June 26, 1962 E. D. HARTLEY 3,040,664

DUAL CAVITY FLUID HANDLING DEVICE

Filed April 13, 1959 5 Sheets-Sheet 1

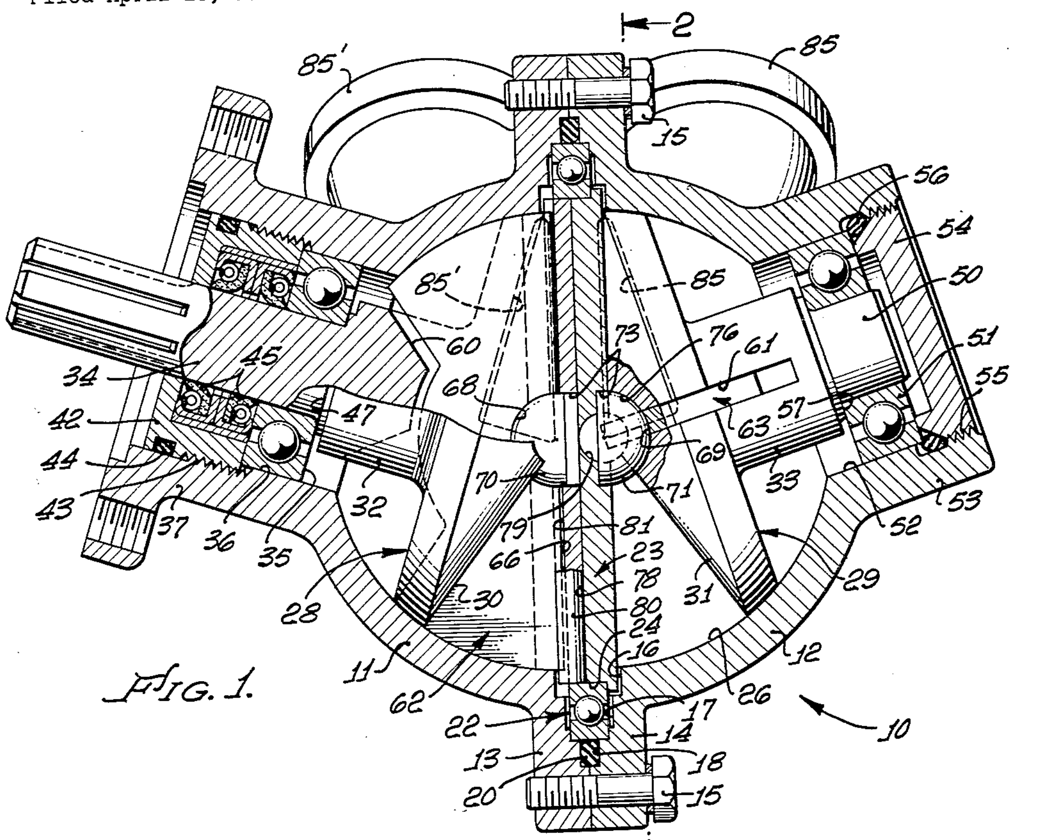

FIG. 1.

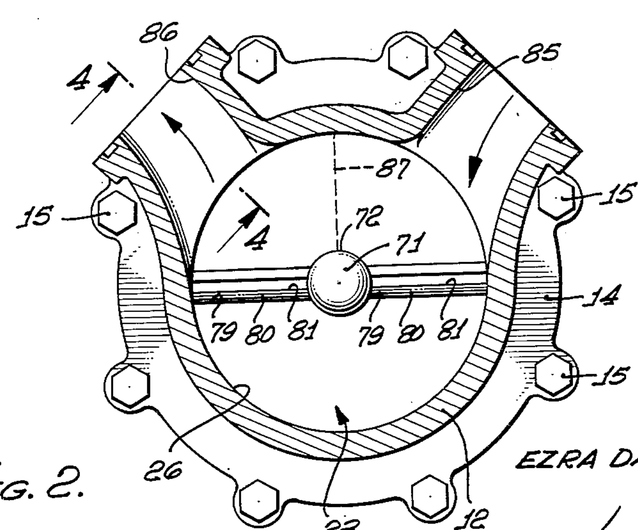

FIG. 2.

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DUAL CAVITY FLUID HANDLING DEVICE

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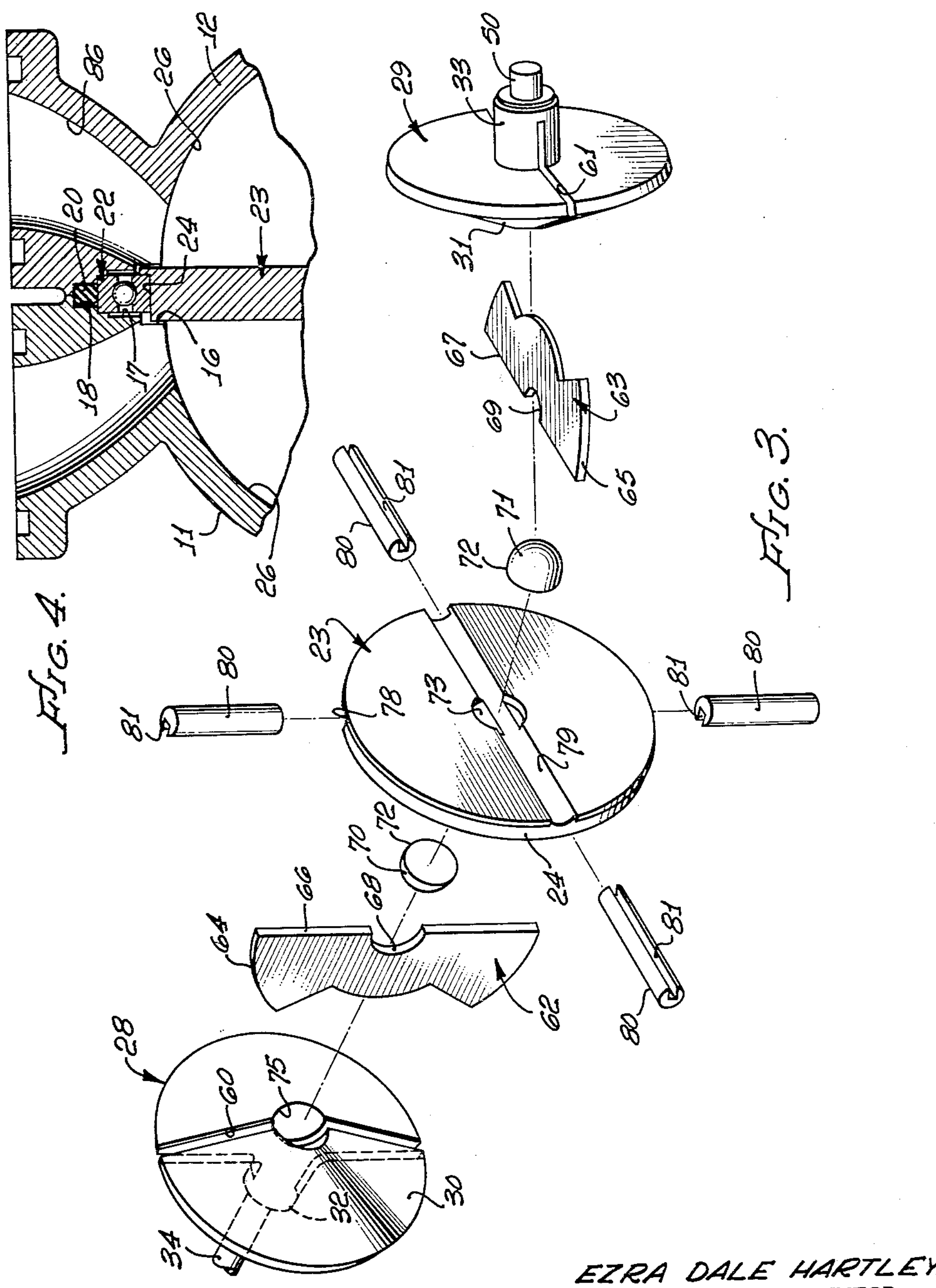

FIG. 4.

FIG. 3.

