Dec. 6, 1966 B. C. HUDGENS 3,289,602

FLUID PRESSURE DEVICE

Filed Sept. 3, 1965 2 Sheets-Sheet 1

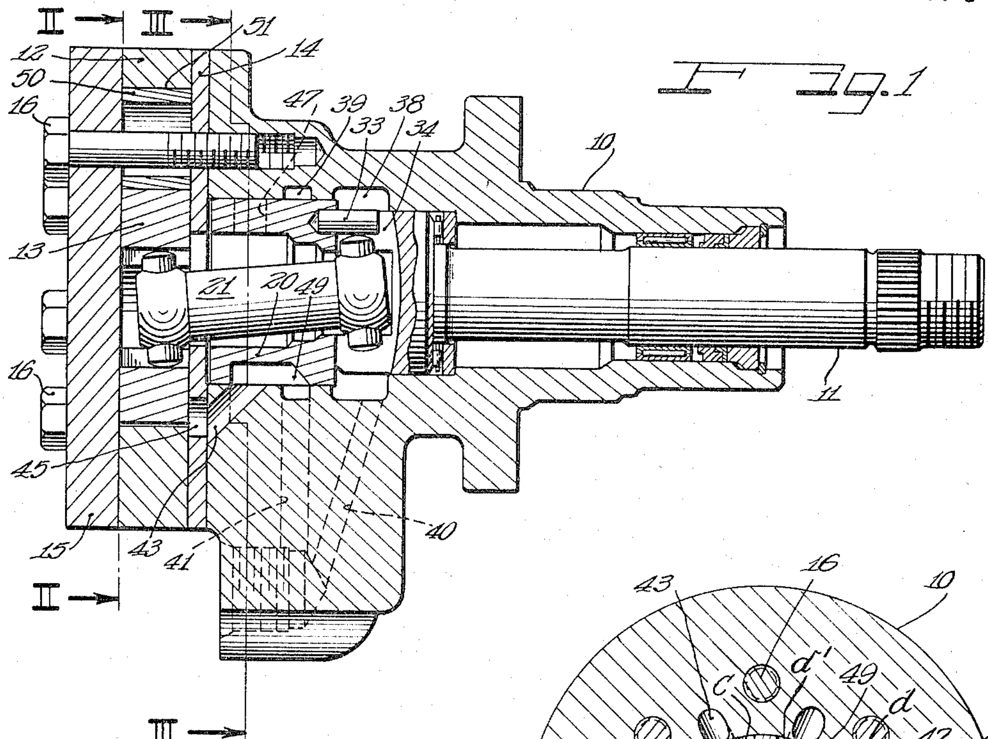

Fig. 1

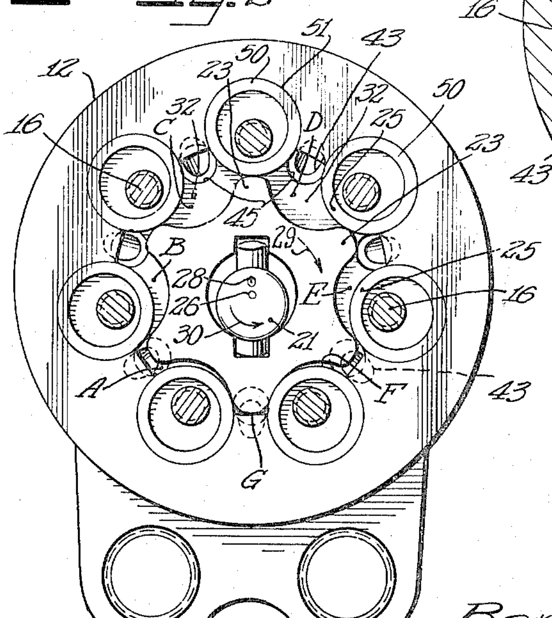

Fig. 2

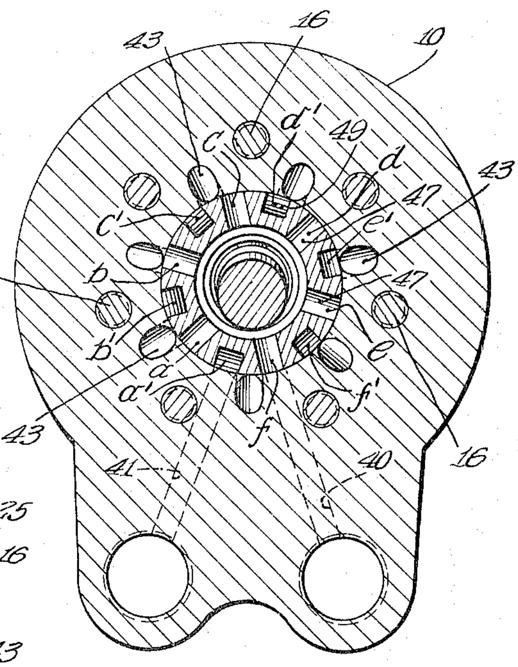

Fig. 3

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