Sept. 24, 1968 A. C. MAMO 3,402,669

CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 1

FIG. 1

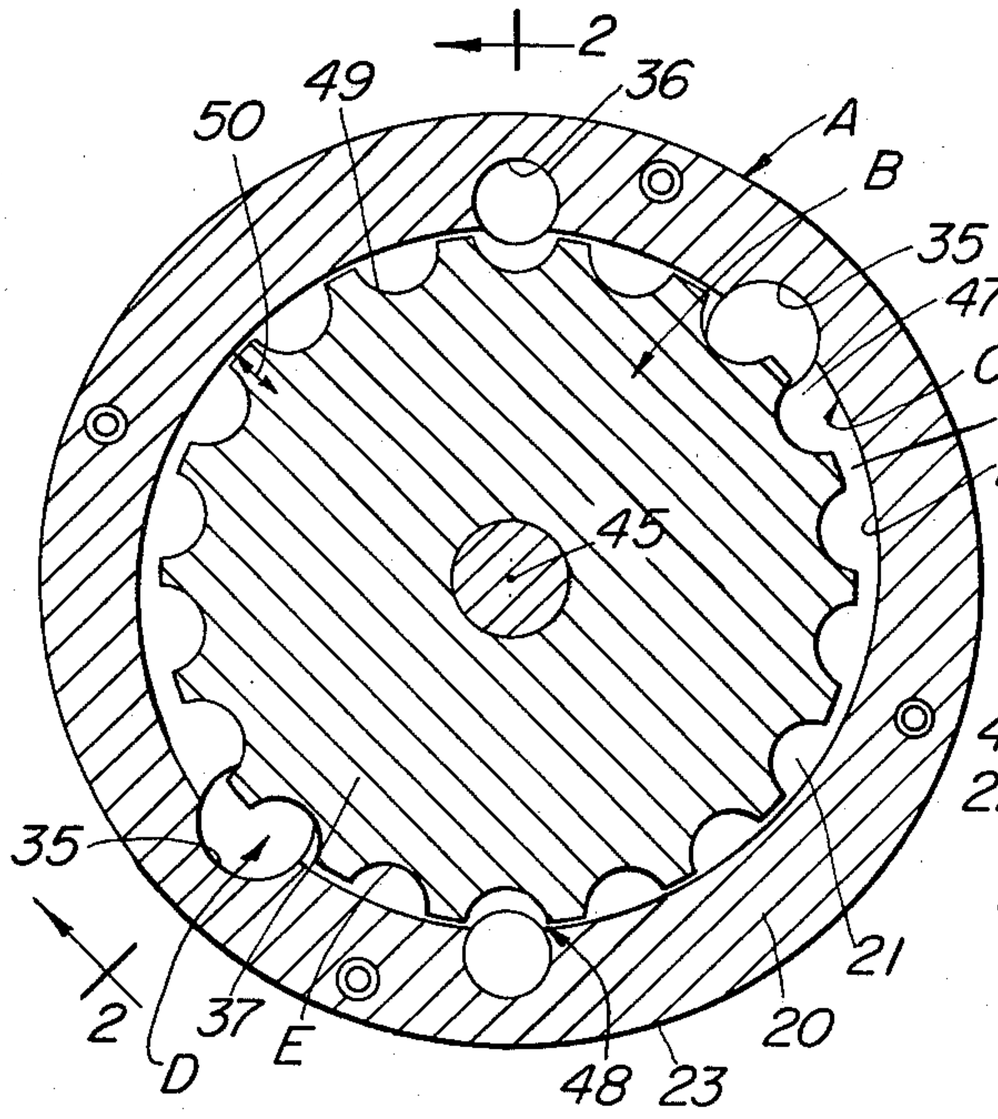


FIG. 2