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June 26, 1962 E. D. HARTLEY 3,040,664

DUAL CAVITY FLUID HANDLING DEVICE

Filed April 13, 1959 5 Sheets-Sheet 3

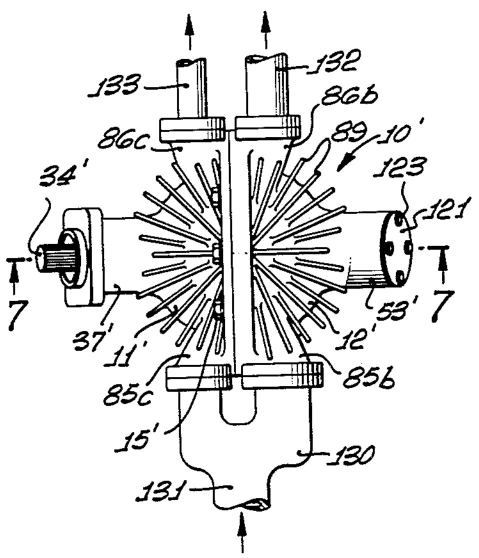

FIG. 5.

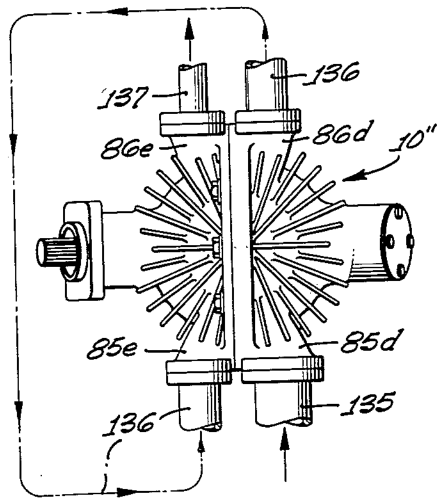

FIG. 6.

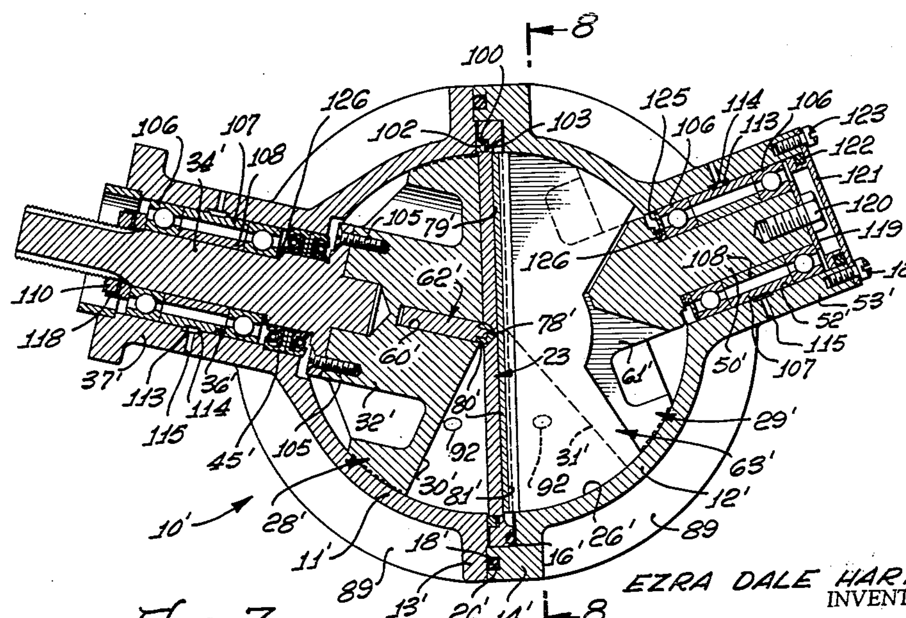

FIG. 7.

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DUAL CAVITY FLUID HANDLING DEVICE

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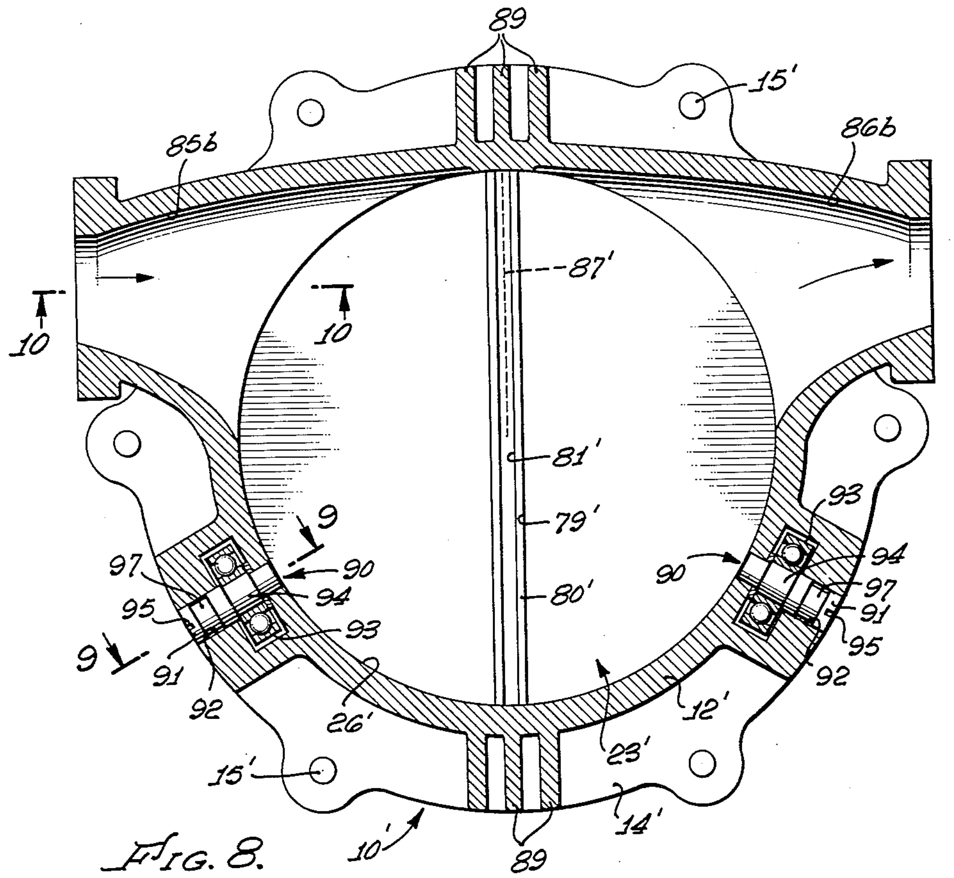

FIG. 8.

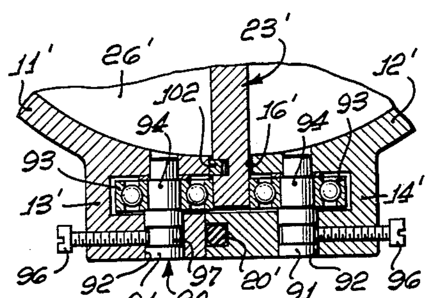

FIG. 9.

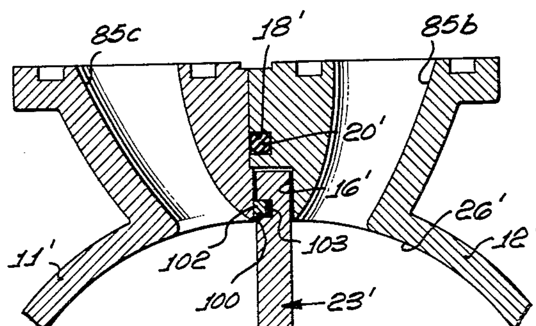

FIG. 10.

