GEAR TYPE PUMP OR MOTOR

Filed Nov. 29, 1966 3 Sheets-Sheet 1

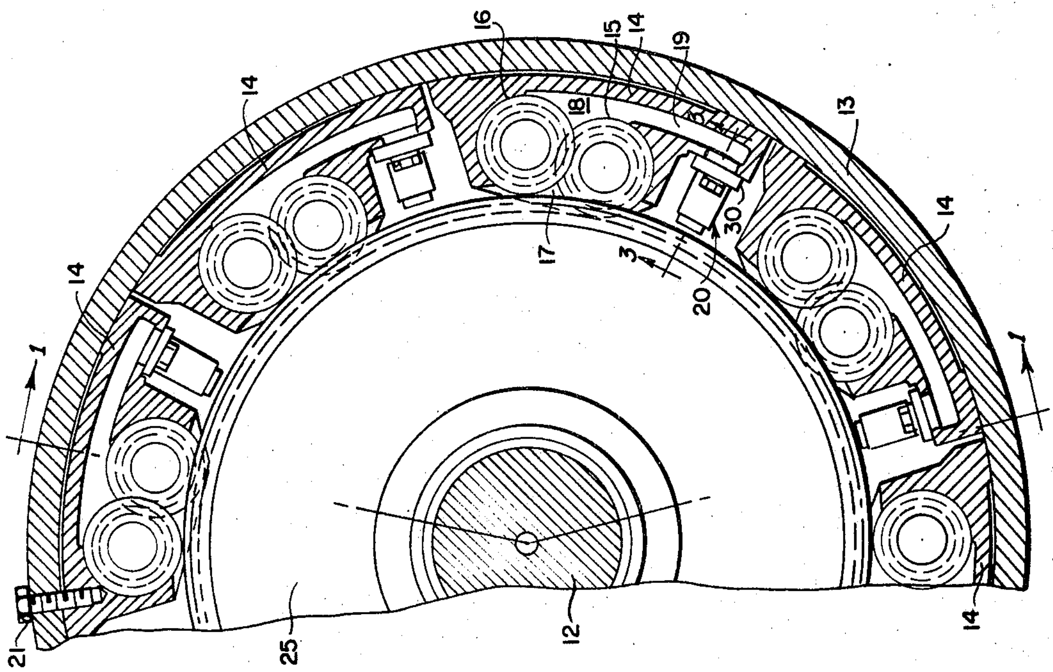

FIG. 2

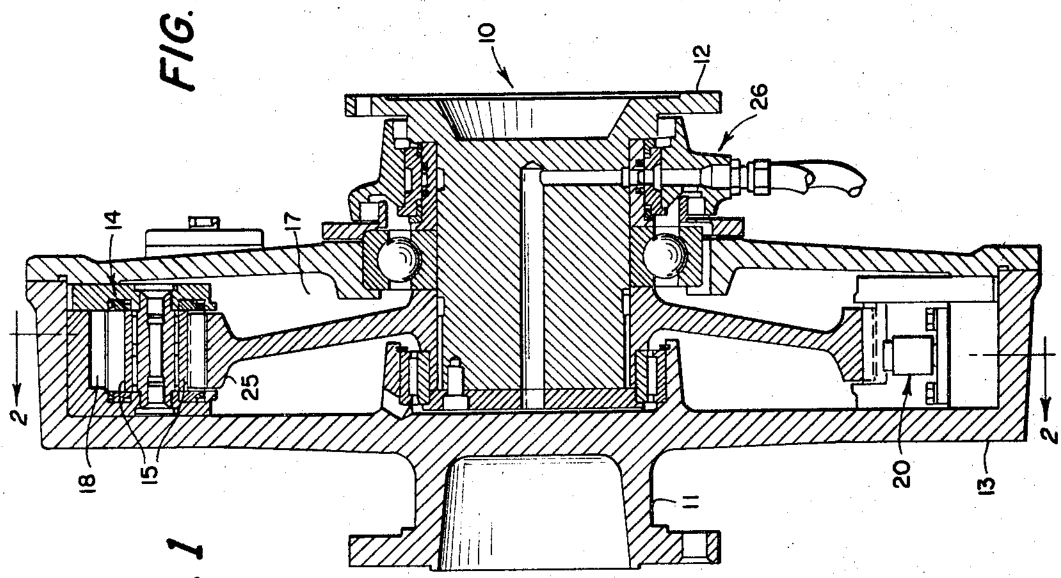

FIG. 1

INVENTOR
KARL GUSTAV AHLEN

BY Larson and Taylor
ATTORNEYS

Sept. 17, 1968 K. G. AHLEN **3,401,778**

GEAR TYPE PUMP OR MOTOR

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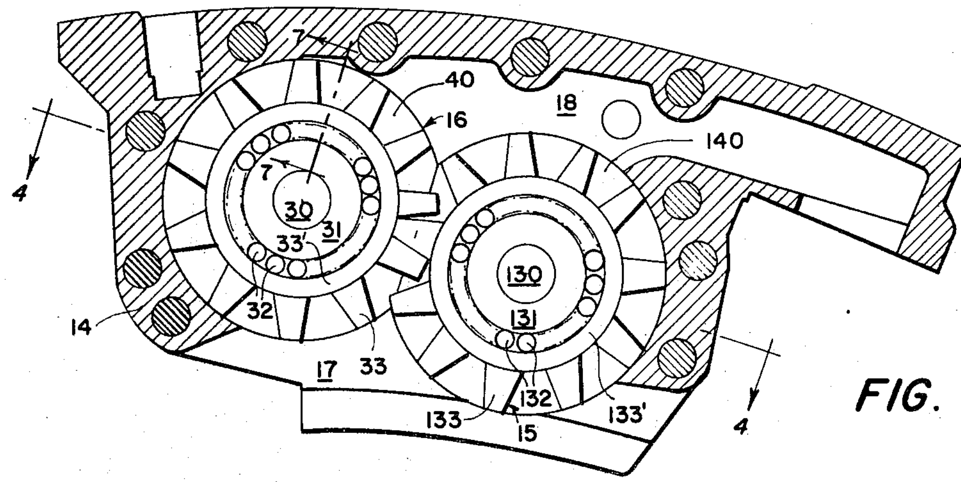