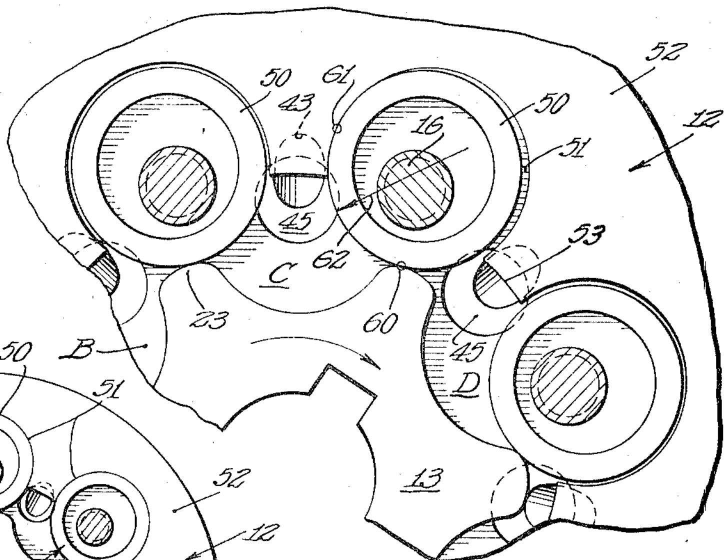
Fig. 5



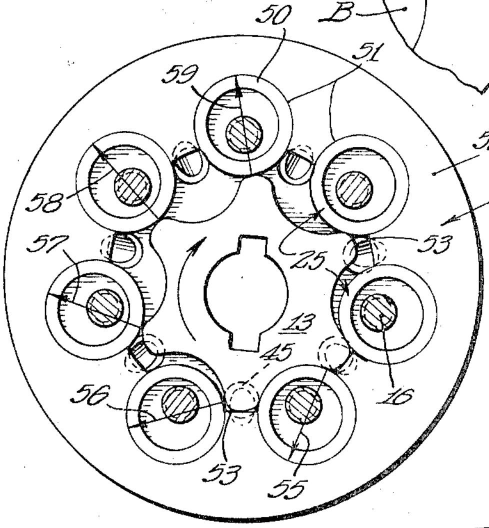


Fig. 4

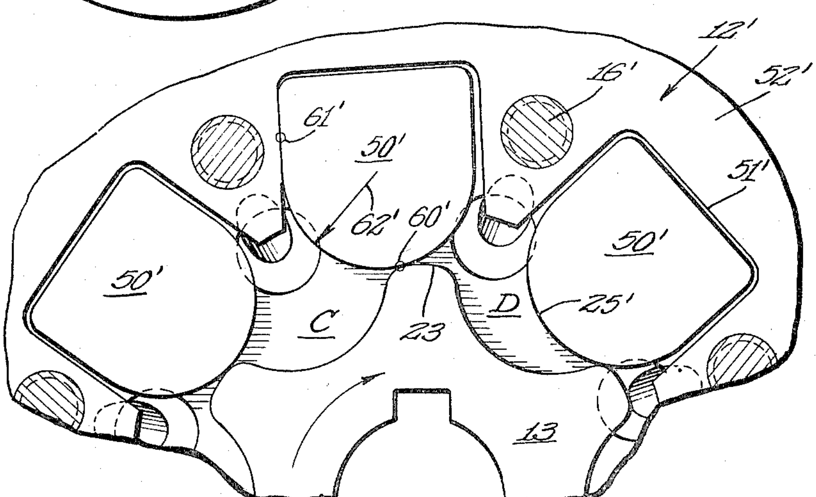


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Fig. 6

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3,289,602
FLUID PRESSURE DEVICE