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DUAL CAVITY FLUID HANDLING DEVICE

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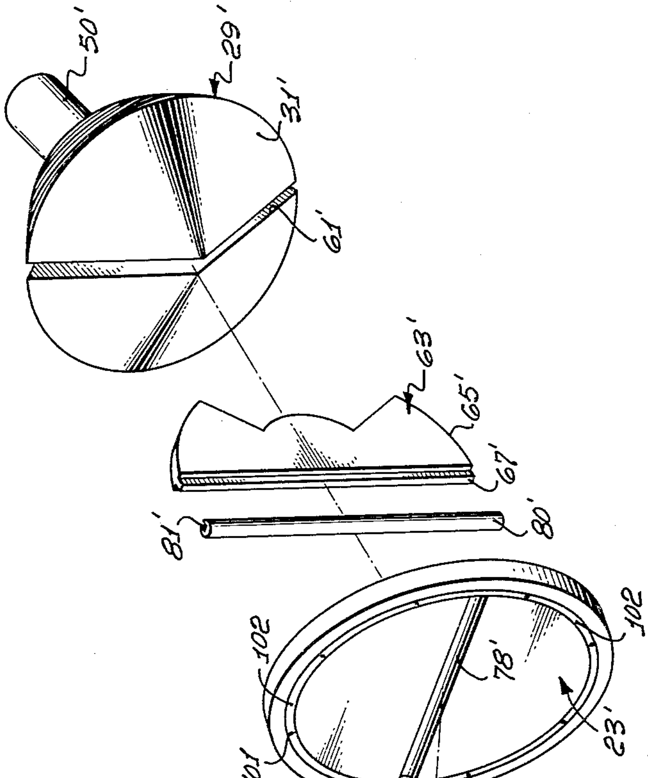
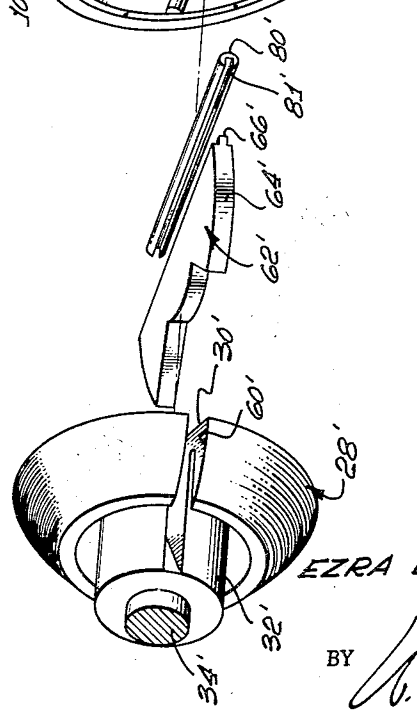

FIG. 11.

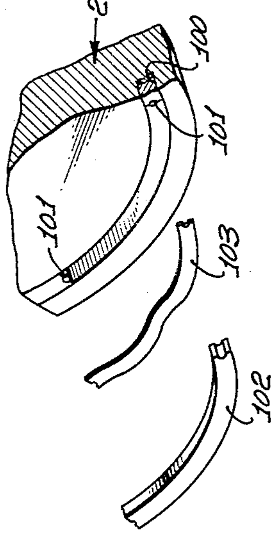

FIG. 12.

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United States Patent Office 3,040,664
Patented June 26, 1962

3,040,664

DUAL CAVITY FLUID HANDLING DEVICE

Ezra Dale Hartley, Los Angeles, Calif., assignor to The Flo-Motive Corporation, Los Angeles, Calif., a corporation of California

Filed Apr. 13, 1959, Ser. No. 806,049
26 Claims. (Cl. 103—4)

This invention relates to fluid handling devices and more particularly to an improved dual cavity, positive displacement fluid handling device having many features of novelty and advantage over prior constructions and exhibiting unusual versatility and adaptability for use in many diverse applications as will be explained in detail below.

A satisfactory fluid handling device suitable for many diverse applications and operating with minimum losses and maximum efficiency and accuracy with either gases or liquids has long been sought by designers. Among the many requirements sought in a satisfactory device of this type are compactness, light weight, a minimum number of moving parts, as well as a device requiring little or infrequent lubrication, exhibiting minimum flow losses, having dynamically balanced parts capable of functioning at substantially the same efficiency over a wide range of temperatures and pressures and equally efficient with fluids of widely different viscosities.

There are certain applications for fluid handling devices in which it is highly advantageous that a given stream of fluid be divided accurately into two streams of predetermined relative sizes. There are others where it is desired to raise the pressure of the fluid stream to a higher pressure than is easily feasible in a single stage. In still other applications, it is desirable to employ a pressurized fluid as the motive power to transfer a separate fluid from one pressure condition to a circuit operating at a higher pressure. Another requirement made upon fluid handling devices is that of providing a steady flow of fluids substantially free of pulsation.

To meet these and many other diverse desirable operating characteristics of fluid handling devices there have been proposed heretofore a profusion of designs, some of which employ rotors and associated vanes bearing a superficial resemblance to the present invention and operating within either a spherical or semi-spherical cavity. However, these designs fall far short of meeting operating requirements satisfactorily. This fact is readily substantiated by the inability of designers to provide fluid handling devices meeting the operating requirements of certain particularly exacting environments. For example, one of the serious limitations imposed on the design of modern aircraft is the inability of fuel pumps and superchargers to meet requirements of military aircraft under the widely varying pressure, temperature and capacity requirements at both low and high altitudes. Pumps quite suitable and adequate at lower elevations are totally unsuitable and fail at higher altitudes, or operate so erratically and uncertainly as to be unacceptable.

By the present invention there is provided a dual cavity fluid handling device obviating the foregoing and many other shortcomings of prior designs. The device is characterized in many, many respects including a substantially non-pulsing output flow, equal effectiveness when driven in either direction, an unusually high positive displacement volume, suitability for operation at widely different speeds, excellent dynamic balancing, unusually low resistance to fluid flow, lack of need for any except the simplest fluid seals, a minimum number of moving parts and the simplicity of the means provided to support thrust loads.

Structurally, the fluid handling device comprises a spherical housing supporting therewithin a rotatable plate dividing the cavity into dual fluid handling cavities. Rotatably seated in each half of the housing is a conically-ended rotor having its axis inclined to and interlocked with the separating plate whereby both rotors rotate in unison with the one being driven by the other through pivoting vanes socketed in the opposite faces of the plate as well as in slots extending diametrically of the rotors. By this arrangement the interlocked vanes and separating plate provide a universal driving coupling interconnecting the two rotors to the end that one rotor is driven from the other with the result that the passage of fluid through the respective fluid chambers occurs in a predetermined volumetric relationship.

Furthermore, according to one structural embodiment of the invention, the coupling between rotors is effective in providing mutual support for the adjacent conical ends of the rotors thereby eliminating the need for supporting bearings in this zone. A further feature of the coupling arrangement is the reduction to a bare minimum of the movement of the parts relative to one another.

Another feature of the design is a supporting bearing arrangement for the outboard ends of the rotors having provision for the adjustment of the bearings axially of the rotor shafts and in a manner to assure operation of the conical surfaces of the rotors in smooth rolling and sealing contact with the intervening dividing plate, as well as to accommodate for wear and proper alignment of the rotating components. According to one preferred construction, means are provided effective to shift either rotor bearing assembly as appropriate and necessary to preload the rotors against the opposite faces of the cavity dividing plate. In another preferred arrangement, matched antifriction bearing assemblies are designed to be accurately and precisely adjusted to preload the conical surfaces of the rotors against the opposite faces of the dividing plate following which a suitable adhesive sets to lock the parts in adjusted position.

Another novel feature of one construction relates to a ball and socket interconnect provided between the tips of the rotors and seating wells centrally of the faces of the dividing plate. Cooperating with this ball and socket connection in supporting the adjacent ends of the rotors is a bearing ring supported between the main housing and the rim of the circular plate separating the two fluid displacement chambers. In this manner, the forces acting on the two rotors are rigidly supported in part by the outboard bearings and in part by the ring bearing for the dividing plate held captive between the conical ends of the rotors.

In an alternate construction, the entire load is substantially carried by outboard bearing assemblies for the two rotors. In this arrangement slotted grooves arranged at right angles to one another in the opposite faces of the dividing plate cooperate with the rotor vanes in holding the adjacent ends of the rotors accurately centered within the spherical cavity without the aid of a bearing ring encircling the rim of the dividing plate.