FIG. 3

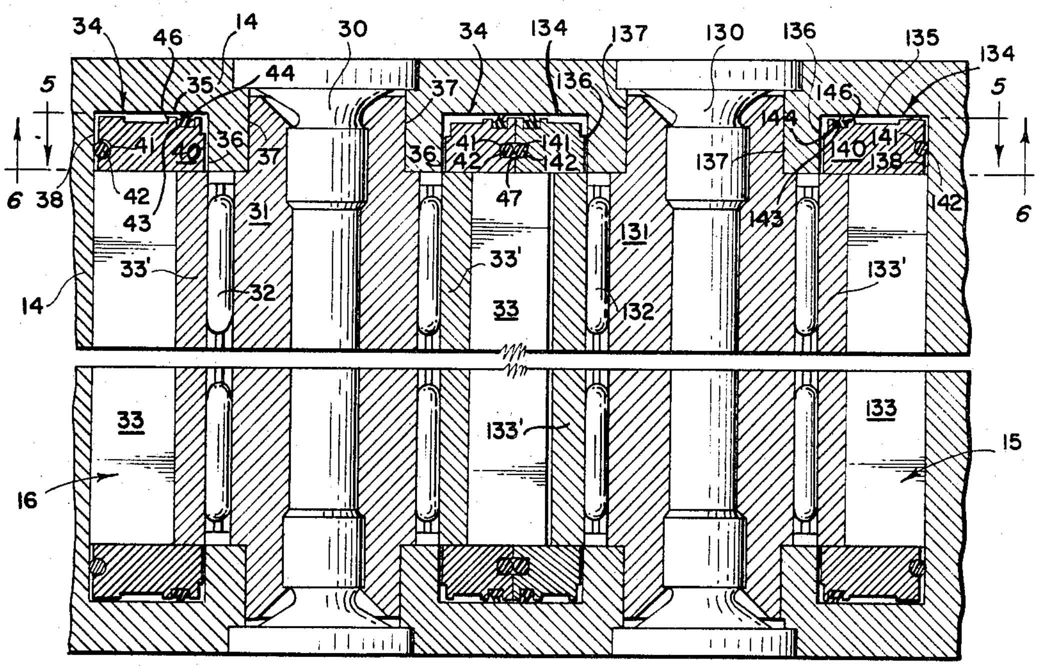

FIG. 4

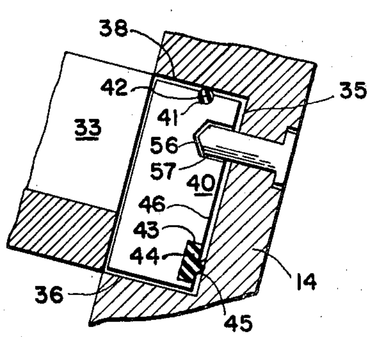

FIG. 7

INVENTOR
KARL GUSTAV AHLEN

BY Larson and Taylor
ATTORNEYS

Sept. 17, 1968 K. G. AHLEN 3,401,778

GEAR TYPE PUMP OR MOTOR

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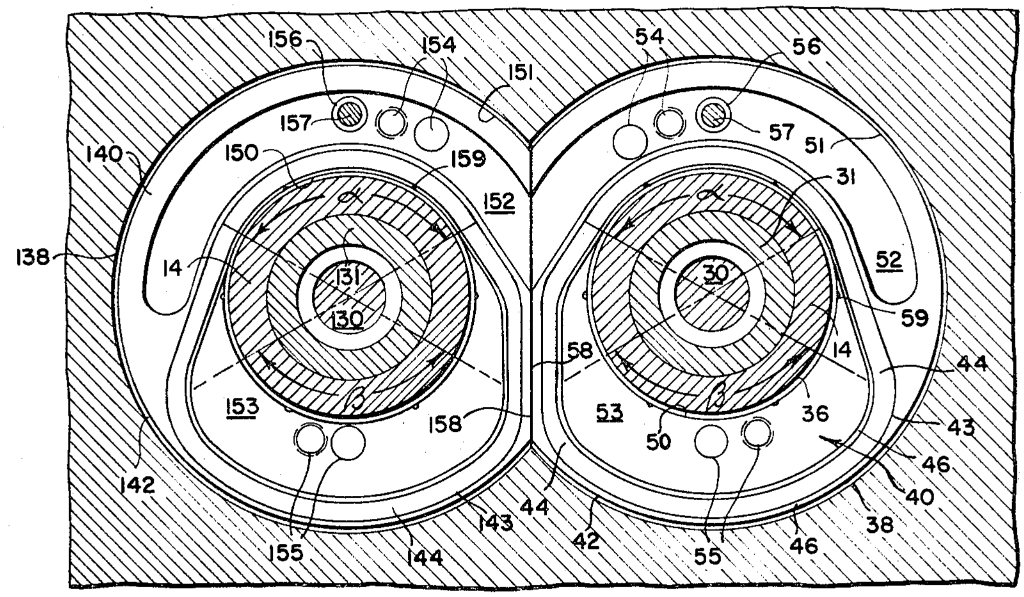

FIG. 5

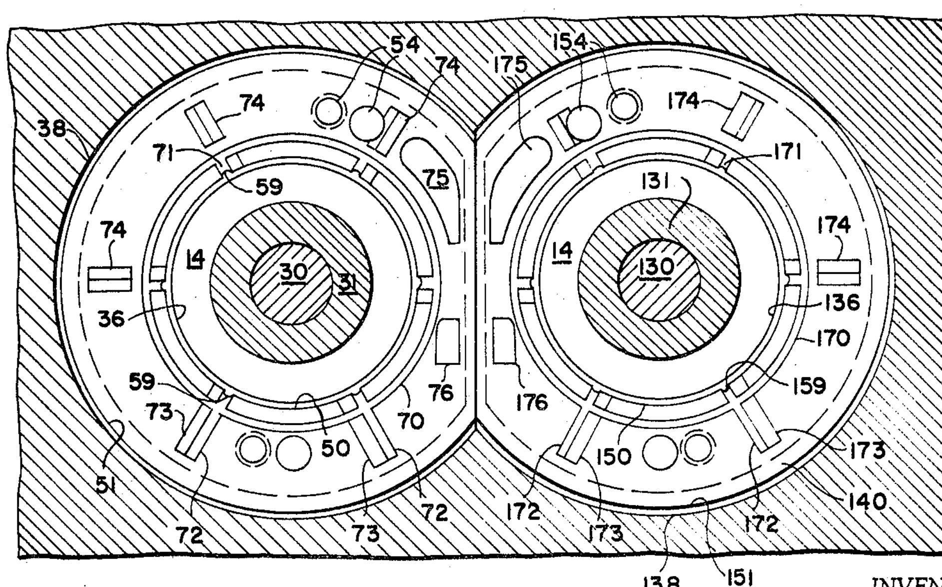

FIG. 6

INVENTOR

KARL GUSTAV AHLEN

BY Larson and Taylor

ATTORNEYS

United States Patent Office

3,401,778

Patented Sept. 17, 1968

3,401,778

GEAR TYPE PUMP OR MOTOR

Karl G. Ahlen, Stockholm, Sweden, assignor to S.R.M. Hydromekanik AB, Stockholm, Sweden

Filed Nov. 29, 1966, Ser. No. 597,631

Claims priority, application Great Britain, June 28, 1966, 28,876/66

19 Claims. (Cl. 192—61)

This invention relates to an improved gear pump or gear motor, and in particular it relates to an improved end plate arrangement for a gear pump or motor.

In the operation of a gear pump or a gear motor it is essential for efficient operation that a tight sealing relationship be maintained between the axial ends of the gears and the end walls of the pump or motor. To this end it is known to employ axially movable end plates and to exert a force on the end plates urging them against the adjacent ends of the gears.

However, even with such end plates, it is still difficult to obtain and maintain a completely satisfactory sealing relationship at all times between end plates and gear ends under high pressure conditions or under conditions where the shape of the elements of the pump or motor are affected by differences in temperature.