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13 Claims. (Cl. 103—130)

This invention relates to a fluid pressure device, including a pump or a motor and more particularly deals with a hydraulic pump or motor of the internal gear type wherein the outer gear has one more tooth than the inner gear and one of the gears orbits about the axis of the other gear when it is rotated about its own axis and wherein the teeth or lobes of one of the gears are individual units accommodating wide tolerance variations and automatically sealed relative to the gear body under the influence of hydraulic pressure developed between the gears.

This application is a continuation-in-part of my copending application entitled "Hydraulic Device," U.S. Serial No. 352,019, filed March 16, 1964.

In my aforesaid patent application, it is pointed out that the "internal gear set" of a pump or motor of the type having an orbiting rotating element is costly to produce, requiring close tolerances for sealing and free turning of the parts. In order to reduce the manufacturing costs and accommodate wider tolerance variations, the hydraulic device of the aforesaid application is equipped with a vane carried by each tooth of the rotor of the internal gear set. These vanes are spring pressed against the teeth of the enveloping gear of the internal gear set, and since they can move radially relative to the rotor teeth, sealing contact between the gears of the internal gear set is insured even though the two sets of teeth are not accurately lapped into interfitting sealed relation.

The present invention now provides moveable vane elements or teeth segments in the outer gear of the internal gear set and eliminates the heretofore used spring members. The vanes or tooth segments of the present invention may take the form of rollers, shoes, or the like individual pieces freely carried in recesses in the outer or enveloping gear of the internal gear set. The segment receiving recesses are larger than the segments to permit movement of the segments relative to the gear body, but at least a portion of the recess walls are adapted to be mated with the segments to provide a good sealing engagement of the segments with the gear body. The coacting sealing surfaces of the segments and gear body can be shifted to accommodate wide tolerance variations and wear of the parts in use. The segments can be provided with hardened surfaces to resist wear while the gear body carrying the segments can be composed of a less hardened high strength metal which is easy to machine.

The invention will hereinafter be specifically described as embodied in an internal gear pump having an orbiting rotor embraced by a stator where the rotor is driven through a wobble stick connection with the drive shaft and a driven cylindrical valve surrounding the wobble stick controls fluid flow through the pump. It will be understood, however, that the principles of this invention are generally applicable to fluid pressure devices embodying internal gear sets. The illustrated device is especially useful in power steering linkages for controlling the power steering motor from a steering wheel. The invention is not limited to the hereinafter described specifically illustrated embodiment.

It is an object of this invention to provide a fluid flow device with an internal gear set having individual tooth segments in at least one of the gears of the set.

Another object of the invention is to provide a fluid pressure device with an internal gear set having the lobes of the enveloping gear formed from individual segments loosely carried by the enveloping gear body and shiftable in response to fluid pressure developed by the set into sealing engagement with the gear body.

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Another object of the invention is to provide vanes on the enveloping stator of an internal gear set, which vanes will accommodate wide tolerance variations in the manufacture of the gear set and will create their own seals under the influence of pressure developed by the device.

A specific object of this invention is to provide the stator of an internal gear pump of the type having an orbiting rotor with individual lobes shiftable in the stator into proper operating positions regardless of wear developed between the gears.

Another specific object of this invention is to provide an internal gear pump with a stator having recesses carrying cylindrical rollers in loose-fit relation for engaging the teeth of an orbiting rotor.

Another specific object of the invention is to provide an internal gear pump or motor of the type having an orbiting rotor surrounded by a stator wherein the stator teeth are composed of individual vanes shiftable into proper operating positions under the action of the rotor and pressures developed between the rotor and stator.

