator diameters, but with a pure rolling action. Centrifugal force causing the separator ends to press outwardly against the rings during rolling, causes the rings to run concentric with the other bearing parts. Any displacement, inward or outward, of a separator causes its ends to touch with a reverse action somewhere upon the shoulders **35**, and at once push the separators and rings back into concentric relations. The main rollers **31** cannot thrust the separating rollers in or out radially because the latters' axes are substantially maintained in the plane **31***a* through the adjacent main roller axes. Jamming outward or inward is prevented so that the separators touch the shoulders **35** lightly. The retaining ring **36** can be of light construction as its only function is to keep the small separating rollers in place. The ends **33** of the separators normally clear the shoulders **35** by only a few thousandths of an inch. These members are hardened and even ground to accuracy, and may be mounted to just clear the shoulders **35**. A bearing for a shaft of 1¼" diameter having applicant's earlier type would run indefinitely without injurious heat at 40,000 R. P. M.

To prevent the rings from drifting endwise off from the separator ends they may be guided by grooves illustrated at **37**. The advantage of this roller system is that it can be applied to many makes of antifriction roller bearings.

Endwise fluid pressure balance is needed to prevent wear between the rotors and casing endwise. The pinion has the same fluid pressures at both ends. The spider is balanced by fluid pressure in the rotor chambers being communicated to the rear side thru the spaces **16**.

To provide for any minor residual unbalance, the driving plate runs against the bushing **8** and its shaft carrying the pulley **15** runs against a washer **15***a* and sleeve **34** of the roller bearing. The sleeve runs against the rollers **31** which run against the inner shoulder **36***a* on the outer race which is fixed to the pump casing.

It is obvious that my invention may also be used as a motor by causing fluid under pressure to travel thru the mechanism in reverse, causing the rotors to run counter-clockwise. The outgoing shaft **14** then may apply power, thru the pulley or by direct action, to outside use. Nor is it limited to liquids, for gases also may be employed.

My mechanism is designed for one direction operation. As a pump it operates clockwise. If counter-clockwise operation is sought, curves for the opposite sides of the rotor teeth are employed.

Many variations in passageways, valving, sizes, character of curves, with good or inferior tooth engagements, due to imperfections of manufacture or to design, lie within the scope of my invention.

I claim:

1. A rotary fluid mechanism comprising a casing, an intake passage and a discharge passage in the casing, a drive shaft, an impeller mounted on said shaft and in said casing, a pinion gear in said casing within and eccentric to said impeller, a shaft and journal for said pinion gear supported by said casing, said pinion gear having teeth, each of which has an enlarged portion on one side face relative to the other side face, said impeller having teeth relatively thin compared with the thickness of the teeth of the pinion gear in cross section relative to a plane at right angles to the axis of rotation of the impeller and which are so shaped to drive fluid around in

said casing to develop centrifugal fluid force to drive fluid out of said discharge passage, one face of said teeth of said impeller having contours for maintaining continuous fluid tight engagement at steady angular speeds with contours on the said other side face of said teeth of said pinion gear, the said enlarged portions of the teeth of said pinion gear each being limited in size and shape to pass the inner ends of the teeth of said impeller during tooth engagement, and said pinion gear acting as a displacement member.

2. A rotary fluid mechanism according to claim 1 in which the enlarged portions of the teeth of the pinion gear and said impeller teeth have more than normal sealing clearance with respect to each other and the casing to provide a fluid passageway between said teeth from the intake passage to said discharge passage to prevent excessive pressure in said discharge passage.

3. A rotary fluid mechanism according to claim 1 in which the teeth of both the impeller and the pinion gear have more than normal sealing clearance to prevent excessive pressure in said discharge passage.

4. A rotary fluid mechanism according to claim 1 in which a wall is provided in the casing to separate the intake and discharge passages, and in which a by-pass valve is provided in said wall to limit the fluid pressure in one passage as compared to that in the other passage.

5. A rotary fluid mechanism according to claim 1 in which a wall is provided in the casing to separate the intake and discharge passages and having an aperture therein, and in which a by-pass valve is provided in said

wall to limit the fluid pressure in one passage as compared to that in the other passage, said by-pass valve being in the form of a flap and extending through the aperture, one end of said flap being attached to one side of said wall and when closed, the other end of said flap resting against the other side of said wall.

6. A rotary fluid mechanism according to claim 1 in which the teeth of said pinion gear and said impeller differ in number by more than one.

7. A rotary fluid mechanism according to claim 1 in which the teeth of said pinion gear and said impeller differ in number by two.

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