If the rotors are provided with similarly inclined conical ends, the displacement volumes of the two fluid displacement cavities are identical as is desirable for certain applications of the device. In other applications, it is desired to provide for different displacements in the separate cavities, an objective readily accomplished by providing conical ends of different inclinations on the two rotor ends and mounting these rotors at the different inclinations to the dividing plate as is necessitated by this design of the rotors. When the device is so constructed, the displacement of the two cavities is a function of the inclination of the conical rotor ends, rendering the fluid handling device suitable for various uses as, for example, a two-stage compressor in which the output from the larger cavity is introduced into the inlet of the smaller cavity for further compression therein. Or the inlets of the two cavities may be connected in parallel with a common fluid source in which case the rotors are effective to divide the flow accurately into two streams of predetermined proportions.

Another capability of the invention is that the supply of pressurized fluid to the inlet of one of these dual chambers is effective to drive the associated rotor which, in consequence, is then effective to drive the second rotor as well as to deliver power through the output shaft to some secondary power need external to the device.

Accordingly, it is a primary object of the present invention to provide an improved fluid handling device overcoming the shortcomings of prior designs and exhibiting many advantages thereover.

Another object of the invention is the provision of an improved high efficiency, simply constructed dual cavity fluid handling device, the rotating components of which are dynamically balanced.

Another object of the invention is the provision of a compact lightweight dual cavity pump characterized by the provision of dual conical-ended rotors having their axes inclined at an angle to one another and intersecting centrally of the spherical cavity, with one longitudinal element of each rotor in rolling contact with a rotating plate positioned therebetween.

Another object of the invention is the provision of a dual cavity pump supporting therein similar conical ended rotors each having a pivoting vane mounted in a slot extending diametrically of the rotor with its free outer edge socketed in grooves formed in the juxtaposed surfaces of a rotating plate positioned between the adjacent ends of the rotors.

Another object of the invention is the provision of a dual cavity fluid handling device incorporating a pair of opposed conical-ended rotors held pressed into rolling contact with an intervening separating disc aided by fluid pressure acting on the ends of the rotors remote from the separating disc.

Another object of the invention is the provision of a fluid handling device featuring a novel bearing arrangement adapted to preload the rotating fluid displacement components in preloaded rolling contact with one another.

Another object of the invention is the provision of a multiple purpose fluid handling device having dual positive displacement cavities each having an inlet and an outlet port adapted to be connected selectively in separate fluid circuits, in series with one another or in parallel, as required to suit the operating needs of a particular application.

Another object of the invention is the provision of a dual cavity fluid handling device each cavity of which houses separate rotors interconnected to be driven in unison from a common driving source, the device being selectively operable as a gas compressor, a motor, a pump, and as a flow divider and proportioning device depending upon the manner in which the dual cavities are connected in circuit.

These and other more specific objects will appear upon reading the following specification and claims and upon considering in connection therewith the attached drawings to which they relate.

Referring now to the drawings in which preferred embodiments of the invention are illustrated:

FIGURE 1 is a longitudinal sectional view through one preferred embodiment of the fluid handling device;

FIGURE 2 is a transverse sectional view on a reduced scale taken along line 2—2 on FIGURE 1;

FIGURE 3 is an exploded perspective view of the principal rotating components;

FIGURE 4 is a fragmentary transverse view through the two outlet ports and taken along line 4—4 on FIGURE 2;

FIGURE 5 is a top plan view of a second preferred embodiment of the invention shown connected to function as a flow divider;

FIGURE 6 is similar view to FIGURE 5 showing the device connected to function as a two-stage compressor or as a supercharger for a heat engine;

FIGURE 7 is a longitudinal transverse view on an enlarged scale taken along line 7—7 on FIGURE 5;

FIGURE 8 is a transverse view on an enlarged scale taken along line 8—8 on FIGURE 7;

FIGURE 9 is a fragmentary transverse sectional view taken along line 9—9 on FIGURE 8; and

FIGURE 10 is a transverse sectional view through one pair of fluid ports taken along line 10—10 on FIGURE 8.

FIGURE 11 is an exploded perspective view of the rotating components of the second embodiment; and

FIGURE 12 is a fragmentary exploded view on an enlarged scale and partly in section showing details of the face seal for the separator plate.

Referring now more particularly to FIGURES 1 to 4 showing one preferred embodiment of the invention, there is shown a fluid handling device designated generally **10** and comprising a generally spherical housing formed in two similar halves **11** and **12**. Projecting from the rim edges of these halves are mating annular flanges **13** and **14** adapted to be tightly clamped together, as by cap screws **15**. The abutting faces of flanges **13** and **14** are formed with a series of stepped annular grooves **16**, **17** and **18** of decreasing width having different functions. Innermost grooves **18** seat a suitable sealing element, such as O-ring **20**, whereas the narrow inner portion of groove **17** provides a mounting anchorage for the outer race of ball bearing assembly **22** and utilizes the outer race of assembly **22** to align the two halves **11** and **12** of the housing. A circular disc or plate **23** divides the spherical cavity **26** formed by housing halves **11** and **12** into two equal chambers and has a shouldered peripheral edge **24** having a press fit over the inner race of bearing assembly **22**.

Rotatably supported within the identical semi-spherical cavities of the described housing are a pair of generally similar rotors **28**, **29** each having a general mushroom configuration, including conical facing ends **30**, **31**, and a pair of supporting shafts **32**, **33**, respectively. Shaft **32** has a reduced end **34** journaled in a ball bearing assembly **35** the outer race of which has a snug frictional fit with a bore **36** formed in a boss **37** projecting outwardly from casing half **11**. The outer end of shaft **34** is slotted or otherwise suitably formed for quick driving connection with the complementally shaped coupling of a power source.

Bearing assembly **35** is adjustable along bore **36** through rotation of a bushing **42** having threaded engagement at **43** with the outer end of bore **36**. An O-ring **44** carried by the outer end of bushing **42** forms a fluid seal with side walls of the bore as the bushing is assembled within the bore. Housed within a well of bushing **42** are suitable shaft seals **45** of any suitable construction. The inner end of bushing **42** is adapted to engage the adjacent end of the outer race of bearing **35** and is thereby effective to shift the entire bearing assembly inwardly along bore **36** causing the inner race to engage shouldered portion **47** of rotor shaft **32** to urge conical surface **30** of rotor **28** into smooth and preloaded rolling contact with the adjacent face of plate **23**. It will be understood that the outer end of bushing **42** may be provided with slots or the like for seating a spanner wrench employed to adjust the bushing to preload rotor **28** against plate **23** this operation being carried out concurrently with the similar adjustment of the supporting bearing for rotor **29**.

Rotor **29** is constructed similarly to rotor **28** and the outer end of its shaft **33** is provided with a reduced end portion **50** journaled in a ball bearing assembly **51**, the outer race of which likewise has a snug press fit with a bore **52** formed in a boss **53** of housing half **12**. End cap **54** for the outer end of bore **52** has threaded engagement at **55** with the bore **52** and its inner end bears against the adjacent end wall of the outer race of bearing assembly **51** for the purpose of adjusting the bearing along supporting bore **52**. In this connection it is pointed out that inward adjustment of the outer race ring concurrently with the properly coordinated adjustment of bushing **42** forces the inner race of assembly **51** against shoulder **57** of rotor shaft **33** thereby urging conical surface **31** of this rotor into preloaded rolling engagement with the adjacent face of plate **23**. In consequence, plate **23** is compressed between the conical ends of the rotors to the degree found to provide optimum operation under a particular set of operating conditions.

An O-ring seal **56** is pressed against inner side wall of boss **53** by the chamfered inner end of adjustable cap **54** to seal this bore against leakage. Cap **54**, like bushing **42**, is provided with openings in its outer end wall to receive a wrench useful in adjusting the combined radial and thrust bearing assembly **51** and in preloading the rotors against plate **23**.