Other and further objects of this invention will be apparent to those skilled in this art from the following detailed description of the annexed sheets of drawings, which by way of preferred examples only illustrate two embodiments of the invention.

On the drawings:

FIGURE 1 is a longitudinal cross-sectional view with parts in elevation of a fluid pressure device especially adapted for power steering usage and incorporating the features of this invention;

FIGURE 2 is a transverse sectional view taken generally along the line II—II of FIGURE 1;

FIGURE 3 is a transverse sectional view taken generally along the broken line III—III of FIGURE 1;

FIGURE 4 is a view similar to FIGURE 2, but illustrating the direction of rotor force on the individual shiftable stator lobes;

FIGURE 5 is an enlarged fragmentary view of a portion of FIGURE 4 illustrating the manner in which the pressure differential across the vane or tooth segment creates the seal; and

FIGURE 6 is a view similar to FIGURE 5, but illustrating a modified vane construction.

As shown on the drawings:

The present invention as mentioned hereinabove relates to the new and improved means for sealing between the rotor element and stator gear element of the internal gear set which is utilized in a hydraulic device. In order to properly describe the environment within which the present invention resides, it is necessary to discuss to some length the construction of the hydraulic device. To this end, the device illustrated, particularly in FIGURES 1, 2 and 3, will be described as a pump with those skilled in the art readily recognizing the interchangeability of the device to function as a motor.

The pump which has been shown comprises a housing **10** which receives a drive shaft **11** through a bore therein which shaft can be driven by any appropriate means. The pump elements are located in the housing opposite the shaft **11** or at the left end of FIGURE 1, and these elements comprise an outer fixed stator gear element **12** and an inner rotatable rotor element **13** together forming the "internal gear set." These two elements are located between a wear plate **14** on the left end of the housing **10** and a cover **15** suitably held in position by screws **16** threaded into the left end of the housing.

A commutating valve spool **20** is rotatably located in the bore of the housing between the shaft **11** and the pump elements and through this valve spool extends a shaft or wobble stick **21** which transmits the rotation of shaft **11** to the rotor element **13**. It will be apparent from reviewing FIGURES 1 and 2 that the connection between the one end of shaft **21** and the end of shaft **11** amounts to a universal joint connection as well as the connection of the other end of shaft **21** to the rotor element. The purpose of these connections is to permit the rotor element **13** to partake of its rotational and orbiting movement which will be discussed in more detail hereinafter. In the specific device shown herein, the rotor element is formed with teeth **23**, in this embodiment specifically six in number, and the stator gear element carries one more tooth, sometimes referred to as lobes **25**, in this embodiment specifically seven in number. The rotor element has an axis indicated by the point **26** in FIGURE 2 and the stator element has an axis indicated by the point **28**.

Rotation of the shaft **11** in an appropriate direction will cause the rotor to rotate about its own axis **26** in the direction of arrow **29** (FIGURE 2). However, the rotor axis **26** will orbit about the stator axis **28** in the opposite direction or in the direction of arrow **30**. The result of this action is to cause the pockets **32** formed between the stator lobes **25** to pass through alternately contracting pressure cycles or strokes and expanding intake or suction strokes. In the specific pump shown, namely a rotor element with six teeth and a stator element with seven teeth or lobes, the rotor axis will orbit about the stator axis six times for each complete rotation of shaft **11**, and each orbit will produce seven pressure pulses or strokes. At any one instant of time there are 3+ pockets undergoing an exhaust or pressure stage and 3+ pockets in the intake or suction stage. One revolution of the drive shaft will therefore produce six orbits or 42 pump or pressure pulses.

To facilitate connection of the alternately expanding and contracting pockets **32** with a source of fluid supply and exhaust, the shaft **11** has been connected to the commutating valve sleeve **20** by way of a drive pin **33** which projects from the valve sleeve into one end of a slot **34** in the left end **35** of the drive shaft. This causes the commutating valve spool **20**, the rotor element **13**, and the drive shaft **11** to travel at substantially the same rotational speed.