It is to be understood that the present invention is applicable to both hydraulic pumps and motors and to pneumatic pumps (compressors) and motors (expanders). However to simplify the description of the invention it will be discussed with particular reference to a pump.

One application where the fluid device of the present invention would be most useful would be in a hydrostatic coupling device of the type disclosed in my Patent No. 3,258,093, issued June 28, 1966. This patent discloses a basic hydrostatic coupling device suitable for use at high rotative speeds. However even in the arrangement disclosed therein, many of the problems encountered as a result of the high stresses and the significant centrifugal forces acting on the gear pump units have not been entirely overcome. For example, when the hydrostatic coupling rotates at high speeds centrifugal forces cause misalignment of the various elements of the pump. Consequently the original precise sealing relationship between the various elements of the pump is seriously disturbed. Another consequence of this misalignment between the various pump elements is that the wear on the elements, particularly on the end plates, is substantially increased. By employing an improved gear pump having the features of the present invention the operation of the hydrostatic coupling can be greatly enhanced.

In a gear type device subjected to these conditions it is desirable to mount the gears and the shafts on which the gears are mounted as rigidly as possible so that the gears will rotate about the same geometrical axis despite the high centrifugal forces and high frictional stresses created in the environment of the high speed hydrostatic transmission.

Heretofore, to construct a strong and rigidly mounted pump gear and shaft with sealing plates it was necessary to employ either a very long gear shaft or large shaft bearings. However, since it was not possible to eliminate all movement of the elements of the pump from the true geometrical axis, with the previous rigidly mounted pump gear structures it was necessary to increase the clearance between the ends of each pump gear and the cooperating end sealing plates. Without this clearance the

end plates would not function properly owing to the variations in alignment between the ends of the pump gears and the sealing plates under changing load conditions.

In the present invention there is provided an improved arrangement in gear pumps or motors in which the effects of high centrifugal forces and other forces resulting from high pressure operation and temperature variations acting unevenly on the pump elements are substantially reduced. First, the gears and the gear shafts in the pump of the present invention are mounted very rigidly to assure that the gears will rotate about the same geometrical axis as much as possible. The shaft bearings are located outside of the pumping area and rows of needle bearings are provided between each shaft and its respective gear. This arrangement provides rigidity in an extremely compact space. Therefore, since the mass of this extremely rigid gear and shaft structure has been substantially reduced, the effects of centrifugal force on these elements will be reduced proportionally.

With the present invention, however, there is provided, in addition, a radically different type of end plate structure which will provide precise and extremely improved sealing at the ends of the rigidly mounted gears under all load conditions to which the gear pump is subjected.

Basically, there is provided, according to the present invention, an end plate which floats in a recess so that it can change its orientation freely (within predetermined limits) with a universal, or three dimensional, movement to maintain its alignment with the end of the gear at all times, even when the gear is moved away from its original geometrical axis by the unevenly applied forces. The end plate also includes a pressure means for exerting a force on the end plate urging the same towards the gear. This pressure means includes structure to compensate for the varying pressures applied to different portions of the end plate on the side thereof facing the gears.

Thus, it is a purpose of the present invention to provide an improved gear pump or motor having improved end sealing plates and providing, in addition to a means for compactly and rigidly mounted pump gear and shaft, a universally movable end plate. Because of this universal movement, metal to metal contact between the end plate and the end plate mounting structure is virtually eliminated.

Briefly, the structure of the gear pump or motor end plates according to the present invention is as follows. As noted above, the gears and gear shafts are mounted compactly and rigidly within the gear pump or motor housing. In the housing, adjacent the ends of the gears, there are provided annular recesses adapted to receive the end plates which are annular in shape. That is, there are four annular recesses, one at each end of each gear. The two annular recesses at the same end of the pump overlap in the area where the gear teeth mesh. It is to be noted that the inner circumference of these annular recesses in the housing are spaced radially outwardly from the housing aperture through which the gear shafts pass. Therefore, the gear shafts are mounted independently of the end plates.

A substantially annular shaped end plate is received in each of the four annular spaces. The floating universal movement of each end plate in its recess is provided in the following manner. A first sealing ring is mounted in a groove in the rear side of the annular end plate (that is, the side facing away from the gear). This ring performs two basic functions. First, it separates the rear side of the end plate into two sections for a reason to be discussed below. In addition, this sealing ring spaces the end plate from the bottom of the recess, thereby avoiding metal to metal contact between the rear of the end plate and the bottom of the recess. The outer circumference of the end plate also has a circumferential groove therein. A second sealing ring having a diameter larger than the depth of the groove is placed in this groove to space the outer circumference of the end plate from the outer circumference of the annular recess. Consequently metal to metal contact between the outer circumference of the end plate and the recess is avoided. The inner circumference of the end plate is of a diameter slightly larger than the inside diameter of the recess. Consequently, the maximum contact between the inner circumference of the end plate and recess will be a line contact which would occur when the end plate is perpendicular to the gear shaft axis. When the end plate tilts relative to the gear shaft axis, then the end plate will contact the inner circumference of the recess only at two points.