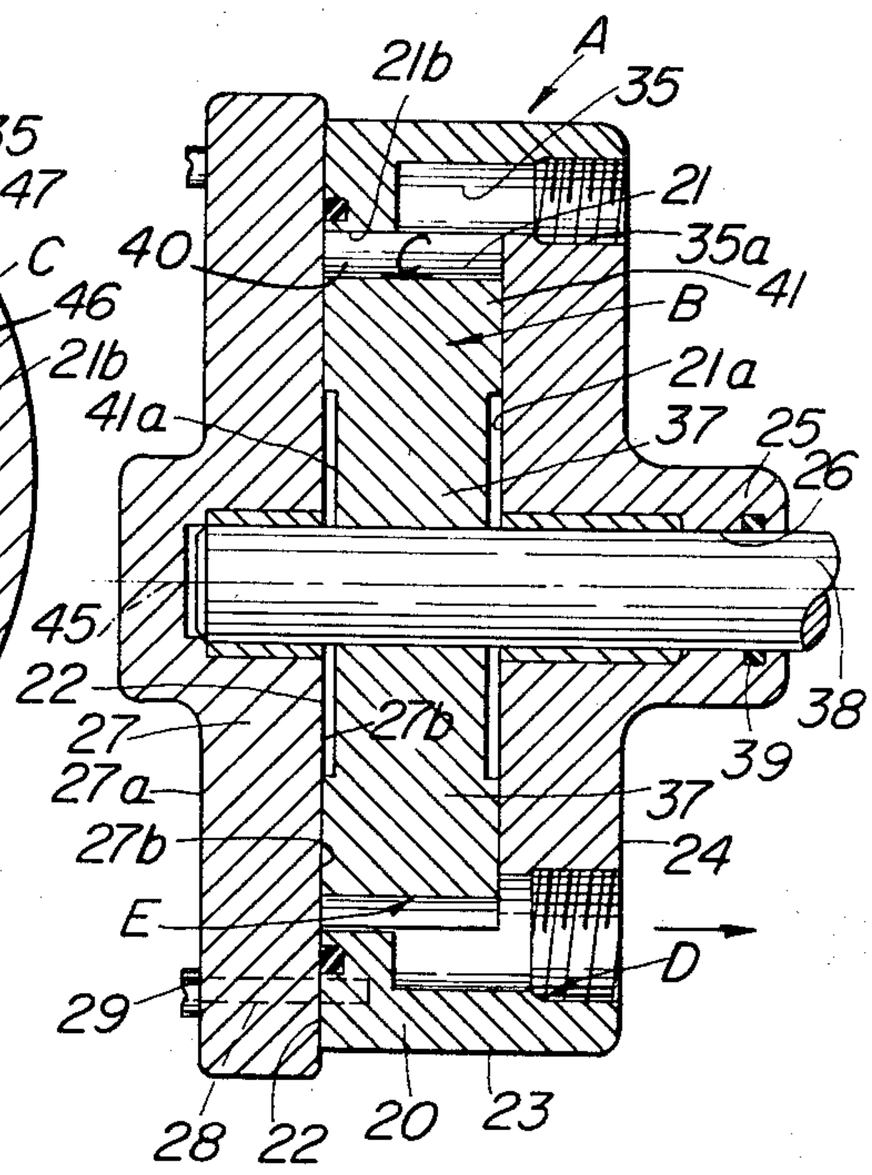
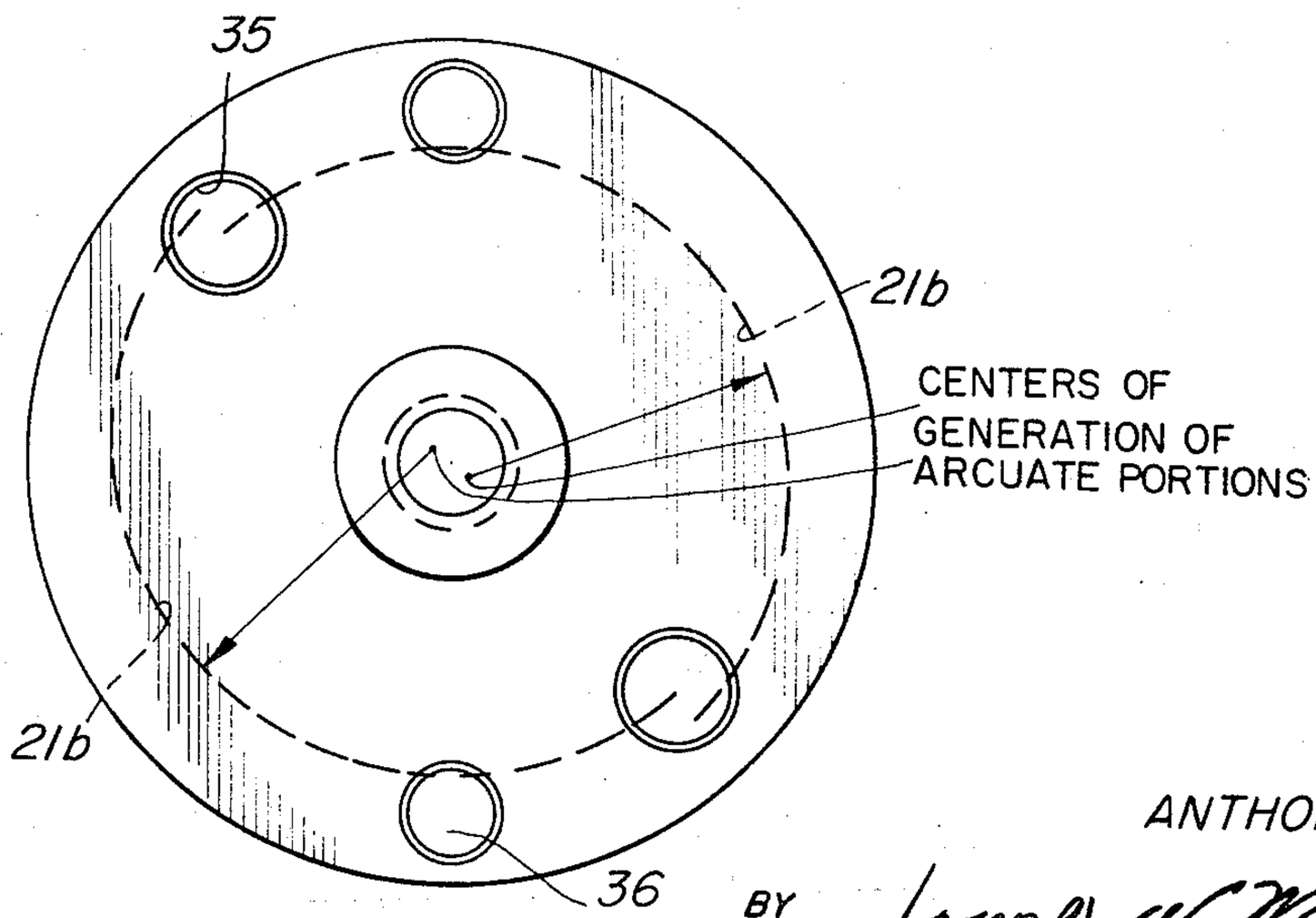