The bore of the housing **10** is provided with two annular grooves **38** and **39** which communicate respectively with radial passages **40** and **41** opening in the sides of the housing **10** and adapted for connection to conduits through which the fluid enters and leaves the pump. The groove **38** is located in the plane of the slot **34** and therefore is in constant communication with the interior of the commutating valve spool **20** while the groove **39** is located within the axial limits of the valve spool **20**. A circumferential series of ports **43** extend at one end to the bore of the housing and at the other end through holes **45** in the wear plate **14** to the pockets **32** between the lobes of the stator gear element **12**. The commutating valve spool is provided with a plurality of circumferentially spaced radial passages **47**, (further identified by letters *a*, *b*, *c*, *d*, *e*, and *f*) which extend from the outer surface thereof to the hollow inside, and located between these radial passages **47** are provided corresponding numbers of axially extending grooves **49** (further identified by letters *a'*, *b'*, *c'*, *d'*, *e'* and *f'*). These grooves **49** extend axially a length sufficient to provide a fluid bridge between the ports **43** and the annular groove **39** when the grooves **49** and ports **43** are in axial alignment with each other.

In order to give an understanding of the mode of operation of the pump and cooperation of the stator gear element **12** and the rotor element **13** with the commutating valve spool, reference is best had to FIGURES 1, 2 and 3 of the drawings. The sectional views shown

in FIGURES 2 and 3 show the rotor element and the commutating valve spool in the same angular position they occupy relative to each other because of their mechanical connection with the drive shaft **11**. Assuming that the drive shaft **11** is being rotated in a clockwise direction **29** as viewed in FIGURE 2, it will be seen that pockets A, B and C are on an intake stroke, pockets D, E and F are on a pressure stroke, and pocket G is at the point of completing a pressure stroke and just prior to starting an intake stroke. The pockets A, B and C are being connected in FIGURE 3 to the fluid input side of the pump through radial passages *a*, *b* and *c*, whereas pockets D, E and F communicate with the pressure or output side of the pump by way of axial grooves *d'*, *e'* and *f'*, whereas pocket G is in what may be referred to as a transition position being neither connected to the radial passage *f* or axial groove *a'*. From this description it will be apparent how the rotating and orbiting rotor element produces successively alternating pump and intake strokes in the various pockets **32** and how the action of the commutating valve spool appropriately connects the pockets undergoing pump strokes to the pressure side of the pump and at the same time connects the expanding pockets undergoing an intake stroke to the intake side of the pump. It will thus be seen that as a given tooth on the rotor element moves along a curved surface **25** into a pocket **32** to produce a pump stroke, that a given pocket rotationally behind the referred to tooth is producing an intake stroke. Referring to FIGURE 2, it will be seen that the tooth which is undergoing the intake stroke may be immediately behind the tooth producing the pump stroke or it may be the second or third tooth behind depending upon which tooth is taken as the reference tooth, because as described hereinabove, at least three pump pulses are being produced and at least three intake pulses.

If the shaft **11** is rotated in a reverse direction it will be obvious that the condition of the pockets will be reversed and passages *a*, *b* and *c* will be connected to the output side of the pump and grooves *d'*, *e'* and *f'* will be connected to the input side of the pump. The general overall construction of this hydraulic device is shown in U.S. Patent, No. 3,087,436, issued April 30, 1963.

The improvement of the present invention comprises a vane or individual tooth for each of the seven lobes of the stator **12**. In the embodiment of the invention illustrated in FIGURES 1 through 5, these vanes or tooth segments **50** take the form of hollow cylindrical metal tubes each receiving a screw **16** therethrough and each fitting in a somewhat oversized fragmental cylindrical recess **51** in the stator body. The recesses **51** open inwardly to accommodate projection of the tubes **50** to form the lobes **25**. As shown in FIGURE 1, the tubes **50** extend the full distance between the wear plate **14** and the end cover **15**, but they have a sliding fit with the cover and wear plate at their ends so that they can rotate and shift radially in the recesses **51**. The sliding fit between the ends of the tubes or rollers and the wear plate and end cover is such that leakage will be minimized.