Thus, it can be seen that the present invention provides an end plate which virtually floats within the recess having metal to metal contact with the recess at most along one axial line and at least at only two points on the inner circumferencial surfaces of the end plate and recess.

The operative advantages of this floating unversally mounted end plate are substantial. It will allow an excellent seal to be maintained at the ends of the gears despite unavoidable changes in orientation of the pump elements resulting from centrifugal forces and other forces to which the elements of the pump or motor are subjected.

The invention also includes means for providing pressure forces acting on the rear side of the end plate urging the same toward the gears. This is accomplished by dividing the area behind the end plate into two portions, a high pressure portion and a low pressure portion, behind the high pressure chamber and the low pressure chamber of the device respectively. The seal between the two portions is provided by the said second sealing ring which separates the rear of the end plate from the bottom of the recess.

In addition, appropriate grooves are provided on the front side of the end plate, that is, the side facing the gears, for receiving fluid to lubricate the front side of the end plates. The portions of the front side near the inner circumference of the end plate and lying within the low pressure chamber are lubricated by low pressure fluid while the portion of the front face lying in the high pressure chamber is lubricated by high pressure fluid. The front face of each end plate also has formed thereon large improved trap pocket release ports to collect fluids forced from between the teeth of the gears as they intermesh.

As noted earlier it is desirable to eliminate the effects of centrifugal force acting upon all of the elements, including the end plates. Therefore, it would be desirable to reduce the mass of the end plates by making them as thin as possible. If the two sealing rings could lie in the same radial plane, then the thickness necessary for only one of the two sealing ring grooves would determine the thickness of the end plate. However, for reasons to be discussed in more detail below, it is desirable that the second sealing ring have at least one portion located very close to the outer circumference of the rear end plate surface. Thus, the two sealing rings will overlap at one portion of the plate when viewed in the axial direction. Therefore, the thickness of the end plate should be just large enough to allow for both of the sealing ring grooves.

It is also desirable to design the end plates to have a fairly uniform overall shape, free of unnecessary and non-uniform projections. With that construction, centrifugal force will act upon the entire end plate equally, whereas if projections were provided, then centrifugal force would act to a greater extent on the side of the end plate having the projection, thereby tending to unnecessarily tilt the end plate.

It should be apparent that the end plate of the present invention can be made much more economically than previous end plates. First, the cost of material is reduced since the end plate would be as thin as possible requiring only sufficient thickness for the sealing ring grooves. Secondly, since metal to metal contact is substantially reduced, the plates may be manufactured by means of pressure die casting leaving only one operation to be made by way of machining, namely, the planing of the front surface of the end plate which is to be placed adjacent the gear.

Thus it is an object of this invention to provide a universally movable floating end plate for a gear pump or motor.

It is another object of this invention to provide a gear pump for a hydrostatic coupling wherein the efficiency of the gear pump will be retained even when the pump is subjected to very high centrifugal forces.

Another object of this invention is to provide an improved gear pump or gear motor driven by fluids such as oil or water but also by gas such as steam compressed air or working as an expander wherein the end sealing plates are arranged to float within a recess in the wall of the pump or motor housing thereby virtually eliminating metal-to-metal contact.

It is another object of this invention to provide an improved hydrostatic coupling wherein the gear pump units thereof retain optimum operating efficiency even during high rotational speeds of the coupling.

Other objects and the attendant advantages of this invention will become apparent from the detailed description to follow of a preferred embodiment of the invention, together with the accompanying drawings, which describe and show the invention in the form of a hydraulic gear pump incorporated in a hydrostatic coupling.

FIGURE 1 is a partial sectional view of a hydrostatic coupling of the gear pump type.

FIGURE 2 is a sectional view of a hydrostatic coupling taken along line **2—2** of FIGURE 1.

FIGURE 3 is a sectional view of a gear pump of the type employed in the hydrostatic coupling of FIGURES 1 and 2.

FIGURE 4 is a sectional view of the gear pump taken along line **4—4** of FIGURE 3.

FIGURE 5 illustrates the rear side of the end sealing plates of the gear pump taken along line **5—5** of FIGURE 4 but reversed vertically so that the top of the plate in FIGURE 5 will correspond to the top of the plate as viewed in FIGURE 3 and FIGURE 6.

FIGURE 6 illustrates the front side of the end sealing plates taken along line **6—6** of FIGURE 4.

FIGURE 7 is an enlarged partial sectional view taken along line **7—7** of FIGURE 3.