Referring more particularly to FIGURE 3, it is pointed out that each of rotors **28** and **29** includes a transverse slot **60**, **61**, respectively, opening through its conical end, the inner end of each slot preferably having the general configuration best illustrated in FIGURE 1. Fitting closely within each slot is a vane **62**, **63** of identical configuration. The opposite ends **64**, **65** of each is spherically contoured to seat snugly against the wall of spherical cavity **26** formed in housing halves **11** and **12**. As will be recognized, these mating surfaces cooperate in maintaining the longer straight edges **66**, **67** of the vanes properly spaced relative to the adjacent surfaces of plate **23**. Straight edges **66**, **67** of the vanes are interrupted centrally by semi-spherical cutouts **68**, **69** having a close sealing fit with the spherical ends **70**, **71** of identical centering and sealing members **72**, the cylindrical adjacent ends of which seat snugly within wells **73** opening centrally through the opposite faces of plate **23**. It is pointed out that spherical surfaces **70**, **71** of members **72** seat in complementally shaped sockets **75**, **76** at the apex of rotors **28**, **29**.

Sealing engagement of vane edges **66**, **67** with the opposite faces of plate **23** is facilitated by semi-cylindrical grooves **78**, **79** extending diametrically along the opposite faces of plate **23** and oriented 90 degrees from one another. Grooves **78**, **79** seat cylindrical sealing members **80** of suitable soft bearing or wear resisting material each having a longitudinal groove **81** sized to have a press fit over the straight edges **66**, **67** of vanes **62**, **63**.

There remains to be described the inlet and outlet ports for each of the fluid displacement cavities provided between the conical ends of rotors **28** and **29** and separated from one another by the intervening plate **23**. These ports are best illustrated in FIGURES 2 and 4 from which it will be seen that the upper portion of each half of the fluid handling device as viewed in the drawings is provided with a pair of identical ports including an inlet port **85** and an outlet port **86**, inlet port **85** opening through substantially in entire periphery of the upper right-hand quadrant of the displacement chamber as viewed in FIGURE 2, and outlet port **86** opening similarly into the upper left-hand quadrant as viewed in the same figure. It is pointed out and emphasized that a longitudinal face element of each of the conical surfaces **30**, **31** is positioned to lie in sealing contact with juxtaposed surfaces of separating plate **23**, the line of sealing engagement of conical surface **31** being represented in FIGURE 2 by the dotted vertical line **87**. This line of sealing contact separates the inlet quadrant of the displacement chamber from the outlet quadrant at all times and in all rotary positions of rotor **29**, it being understood that the same is equally true of rotor **28** and the adjacent face of plate **23**.

The inner end of each port **85**, **86** opens into the displacement chamber, the configuration of these port entrances at their merger with cavity **26** being represented in FIGURE 1 by the triangular shaped dotted line **85**. It will be noted inlet port extends over substantially the entire curved surface of the inlet quadrant and that the outlet port **86** likewise extends over the same portion of the outlet quadrant of the displacement chamber. Likewise, it is pointed out that the displacement chambers disposed on the opposite sides of separating plate **23** are of identical construction, the inlet for the left-hand cavity shown in FIGURE 1 being designated **85'**.

As will be recognized from the foregoing description of the structure, it is necessary to drive only one of the rotors owing to the universal coupling interconnecting the adjacent ends of the two rotors and formed by vanes **62**, **63** in cooperation with plate **23**. For example, assuming that power is supplied to the outer end of shaft **34** for rotor **28**, rotation of this rotor and of vane **62** mounted therein serves, through its connection provided by elements **80** seated in transverse groove **78** of plate **23**, to rotate plate **23** in the plane of its bearing assembly **22**. Inasmuch as elements **80** on edge **67** of vane **63** carried by the other rotor **29** are seated in a similar groove **79** of plate **23**, it follows that rotation of plate **23** is effective to rotate rotor **29** in unison with the first rotor. Preferably, the vanes **62**, **63** are oriented to lie at right angles to one another circumferentially of the rotors, this arrangement contributing materially to the smooth action of the device, its dynamic balancing and the smoothness of the load imparted on the driving shaft. A larger number of vanes may be employed but a single vane in each rotor has been found particularly effective and efficient.

It will therefore be clear that all parts in spherical cavity **26** rotate in unison and that the opposite radial edges of vanes **62**, **63** are rigidly supported, either by associated rotor or by the common connecting plate **23**. Accordingly, the only surfaces of the two displacement cavities not in motion with the fluid being displaced are the spherical portions of the displacement chambers themselves. These surfaces are relatively small in area and normally about one-fourth of the total surface area of the displacement cavities. For this reason, surface friction losses of the fluid being moved are reduced to a minimum. The described fluid handling device is particularly efficient not only with low viscosity fluids but also when handling high viscosity fluids.

Referring now to FIGURE 5 and FIGURES 7 to 12, there is shown a second preferred embodiment of the fluid handling device designated generally **10'**, the same or similar parts to those of the first embodiment being indicated by the same reference characters distinguished by the addition of a prime. The second embodiment differs in no major respect from the first. These differences include the inclination of the two rotors **28'**, **29'** to plate **23'** at somewhat different angles; the substitution of adjustable stabilizing bearings for the annular bearing embracing the rim of plate **23'**; and the use of matched bearings and a novel locking technique to support the outboard ends of the two rotor assemblies. Although these differences are rather minor from a structural standpoint, it is pointed out that the versatility and flexibility of the resulting structure is increased to a marked degree. It is also possible to manufacture this construction to very rigid manufacturing tolerances at reasonable cost.

Referring more particularly to the mentioned structural differences best illustrated in FIGURES 7 to 11, it will be understood that the exterior of the two housing halves **11'**, **12'** are provided with a plurality of integral heat radiating fins **89** distributed over the entire exterior surface of the housing as an aid in dissipating heat generated in considerable quantity when the device is used as a gas compressor. If desired, the housing may be liquid-cooled, as where the supply of cooling air is restricted.

Mating flanges **13'**, **14'** of housing halves **11'**, **12'** are held assembled by cap screws **15'** and an annular groove **18'** formed in the face of flange **14'** contains an O-ring seal **20'** preventing leakage from the spherical cavity **26'** in the assembled position of the parts. An inwardly opening annular groove **16'** formed between flanges **13'**, **14'** embraces the peripheral rim portions of dividing plate **23'** in the manner made clear by FIGURE 7. The bottom and side walls of groove **16'** have small but ample running clearance with the juxtaposed surfaces of plate **23'** and permit limited adjustment of plate **23'** without actually contacting the groove walls. In lieu of a supporting bearing corresponding to assembly **22** of the first described embodiment, plate **23'** is stabilized and constrained to rotate in a desired plane without contact with the walls of groove **16'** by two pairs of roller bearing assemblies **90**, **90**, the bearings of each pair preferably being disposed to bear against the opposite rim faces of plate **23'** with their axes located along radial lines parallel to the faces of plate **23'**.

Bearing assemblies **90** include a pair of shafts **91** fitting within bores **92** of the casing halves. Roller bearing assemblies **93** are mounted on an eccentric portion **94** preferably integral with each shaft **91**, the eccentric being so rotated that the peripheral rim of the outer race of each bearing **93** bears against the rim face of plate **23'**. Owing to the mounting of the bearings **93** on eccentrics **94**, rotation of the shafts **91** by by the aid of a screwdriver inserted in kerf **95** (FIGURE 8) at the outer end of each shaft, serves to shift the bearings **93** toward or away from the adjacent rim face of plate **23'**. Accordingly, it will be recognized that plate **23'** can be accurately adjusted from the exterior of the housing to operate in a desired plane within narrow sealing groove **16'**. Once adjusted, plate **23'** is locked in position by the tightening of locking screws **96** against the annularly grooved portions **97** of shafts **91**. Although not so shown, it will be understood that the outer ends of shafts **91** are provided with suitable seals such as O-rings to prevent leakage of fluid past the ends of shafts **91**.

As is best shown in FIGURES 7 and 11, the opposite faces of plate **23'** are provided with semi-circular grooves **78'**, **79'** arranged at right angles to each other with a length slightly greater than the diameter of spherical cavity **26'**. It is desirable that grooves **78'**, **79'** terminate short of the periphery of plate **23'** so as not to traverse the rolling surfaces traversed by the peripheries of the stabilizing bearings **93**. Seating with a close rolling fit within the semi-cylindrical grooves **78'**, **79'** are the sealing and bearing elements **80'**. Grooves **81'** extending the full length of these elements have a close friction fit with shouldered edges **66'**, **67'** of vanes **62'**, **63'**.