The stator **12**, therefore, comprises, as shown in FIGURE 4, a metal ring or apertured disk **52** with a ring of fragmental cylindrical recesses **51** spaced equally around the inner periphery thereof and loosely receiving the rollers or tubes **50** therein. This ring or disk **52** has finger-like portions **53** between each of the recesses **51** providing gaps between the rollers **50** for receiving the stator teeth **23**. About one-third of the periphery of each roller **50** extends inwardly from the recesses **51** to provide the active lobe surfaces **25** for the stator.

As also shown in FIGURE 4, as the rotor **13** rotates and orbits within the stator **12**, its teeth **23** exert forces on the rollers **50** generally indicated by the arrows **55**, **56**, **57**, **58** and **59**. This causes the rollers to shift into tight seating engagement with the oversized recesses **51** in the vicinity of the arrow heads. Looseness and noisy operation even under unloaded conditions will, therefore, be avoided. However, as shown in FIGURE 5, the main or controlling force vector on the rollers is created by the pressure differential between the successive pockets of the pump.

As shown in FIGURE 5, the pocket C is the last intake pocket before the pressure cycle starts with the successive pocket D. Pocket D is thus under greater pressure than pocket C and this pressure differential across the vane roller **50** urges the vane into sealing contact with the tooth **23** at point **60** and with the recess **51** at point **61**. The general direction of the loading force on the roller vane **50** is indicated by the arrow **62**. Leakage from the high pressure pocket D back to the lower pressure pocket C is thus effectively prevented even though the roller vane **50** is loosely fitted in its recess **51** of the stator body **52**.

It will also be evident from FIGURE 5 that the sealing efficiency of the assembly is insured even though wide tolerance variations occur between the roller vanes **50** and the vane body **52** of the stator **12**. On the intake side of the pump, the roller vanes **50** need not be in the tight sealing engagement with their recesses **51**, but, as pointed out in connection with FIGURE 4, the rotor reaction on the vanes on the intake side of the pump is such that the vanes will not rattle or vibrate in the body ring **52** of the stator because the force vectors as shown by the arrows **55** through **59** insure seating of the roller vane at some area of the recess even when the vanes are not subjected to the pressure differential occurring on the pressure side of the pump. Thus, the "loose" tooth segments or roller vanes are automatically positioned in their free-fitting recesses under the forces developed within the pump and the assembly in effect is self-compensating. If wear develops on the rollers or on the teeth of the rotor, this self-compensating action will still maintain the efficiency of the pump and leakage paths will not be opened up. In addition, friction of operation can be somewhat reduced because the rollers **50** can shift and rotate to some degree under the influence of the rotor.

The tightness of the screws **16** controls too, to a certain extent, the degree of looseness of the rollers **50** relative to the stator body **52**.

In the embodiment shown in FIGURE 6, the modified stator **12'** has a ring body **52'**, provided with rectangular recesses **51'** around its inner periphery receiving gear segments or vanes **50'** of generally rectangular configuration, but having rounded lobe defining inner faces **25'**. The segments **50'** can be solid or hollow as desired, and as shown in FIGURE 6, the screws **16'** are relocated to pass through the ring body **52'** between the recesses **51'** thereof.

The vanes **50'** function in the same manner as the roller vanes **50** and are sealed under influence of the pressure differential between pockets C and D at points **60'** and **61'** with the general direction of the sealing force being indicated by the arrow **62'**. The segments or vanes **50'** do not rotate in the recesses or pockets **51'** as do the rollers **50**, but otherwise function in the same manner as the rollers **50**.

From the above descriptions, it will, therefore, be understood that this invention provides an internal gear pump or motor with an internal gear set having vanes, lobes, or tooth segments loosely carried by the enveloping gear of the set to greatly increase permissible tolerances in the set and to provide a self-compensating, self-sealing assembly. The invention materially decreases the manufacturing costs of pumps and motors of the internal gear type especially where one of the gear elements orbits as it rotates.