Referring first to FIGURES 1 and 2 there is shown a hydrostatic coupling **10** for transmitting power from a first shaft, for example, driving shaft **11**, to a second shaft, for example, shaft **12**. The housing **13** is rigidly connected to the first shaft **11** and has rigidly formed therein a plurality of gear pump units **14**. These pump units are comprised of main pump gears **15** and idler pump gears **16**. When operative, these pump units pump fluid from the low pressure portion **17** of the housing to high pressure chamber **18** from which the fluid flows through passage **19** to valve **20**. When the valve **20** is open the fluid flows back to the low pressure portion **17** and when the valve is closed the pressure in chambers **18** and **19** builds up until the gears of the pump can practically no longer turn. An inner sun gear **25** is rigidly connected to the shaft **12** and in driving engagement with the main gear **15** of each pump unit **14**. When the valve **20** is wide open the rotational movement of shaft **11** and housing **13**, relative to gear **25**, will cause pumping of liquid from **17** to **18**. However the housing will merely continue to rotate about the sun gear **25** as the main pump gears **15** roll around gear **25** in planetary fashion. However, if valves **20** are closed, pressure will build up in chamber **18** until the pump gears practically can no longer turn. Rotation of the housing **13** will then cause simultaneous rotation of the sun gear **25** through its engagement with non-turning main pump gears **15**. The amount of torque transmitted from the outer housing to the sun gears **25** can to a large extent be varied and controlled by regulating the opening pressure provided by valve **20**.

FIGURES 3 through 7 illustrate the detailed arrangement of the gear pump and, in particular, the end plate construction according to the present invention. As noted earlier the two gears of the gear pump differ only in that one is in engagement with the sun gear **25** while the other is an idler gear. Otherwise, the structure, purpose and operation of the two gears and their associated end plates are identical. Therefore, the same numerals will be employed to designate like parts on the two separate gears except that the numerals associated with gear **15** will be in the 100 series and only the structure and operation of the elements associated with gear **15** will be discussed in detail.

Referring now to FIGURES 3 and 4, and for ease of description, in particular to the gear **16** at the left hand side of FIGURES 3 and 4, there is shown a gear shaft **30** surrounded by a sleeve **31**. Needle roller bearings **32** separate the sleeve **31** from an annular toothed pump gear **33**. A substantially annular recess **34** is provided in the pump housing **14** adjacent the end of the toothed pump gear **33**. This recess is formed by a flat rear wall **35**, an inner wall formed by a cylindrical hub **36** and a substantially cylindrical outer peripheral wall **38**. The inner wall **36** of the recess is separated from an aperture **37** in the pump housing **14** through which the shaft **30** and sleeve **31** project. Thus the recess **34** is separated from the portion which supports the shaft **30** and sleeve **31** of the gear.

Mounted within the recess **34** is an end sealing plate **40** having an annular peripheral groove around the outer periphery thereof, which groove has an O-ring **42** mounted therein. This O-ring may be of a synthetic material such as Viton. A second groove **43** is formed in the rear of the plate **40** which groove has mounted therein a rear sealing ring **44** having a rearwardly extending lip **45**. Ring **44** may also be of a synthetic material such as Viton.

The end plates were described above as being substantially annular. However, referring to FIGURES 5 and 6 it can be seen that the end plates actually abut along a straight line in a plane lying midway between the two gears. Thus, FIGURE 4 shows the two O-rings **42** and **142** contacting each other where the plates abut. To provide an additional seal in this area, a plate **47** is placed into the large recess formed by grooves **41** and **141**.

It can thus be seen that the end plate **40** floats in recess on rings **42** and **44** avoiding metal to metal contact with the recess except at a point or line contact on hub **36**. Thus the plate can tilt about its axis as one side of the plate moves out of the recess and ring **42** slides along surface **38** away from bottom surface **35** and the other side of the plate moves into the recess and the other side of ring **42** moves into the recess.

Referring now to FIGURES 5 and 6 the details of the end plates are shown from the rear and from the front side respectively. Referring to FIGURE 5, and in particular to the end plate associated with gear **16**, there is shown the end plate **40** having an inner perimeter **50** spaced from hub **36** and an outer perimeter **51** spaced from the outer peripheral wall **38** of the recess by the outwardly extending portion of O-ring **42**. Raised portions **46** of the plate **40**, together with the rearwardly extending lip **45** on O-ring **44** divides the rear of the plate into two portions, a high pressure pocket **52** behind the high pressure portion of the gear pump and a low pressure pocket **53** (which includes the entire central hub **36**) behind the low pressure portion of the gear pump. Through holes **54** connect pocket **52** to the high pressure side of the pump while through holes **55** connect to the pocket **53** to the low pressure side of the pump. With this construction high pressure fluid will be led to pocket **52** behind the high pressure side of the pump while low pressure fluid will be led to pocket **53** behind the low pressure side of the pump. Therefore, the pressure forces exerted from behind the sealing plates will be equal to those pressure forces acting on the front side thereof so that the net effect will be to urge the sealing plate against the end of the gear in a direction parallel to the gear axis.