As will be recognized from the foregoing description of the coupling provided by the described connection of vanes **62'**, **63'** with grooves **78'**, **79'** in plate **23'**, this plate is thereby held firmly and rigidly against radial movement within its own plane and is constrained to rotate about its own center as well as the center of spherical cavity **26'**.

Additionally, plate **23'** is stabilized and confined strictly to the aforementioned plane by reason of the stabilizing support provided by the two pairs of bearing assemblies **90**, **90** arranged as they are 120 degrees circumferentially to either side of the line of contact of the conical surfaces **30'**, **31'** of the rotors with adjacent faces of plate **23'**. This latter point will be better understood by reference to FIGURE 8 wherein it is noted that the stabilizing bearing assemblies **90**, **90** are disposed in the four and eight o'clock positions about the rim of plate **23'**, or 120 degrees to either side of the twelve o'clock position. Since the longitudinal face elements of the conical ends of the rotors are in sealing contact with the face plate, as is indicated by the dotted line **87'** in FIGURE 8, the two rotors are effective to stabilize the upper portion of the plate.

To prevent leakage from one displacement cavity to the other past the rim of plate **23'**, the rim portions of the plate are preferably provided with suitable sealing means. One particularly effective form of seal is illustrated in FIGURE 12 and includes an annular groove **100** in one face surface of the plate inside the trackway for stabilizing bearings **93**. Disposed in an upright position along the bottom of groove **100** are spacing pins **101** between which are located arcuate sections **102** of a suitable face seal formed of a suitable material such as graphite, sintered bronze, or the like. These sealing elements are notched at their ends to fit about pins **101** and are urged outwardly into sealing engagement with the side wall of grove **16'**, as by undulating leaf springs **103** disposed between the bottom of groove **100** and the underface of elements **102**.

The outboard bearing assemblies supporting shafts **34'**, **50'** of rotors **28'**, **29'**, respectively, will now be described, particular reference being had to FIGURE 7. It will be noted that shaft **34'** is formed separated from rotor **28'** and is secured to hub **32'** thereof, as by cap screws **105**. Shaft **34'** extends centrally through bore **36'** of bearing support boss **37'**. Accurately matched ball bearing assemblies **106** are held spaced apart along bore **36'** by precision matched spacer sleeves **107**, **108**, it being noted that outer sleeve **107** has close sliding fit with bore **36'** and that inner sleeve **108** has a close friction fit on shaft **34'**. These sleeves and bearing assemblies are carefully matched prior to assembly so that, when assembled to bore **36'** and shaft **34'**, the balls of each assembly as well as the line support between rotor **28'** and plate **23'** will be preloaded to the desired load values for maximum operating efficiency and service life. The described outboard bearing assembly for the shaft of rotor **28'** will be understood as duplicated as respects shaft **50'** of rotor **29'**. Accordingly, it is deemed unnecessary to repeat the description of this second bearing assembly. It should be noted that fluid is prevented from entering the bearing assembly as by any suitable fluid seal **125** mounted across the inner end of the innermost bearing unit **106**. Interposed between this sealing ring and the adjacent shouldered portion of shaft **50'** is a shim **126**. It will be understood that a similar shim is located between the shouldered inner end of shaft **34'** and the adjacent inner raceway of the associated bearing unit **106**. These shims may be replaced with other shims of appropriate thickness should it ever become desirable at a later date to adjust the relationship of the rotors relative to divider plate **23'** to accommodate wear or for other reasons.

The two outboard bearing assemblies may be held rigidly in their desired and proper preloaded operating positions relative to cavity **26'** and plate **23'** by means and assembly technique now to be described. The exterior midportion of sleeve **108** and the juxtaposed area of bores **36'** and **52'** are provided with annular grooves **113** and **114**. A plurality of passages **115** open into grooves **113**, **114** to permit charging of these grooves with a suitable adhesive or bonding agent effective to lock the bearing assembly in a desired adjusted operating position when the adhesive takes a set. A preferred adhesive is of the epoxy resin type prepared to take a set after a period of time sufficient to permit fine adjustment of the various components. A setting time of two to three hours is found quite adequate.

When making the adjustment, the parts are first assembled approximately into their desired operating positions, care being taken not to press the bearing assemblies fully into position along bores **36'** and **52'**. After checking to determine that the parts are approaching their approximate operating positions relative to the opposite sides of plate **23'** and the spherical surfaces of cavity **26'**, the epoxy adhesive is introduced through ports **115** to fill grooves **113**, **114**. Desirably, the parts are then first rotated by hand followed by power drive under pressure. Suitable means are then utilized against the outer ends of the bearing assemblies to shift them inwardly along bores **36′**, **52′** to the slight extent necessary to assure preloading of the balls and the rolling contact of rotors **28′**, **29′** with plate **23′**. An experienced assemblyman can determine when the parts are properly adjusted for operation at maximum efficiency. Once this accurately determined adjustment has been determined, the device is allowed to rest for the brief time required for the epoxy or other adhesive to take a set rigidly locking the components assembled. It will be understood that an important preliminary phase of this assembly is the simultaneous and prior adjustment of stabilizing bearing assemblies **90**, **90**. Once all adjustments have been made, lock nut **110** is firmly tightened and the flanges of lock washer **118** having tabs interfitting with splines of shaft **34′** are struck upwardly to lock nut **110** in place to prevent all axial play of shaft **34′** relative to the rigidly anchored bearing units **106**.

The lock provided for shaft **50′** of rotor **29′** includes disc **119** overlying the inner race of the outer bearing unit **106**, this disc being held thereagainst by a cap screw **120** threaded into a bore in the end of shaft **50′**. A closure cap **121** having an O-ring seal **122** seating against bore **52′** is seated across the outer end of bore **52′** and held in place, as by cap screws **123**.

The porting arrangement for the fluid displacement chambers is generally similar to that described in connection with the first embodiment but differs in certain respects as will be best understood by reference to FIGURES 8 and 10. As is there shown, inlet port **85*b*** opens into the upper right-hand quadrant of the displacement chamber whereas outlet port **86*b*** communicates similarly with the entire spherical surface of the upper right-hand quadrant of the displacement chamber. Flow between the inlet and outlet quadrants of the displacement chamber is prevented by the seal provided between face **31** of the rotor and plate **23′**, the line of sealing contact being represented by dotted line **87′** in FIGURE 8. In the particular position of the rotor there shown, groove **79′** seating sealing element **80′** of vane **63′** is positioned vertically and, accordingly, directly overlies seal line **87′**. It will be observed also that in this position of the rotor, the left-hand half of the displacement cavity or chamber is open to inlet port **85*b*** whereas the remaining half of the chamber to the right of the vane opens to outlet port **86*b***.

Owing to the fact that rotor **29′** is inclined to plate **23′** at a greater angle than rotor **28′**, the displacement capacity of the associated displacement cavity is greater than that of the other chamber. In this connection it is pointed out that the volumes of the two displacement chambers may vary widely or be equal depending upon the particular application and use being made of the fluid handling device and the relative angles of the conical surfaces of the rotors. The disproportionate displacement volumes of the two chambers can be availed of to divide or proportion one stream into streams of unequal but accurately related volumes. Another important application to which the unequal displacement chamber construction is adapted is as a two-stage compressor or as a supercharger for supplying combustion air to an engine. Still another application is the use of one of the chambers to receive pressurized fluid to drive the rotor housed therewithin to provide a source of rotary power, the output being usable either to drive the other rotor for fluid transfer purposes or to perform this function concurrently with the supply of power to drive an auxiliary device exterior to the fluid handling device.

When employed as a proportioning or dividing device, the two displacement chambers are connected as illustrated in FIGURE 5. The stream to be divided is conducted into the inlet ports **85*b***, **85*c***, through a Y-coupling **130** having its stem portion **131** connected to the fluid source and the ends of its branches connected respectively to inlet passages **85*b***, **85*c***, leading into the two displacement chambers. Outlet ports **86*b***, **86*c*** are connected to the separate outlet conduits **132** and **133**, respectively, conduit **133** being smaller in cross-section to accommodate the smaller volume discharging from the smaller fluid displacement chamber. Inasmuch as the two rotors are positively coupled to operate at the same speed it follows that the output volume from the chambers associated with each rotor are proportional to the volumetric capacities of the displacement chambers.