Although minor modifications might be suggested by those versed in the art, it should be understood that I wish to embody within the scope of the patent granted hereon, all such modifications as reasonably and properly come within the scope of my contribution to the art.

I claim as my invention:

1. In an internal gear type fluid pressure device including a first gear surrounded by and meshed with a second gear, one of said gears having one less tooth than the other gear, said gears being relatively rotatable, one of said gears being mounted for orbital movement relative to the other gear, said second gear having circumferentially spaced fragmental cylindrical recesses around the inner periphery thereof opening through said inner periphery, cylindrical rollers mounted in said recesses and free to rotate therein, said rollers projecting beyond said openings through the inner periphery of the second gear to provide the teeth for said second gear, said first gear and a portion of the second gear between the rollers cooperating with the rollers to form expanding and contracting chambers therebetween, said first gear simultaneously contacting at least two rollers in all positions of the gears to form seals, two of said seals separating the expanding chambers from the contracting chambers, and passages communicating with the chambers for flow of fluid into the expanding chambers and out of the contracting chambers.

2. In an internal gear type fluid pressure device including a first gear surrounded by and meshed with a second gear, one of said gears having one less tooth than the other gear, said gears being relatively rotatable, one of said gears being mounted for orbital movement relative to the other gear, said second gear having circumferentially spaced recesses around the inner periphery thereof separated by fingers therebetween and opening through said inner periphery, individual gear teeth vane units loosely mounted in said recesses and projecting therefrom to define the tooth contact surfaces of the second gear, said first gear and the fingers of the second gear cooperating with the vane units to form expanding and contracting chambers therebetween, said first gear simultaneously contacting at least two of the vane units in all positions of the gears to form seals, two of said seals separating the expanding chambers from the contracting chambers, passages communicating with the chambers between the gears for flow of fluid into and out of the chambers, and said vane units being shiftable under the influence of pressure developed in the chambers to seek sealed engagement with the gears regardless of wide variations in fit between the gears including the vane units defining the chambers.

3. A fluid pressure device of the internal-external orbiting gear type which comprises a housing having a fluid inlet and a fluid outlet, a shaft rotatably supported in the housing, a first gear, a second gear surrounding and meshed with the first gear, one of said gears having one less tooth than the other gear, said gears being relatively rotatable, one of said gears orbiting about the axis of the other gear and coacting therewith to form expanding and contracting chambers between the gears, means coupling said shaft to a rotatable gear, said orbiting gear orbiting about the axis of the other gear at a higher speed than the shaft speed, ports communicating with the chambers between the gears, a valve in the housing controlling flow between the ports and the fluid inlet and outlet of the housing, said second gear having circumferentially spaced recesses therearound opening through the inner periphery thereof, individual gear teeth vane units loosely positioned in said recesses and projecting beyond the openings in the inner periphery of the second gear to define the entire contact surfaces meshing with the first gear, and said units being shiftable under pressure developed in the chambers into sealing engagement with both gears to isolate the chambers, whereby wide tolerance variations may be accommodated between the gears including the vane units.

4. The device of claim **1** wherein the cylindrical rollers are loosely mounted in the recesses and are shiftable under the influence of pressure developed in the chambers into sealed engagement with the gears.

5. The device of claim 1 wherein the cylindrical rollers are hollow.

6. The device of claim 1 including a valve controlling flow through the passages.

7. The device of claim 6 including a shaft coupled to a rotatable gear.

8. The device of claim 7 wherein the valve and shaft are in coupled relation.

9. The device of claim 2 wherein the individual gear teeth vane units are rollers free to rotate in the recesses.

10. The device of claim 2 wherein the individual gear teeth vane units are generally rectangular and have rounded inner surfaces defining the gear teeth.

11. The device of claim 3 wherein the individual gear teeth vane units are hollow, the housing is composed of stacked parts and draw bolts extending through the hollow units mount the housing parts together.

12. The device of claim 3 wherein the valve is coupled to the shaft.

13. The device of claim 3 wherein the valve is a cylindrical spool valve and a wobble stick extends through the valve to couple the shaft with the orbiting gear.

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