It is to be noted that a slight clearance is provided between the surface **36** and the surface **50**. Because of this clearance, these two surfaces will contact only at a single point when the plate is tilted or along a single line when the plate is perpendicular to the gear shaft axis.

It is necessary to provide sufficient clearance within the recesses for the sealing plates to float therein on the rings **42** and **44**. On the other hand, however, the plates must be restricted against rotation about their own axes. Such rotation of the plates is prevented by the abutting surfaces **58** and **158**, and also by pins **57** projecting from housing **14** into apertures **56**. See FIGURE 7.

Notches **59** are provided in the inner perimeter **50** of plate **40** for providing communication of low pressure fluid from the pocket **53** to the front surface of the end plate for reasons to be discussed in more detail below.

FIGURE 5 also illustrates the preferred path of the sealing ring **44**. The groove for this sealing ring at the pressure side follows the inner periphery **50** of the end plate **40** along an arc indicated by the angle α of approximately 80° to 100°. Along the outer perimeter **51** of the end plate **40** adjacent the low pressure side thereof the sealing ring follows a curved path represented by angle β for approximately 90° to 100°. It is desirable to place the seal **44** as close to the edge **51** as possible to provide a large pressure pocket **53** having a large moment acting to prevent tilting of the plate by offsetting the tilting moment created by high pressure pocket **52**. The grooves also follow the abutting edges **58** and **158**. Those portions of the groove between the curved portions are straight line connections.

Because the ring **44** must extend very close to the outer perimeter of the plate **40**, the two rings **42** and **44** shall not lie in the same radial plane, and the plate must be thick enough to accommodate both grooves **41** and **43**. See FIGURE 4. Aside from this factor, however, the plates should be as thin as possible to reduce the effects of centrifugal force urging the plates outwardly as much as possible.

FIGURE 6 illustrates the front of the end plates and the flow of liquid thereto for purposes of lubrication.

Circular grooves **70** are in communication with notches **59** by radial grooves **71** thereby passing low pressure lubricating fluid around the entire inner periphery **50** and also to extensions **72** of grooves **7** to provide lubricating oil on the low pressure side of the pump in pockets **73**. Similarly, high pressure pockets **74** are provided at the high pressure side of the pump.

When the gear pump is employed as a compressor or pneumatic motor (expander) liquid will not be available to lubricate the contact surfaces between the end plates and the ends of the gears. In this instance these contact surfaces can be formed with an anti-friction material.

Another important feature of the invention is the large trap pocket release ports **75** provided at the high pressure side of the pump. These trap pocket release ports are larger than trap pocket release ports normally employed. The reason for this is that since under some conditions the speed of the gears will be very high it will also be necessary to allow an increased volume of fluid to escape from the trap pockets between the meshing gear teeth with limited pressure drop. The pressure in the trap pocket increases with the square of the speed and therefore these large pocket release ports will decrease the resistance to oil moving out of the meshing gear teeth. It must be kept in mind that at times these gears will rotate at speeds greater than normal pump speeds. If the trap pocket release ports are not sufficient, pressure chokes strong enough to destroy the bearings will occur. Trap port **76** is also provided on the low pressure side of the plate.

Summarizing, there has been provided herein a gear pump or motor having an end plate of specified design so that the pump or motor will function nearly independently of the effects of centrifugal force or other adverse affects caused by high pressure operation or temperature variations. The compactness which allows a minimum distance between the support points of the gear shaft and extremely high peripheral speeds of the gears can be allowed together with the improved end plate for assuring an excellent seal at the end of the plate. Fluid pressure urges the plates towards the gears where the seals are retained against the gear ends in near perfect sealing relationship because of the floating mounting of the end plates in their recesses.

Although the invention has been described in considerable detail with respect to a preferred embodiment thereof, it should be apparent that the invention is capable of numerous modification and variations for different types of uses entirely within the spirit and scope of the invention as defined in the appended claims, wherein I claim:

1. A pump or motor comprising a casing, a low pressure chamber and a high pressure chamber, a pair of rotary intermeshing gears mounted in the casing for carrying fluid from one of said pressure chambers to the other pressure chamber, at least one pair of end plates mounted in said casing at least one end of said pair of gears, and in sealing engagement with the ends of the gears, means for mounting said end plates in said casing for universal movement whereby the said end plates are capable of maintaining their axes perpendicular to the said ends of its meshing gears as the position of the axes of the said gears change with respect to the casing.

2. A pump or motor as claimed in claim **1** including a means for exerting a pressure force on the end plates urging them against the said ends of the said gears.