In FIGURE 6, there is shown a second mode of connecting the flow ports of fluid handling device **10″** which device is identical with that illustrated in FIGURE 5. In this arrangement the device functions as a two-stage compressor and is suitable for use at the very high speeds required in supercharger applications. Low pressure gas to be compressed is delivered through conduit **135** into the inlet port **85*d*** opening into the larger displacement chamber. The compressed gas discharging from port **86*d*** is conducted by conduit **136** and inlet port **85*e*** into the smaller displacement chamber. The partially compressed gas delivered to this latter chamber is there further compresed and discharged through outlet port **86*e*** into outlet pipe **137** leading to a place of compressed gas utilization.

The operation of the several embodiments of the disclosed fluid handling device will be quite apparent from the foregoing detailed description of the components and their operating relationships one to the other. As has been pointed out, either liquids or gases, or a combination of both, may be introduced into the positive displacement chambers associated with each rotor. If pressurized fluid is introduced into one of the chambers, the associated rotor is driven and becomes a source of power for the purposes referred to above. The device is therefore seen to be equally suitable for use as either a gas or liquid pump, as a gas compressor of either the single or double stage type, and as a flow proportioning device, to mention the more obvious applications. Fluid leaking past the peripheral rims of the rotors into the chambers rearwardly of the rotors is of negligible volume. Such leakage merely serves to fill the cavities behind the rotors to a pressure related to the outlet pressure of the associated displacement chamber, this pressure being effective on the rear of the rotors to reduce the thrust load on the bearings. In certain applications it may be desirable to further reduce the thrust load on the bearings and this is conveniently accomplished by providing a definite bleed passage around the rim of the rotor with its inlet end in communication with a high pressure zone of the displacement chamber as, for example, the outlet port of this chamber. These passages are not illustrated but are easily provided for, either by the provision of a small conduit or by a duct formed in the main housing itself.

Used as a liquid pump, the disclosed device is self-priming and equally effective to pump liquid in either direction depending upon the direction in which the rotors are driven. It is a simple matter to change the pumping rate between wide limits with accuracy and speed owing to the fact that the volume pumped is dependent upon the driving speed. The one-piece construction of the vane associated with each rotor, taken with the arrangement of the blades in the pumping chamber, results in the vane moving only through a slight distance about its own center of gravity. Of importance too is the fact that the loaded vane is supported along its opposite longer edges, the only unsupported edge of the rotor being the spherical edge in sealing contact with the spherical surface of cavity **26** and **26′**. Of importance also is the fact that the one-piece vanes are dynamically balanced with the result that there are no unbalanced centrifugal forces tending to produce wear on the opposite ends of the vanes or on the spherical cavity wall.

By preloading the rotors against the plate separating the two fluid displacement cavities, assurance is provided of a fluid seal between a longitudinal face element of each conical end of the rotor and the juxtaposed surface area of the separating plate and that the parts will be held firmly in their proper and most effective operating positions with minimum losses. In the first embodiment the thrust loading on the plate is subject to adjustment through the outboard bearings, while in the second embodiment, adjustment is made by the substitution of appropriate shims. Reduction in the thrust load on the shaft bearings is accomplished by increasing the fluid pressure on the rear surfaces of the rotors as by connecting the cavity back of the rotors through appropriate passages in communication with the fluid discharge ports of the disclosed fluid handling devices.

While the particular dual cavity fluid handling device herein shown and disclosed in detail is fully capable of attaining the objects and providing the advantages hereinbefore stated, it is to be understood that it is merely illustrative of the presently preferred embodiments of the invention and that no limitations are intended to the details of construction or design herein shown other than as defined in the appended claims.

I claim:

1. A positive displacement fluid handling device comprising a casing having a generally spherical cavity, rotatable imperforate plate means dividing said cavity into a pair of generally similar semispherical chambers, separate rotors in each of said chambers each having oppositely facing spherical surfaces forming a seal with the cavity surface and a conical end contacting said plate means in line contact, means journalling said rotors in said casing for rotation about axes inclined to said plate means at an angle greater than 45 degrees but less than 90 degrees, each of said rotors having a diametric slot across the conical end thereof pivotally seating in each thereof a unitary vane confined to one of said semispherical cavities, said vanes each having its outer edge interfitting with seating means therefor on the adjacent face of said plate means and having oscillating movement relative to its support rotor, said vanes and plate means providing a driving coupling interconnecting said rotors and cooperating to keep the rotation of one rotor synchronized with the rotation of the other rotor, means for directly driving one rotor from a prime mover, and fluid supply and discharge port means in said casing communicating with each of said chambers and with adjacent quadrants thereof at points close to but on the opposite sides of said line of contact between said plate means and the adjacent one of said rotors.

2. A fluid handling device as defined in claim 1 characterized in that the conical surface of one of said rotors is inclined to its axis of rotation at a different angle than is the surface of the other rotor whereby the fluid displacement volumes of said similar chambers differ from one another.

3. A fluid handling device as defined in claim 1 characterized in that the conical surfaces of said rotors are similarly inclined to their respective axes of rotation whereby the fluid displacement volumes of said similar chambers are substantially identical.

4. A fluid handling device as defined in claim 2 characterized in means for conducting fluid discharging from the chamber of greater displacement volume to the inlet of the other of said chambers.

5. A fluid handling device as defined in claim 2 characterized in means for connecting the inlets of both chambers to a common fluid supply for passage through said separate chambers whereby the volume of fluid discharging from said two chambers is proportional to their respective displacement volumes.

6. A fluid handling device as defined in claim 1 characterized in that one of said chambers and the rotor mounted therein are effective as a motor when the inlet thereof is connected to a source of pressurized fluid and its outlet is vented to a point of fluid disposal, the rotation of the rotor driven by the pressurized fluid being effective through said interconnected vanes and plate means to drive the other of said rotors to transfer fluid through the associated one of said fluid chambers at a rate determined by the speed of rotation of the first mentioned rotor by the pressurized fluid.

7. A fluid handling device as defined in claim 1 characterized in that said plate means is circular and of appreciably greater diameter than said spherical cavity formed within said casing, said casing having an annular recess opening into said cavity and embracing the peripheral rim of said plate means, and means housed within said recess and cooperating with the rim portion of said plate means in supporting the same for rotation in unison with said rotors to either side thereof.

8. A fluid handling device as defined in claim 7 characterized in that said casing is formed in two major parts separable along a plane close to and parallel to said plate means.

9. A fluid handling device as defined in claim 7 characterized in the provision of means forming a dynamic seal between a wall of said annular recess and a rim surface of said plate means and cooperating to prevent any substantial flow of fluid past the rim of said plate means and between the chambers to either side thereof.

10. In a fluid handling device of the type having a casing formed with a spherical cavity and having separate pairs of inlet and outlet ports opening into said cavity on the opposite sides of a diametric plane therethrough, means rotatably supporting an imperforate dividing plate at said diametric plane and effective to divide said cavity into a pair of similar semispherical fluid chambers, similar and separate rotors mounted one in each of said chambers and journalled for rotation therein about axes lying in a plane normal to said dividing plate and inclined to one another at an included angle of less than 180 degrees but greater than 135 degrees, said rotors each having a spherical surface in running sealing contact with a spherical surface of the associated fluid chamber and a conical surface having line contact with said dividing plate in a plane substantially coincident with the above-mentioned plane of said rotor axes; that improvement which comprises single vane means oscillatably supported in a slot across the adjacent end face of each of said rotors, and means for interlocking the adjacent edges of said vanes with the juxtaposed face areas of said dividing plate along diametric areas of said dividing plate lying substantially at right angles to one another and cooperable therewith to maintain the rotation of said rotors synchronized with one another and the vanes of said rotors in a predetermined circumferentially spaced relationship.