3. A pump or motor as claimed in claim **2** including a pair of substantially annular recesses in said casing adjacent the said ends of the said gears, said end plates being substantially annular and being mounted one within each said recess, each said recess having a substantially annular bottom surface, a substantially cylindrical outer surface and a substantially cylindrical inner surface, resilient means for spacing each said end plate from the said bottom surface and from the said outer surface of its recess, and the inner circumference of each said end plate being of a larger diameter than the inner surface of its recess, whereby each said end plate floats on said resilient means within its recess whereby said universal movement is obtained.

4. A pump or motor as claimed in claim **3** wherein said resilient means includes a first resilient ring spacing the outer circumference of each end plate from the outer surface of its recess, and a second resilient ring in the rear side of the end plate facing the bottom of its recess spacing the end plate from the bottom of its recess.

5. A pump or motor as claimed in claim **4** wherein said second resilient ring divides the rear side of the end plate into a high pressure pocket and a low pressure pocket, said high pressure pocket being located across the end plate from the high pressure chamber of the pump, and the said low pressure pocket being located across the end plate from the low pressure chamber of the pump.

6. A pump or motor as claimed in claim **5** wherein the said second resilient means includes a raised lip portion adapted to contact the bottom of the recess and adapted to be bent to diminish the height of the second sealing ring in a direction axially from the rear side of the end plate.

7. A pump or motor as claimed in claim **5** wherein the said low pressure pocket comprises the area between the end plate and the bottom of the recess bounded by the second resilient ring and the said high pressure pocket lies outside of the area bounded by the second resilient ring, said second resilient ring having a portion extending for more than 90° adjacent to the outer circumference of the end plate across from the low pressure chamber.

8. A pump or motor as claimed in claim **7** wherein the area bounded by the second sealing ring includes the inner circumference of the end plate.

9. A pump or motor as claimed in claim **8** including a means for directing fluid from the low pressure pocket to the front side of the end plate facing the gear to lubricate the said front side.

10. A pump or motor as claimed in claim **5** in which the size of the high pressure pocket between the end plates and the bottom of the recess is bounded by the second resilient ring and said secondary resilient ring has a portion extending approximately for 180° adjacent to the outer circumference of the end plate across from the low pressure chamber.

11. A pump or motor as claimed in claim **4** wherein each of the said rings is located within separate grooves in the end plate and the thickness of the end plate is slightly greater than the sum of the axial dimensions of the two grooves.

12. A pump or motor as claimed in claim **4** wherein the said end plates mounted at the same end of the pump or motor abut in the plane midway between the axes of the two gears and including a guard plate member for sealing the pump or motor chambers where the two plates abut.

13. A pump or motor as claimed in claim **3** including a means for preventing rotational movement of said end plates, said means including a pin extending from the bottom of each recess into an aperture in the side of the plates facing the bottom of its recess.

14. A pump or motor as claimed in claim **3** including a trap pocket release port on the side of the end plates facing the gears and extending circumferentially for more than 45°.

15. A pump or motor as claimed in claim **1** in which the contact surfaces between the ends of the gears and the end plates contain solid anti-friction elements.

16. A hydrostatic coupling device of the gear pump type for transmitting power from a first shaft to a second shaft, wherein one shaft has rigidly connected thereto a plurality of gear pump units planetarily mounted about and engaged with a sun gear connected to the other shaft; wherein said gear pump units comprise a casing, a low pressure chamber and a high pressure chamber, a pair of rotary intermeshing gears mounted in the casing for carrying fluid from the low pressure chamber to the high pressure chamber, at least one pair of end plates mounted in said casing at least one end of said pair of gears, and in sealing engagement with the ends of the gears, means for mounting said end plates in said casing for universal movement whereby the said end plates are capable of maintaining their axes perpendicular to the said ends of the gears as the position of the axes of the said gears change with respect to the casing when subjected to centrifugal forces created when the said one shaft of the hydrostatic coupling rotates about its axis.

17. A hydrostatic coupling device as claimed in claim **16** including a means for exerting a pressure force on the end plates urging them against the said ends of the said gears.

18. A hydrostatic coupling device as claimed in claim **16** including a pair of substantially annular recesses in said casing adjacent the said ends of the said gears, said end plates being substantially annular and being mounted one within each said recess, each said recess having a substantially annular bottom surface, a substantially cylindrical outer surface and a substantially cylindrical inner surface, resilient means for spacing each said end plate from the said bottom surface and from the said outer surface of its recess, and the inner circumference of each said end plate being of a larger diameter than the inner surface of its recess, whereby the said resilient means allows each said end plate to float within its recess whereby said universal movement is obtained.

19. A hydrostatic coupling device as claimed in claim **16** including a trap pocket release port on the side of the end plates facing the gears and extending circumferentially for more than 45°.

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EDGAR W. GEOGHEGAN, *Primary Examiner.*