11. A fluid handling device as defined in claim 10 characterized in that said means for interlocking the adjacent edges of said vanes and said dividing plate includes a pair of spherically surfaced centering members having their adjacent ends seated in receiving wells formed in the opposite faces of the dividing plate and centrally thereof, the spherical portions of said centering members being socketed in the complementally formed juxtaposed portions of said vanes and rotors, said centering members cooperating with said dividing plate and the support for the latter to support the inner adjacent ends of said rotors, and journal means rotatably supporting the remotely disposed end portions of said rotors.

12. A fluid handling device as defined in claim 10 characterized in that the journals supporting said rotors are positioned on the opposite sides of said spherical cavity and remote from said dividing plate, one of said journals having a shaft connected with the adjacent ones of said rotors with its other end accessible from the exterior of said casing for connection to a prime mover, and adjustable means associated with each of said journals effective to accommodate the shifting of either rotor toward or away from the dividing plate as desirable for optimum operating conditions of the rotating components of said fluid handling device.

13. A fluid handling device as defined in claim 10 char-

acterized in that approximately seventy percent of the internal surface areas of said device in contact with fluid in transit therethrough are moving in unison with the fluid thereby greatly minimizing frictional flow losses.

14. In a fluid handling device of the type having a spherical housing for a pair of independent conical-ended rotors rotatably supported within said housing with their conical ends bearing lightly against the opposite faces of a separating plate rotatably mounted within said casing, antifriction bearing means supporting the outer remote ends of said rotors and the peripheral rim portion of said separating plate; that improvement which comprises means operable to shift the antifriction bearing means for said rotors to vary the operating and fluid sealing relationship of the rotors with respect to the opposite faces of said separating plate.

15. A fluid handling device as defined in claim 10 characterized in the provision of means for shifting the position of the antifriction bearing support for the rim of said separating plate so as to support said separating plate for rotation in a desired plane and to accommodate adjustment of the bearings for said rotors.

16. A fluid handling device as defined in claim 15 characterized in that said antifriction bearings for the rim of said separating plate comprise a plurality of nested antifriction rings mounted on supporting shafts, each pair of shafts being disposed on the opposite sides of the separating plate rib, and each of said shafts including an eccentric section supporting said nested rings, whereby shifting of said nested rings laterally of said shaft is accomplished by a partial rotation of the shaft as required to effect the desired shift of the bearing rings.

17. A fluid handling device adapted for use in the positive displacement of a gaseous fluid, said device comprising a spherical housing formed in two major parts adapted to be clamped together at a diametric dividing junction, said housing having a spherical cavity, a pair of conical-ended rotors each having a supporting shaft journalled in said housing with the conical ends of said rotors facing one another, a circular separating plate rotatably supported between said rotors and having line contact with the conical surface of each rotor, said rotors having axes of rotation inclined at different angles relative to the plane of said separating plate and said plate dividing said cavity into a pair of semispherical fluid chambers, each rotor having at least one vane oscillatably supported within a diametric slot across the conical end thereof and cooperating with said housing and said separating plate to provide a positive fluid displacement chamber, said casing being provided with separate pairs of inlet and outlet ports opening into the respective fluid chambers on opposite sides of the line contact between said rotors and said dividing plate, means extending between said vanes and said plate for causing the latter to rotate in unison with said rotors, means to drive one of said rotors from the exterior of said housing, and means interconnecting the outlet port of the fluid displacement chamber on one side of said plate with the inlet port to the fluid displacement chamber on the other side of said separating plate.

18. A fluid handling device as defined in claim 17 characterized in that each of said vanes has a cylindrical like surface extending lengthwise of its longer edge which surface is adapted to have a close running fit in a complementally shaped groove extending diametrically across the surface of said circular separating plate, the vane seating grooves on the opposite sides of said plate being arranged with their axes at an angle to one another relative to the center of said plate.

19. A fluid handling device as defined in claim 18 characterized in that said vane seating grooves have a length substantially co-extensive with the length of the mating edge of said vanes and short of the full diameter of said circular separating plate.

20. A fluid handling device as defined in claim 19 characterized in that one rim face of said separating plate has an annular groove therein seating sealing means cooperable with a juxtaposed surface of said housing to prevent leakage of fluid around the rim of said separating plate.

21. In a fluid handling device of the type having a spherical housing enclosing a pair of rotors each having a shaft on the remote ends thereof journalled in said housing on axes which intersect one another adjacent the center of the housing and at an included angle in excess of 135 but less than 180 degrees, the adjacent ends of said rotors having conical ends one side of each of which has rolling contact with a circular imperforate plate positioned between said rotors and dividing said housing into two semispherical fluid chambers, said plate having a diametric groove in each face thereof arranged at right angles to one another and closed at their ends, the adjacent ends of said rotors being diametrically slotted and seating oscillatable vane means therein, the adjacent facing edges of said vane means being seated in the adjacent groove in said plate, said vane means and said plate cooperating to interconnect said rotors in a manner to transmit rotary movement of the one to the other, and said vane means and said plate mutually assisting one another in supporting the described moving components for rotation about the center of said housing, and said housing having separate pairs of inlet and outlet ports for each fluid chamber with the ports of each pair being disposed on the opposite sides of the line of rolling contact between the associated rotor and said circular plate.

22. A fluid handling device as defined in claim 21 characterized in that the said housing is provided with an inwardly opening annular groove embracing the rim portion of said circular plate, and means for holding said plate captive between said rotors and the vane means thereof with the rim of said plate out of contact with said housing and with said annular groove.

23. A fluid handling device as defined in claim 22 characterized in the provision of rollers journalled in the opposite side walls of said annular groove with their peripheries of an associated pair thereof bearing on the opposite faces of the plate rim and cooperating with said vane means in supporting said plate in a desired plane.

24. In a fluid handling device of the type having a casing rotatably seating a conical-ended rotor in a spherical cavity therewithin, said rotor being inclined so that the conical surface thereof has line contact with a flat plate located in substantially a diametric plane of said cavity, means supporting said plate for rotation with said rotor and with one surface thereof in rolling contact with the conical rotor surface; that improvement which comprises a bearing assembly supporting the conical surface of said rotor in rolling contact with said plate under a predetermined preload pressure, said bearing assembly including antifriction bearing means having a close friction fit with a bore provided therefor in said casing, said bearing means being effective when pressed axially along said bore to urge the conical rotor surface into pressurized rolling contact with said plate, and means for permanently holding the components so positioned comprising sleeve means in said bore and bearing against said bearing, said sleeve means and said bore being formed to receive fluid adhesive capable upon setting to anchor the parts firmly in assembled position with said rotor preloaded in rolling contact with said plate.

25. That improvement defined in claim 24 characterized in that said bearing assembly includes a pair of matched antifriction bearing means spaced axially of one another along said bore, and further characterized in that said sleeve means includes a pair of concentric sleeves for holding said pair of bearing means spaced apart and cooperating therewith as pressure is applied to the bearing assembly to preload said rotor and simultaneously preload components of said bearing means,

the outer of said sleeves and the juxtaposed surface of said bore having provision for receiving a charge of adhesive therebetween and effective upon taking a set to anchor said bearing assembly and said rotor under a desired preload.

26. In a fluid handling device of the type having a spherical cavity having rotatably supported diametrically thereof a dividing plate, said plate separating a pair of rotors having a running seal with the cavity wall and with the conical ends thereof in rolling line contact with the opposite faces of said plate, the opposite ends of said rotors being cut away to provide closed fluid cavities at the remote ends of said rotors, vane means movably supported in the conical ends of said rotors and cooperating with said plate to provide a pair of fluid displacement chambers, inlet and outlet port means in communication with each of said displacement chambers with an associated pair of ports disposed on the opposite sides of said rolling line contact between said rotor and said dividing plate, thrust bearing means supporting said rotors and arranged to hold said rotors pressed in rolling contact with said plate, and means for reducing the thrust load on said bearings comprising means for maintaining fluid under pressure in the portions of said cavity behind the remote ends of said rotors and effective to urge said rotors toward said plate.

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UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,040,664 June 26, 1962

Ezra Dale Hartley

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 13, line 17, for the claim reference numeral "10" read -- 14 --.

Signed and sealed this 26th day of March 1963.

(SEAL)
Attest:
ESTON G. JOHNSON

Attesting Officer

DAVID L. LADD
Commissioner of Patents