FIG. 3



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Sept. 24, 1968 A. C. MAMO 3,402,669

CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 2

FIG. 4

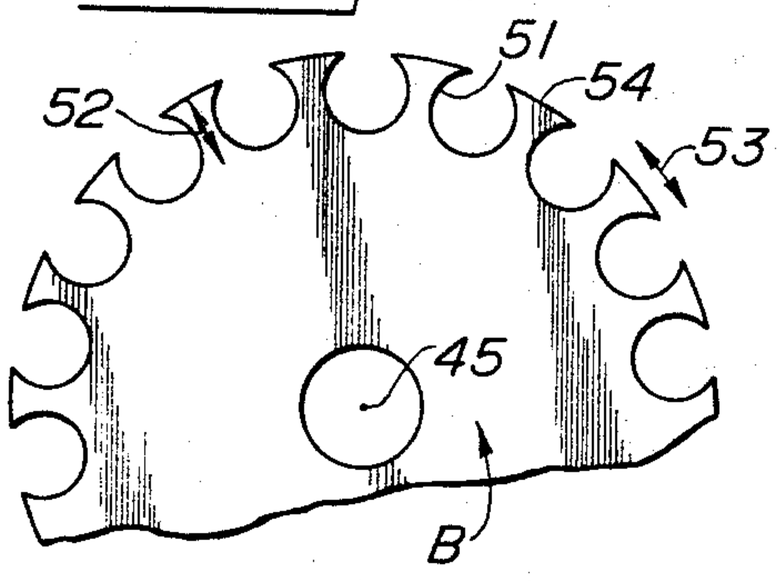


FIG. 5

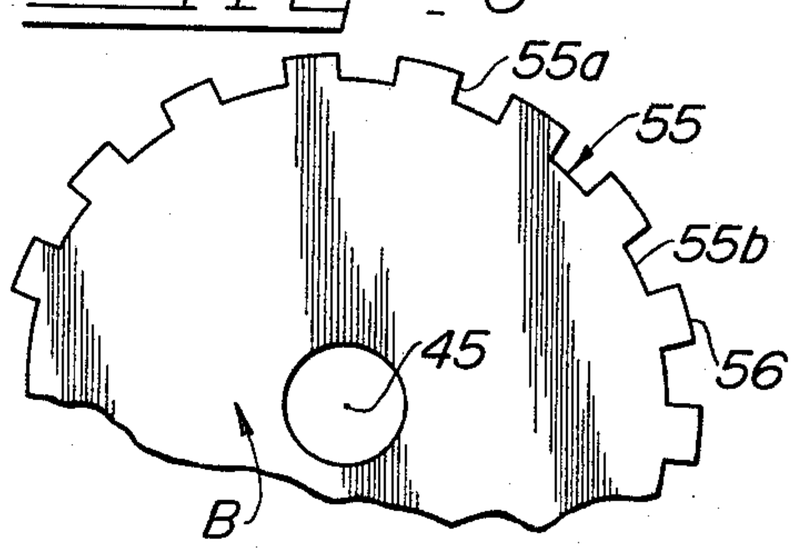


FIG. 7

FIG. 6

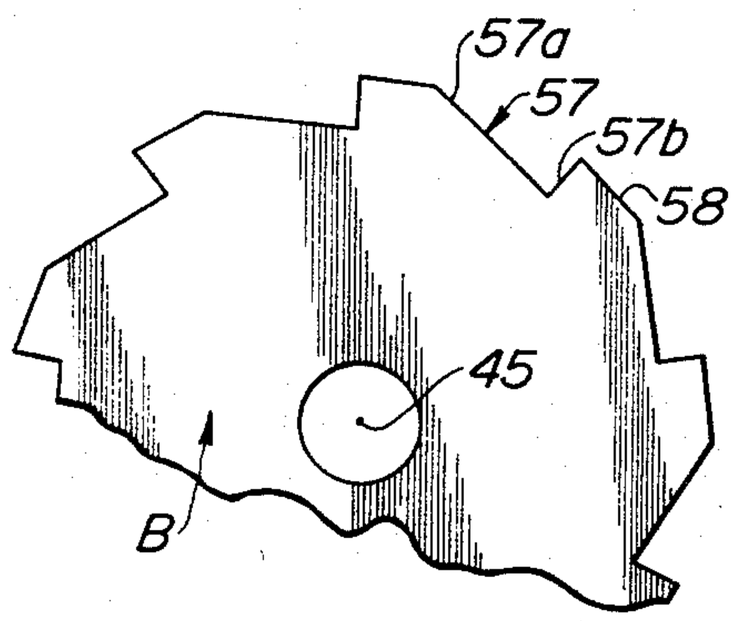


FIG. 8

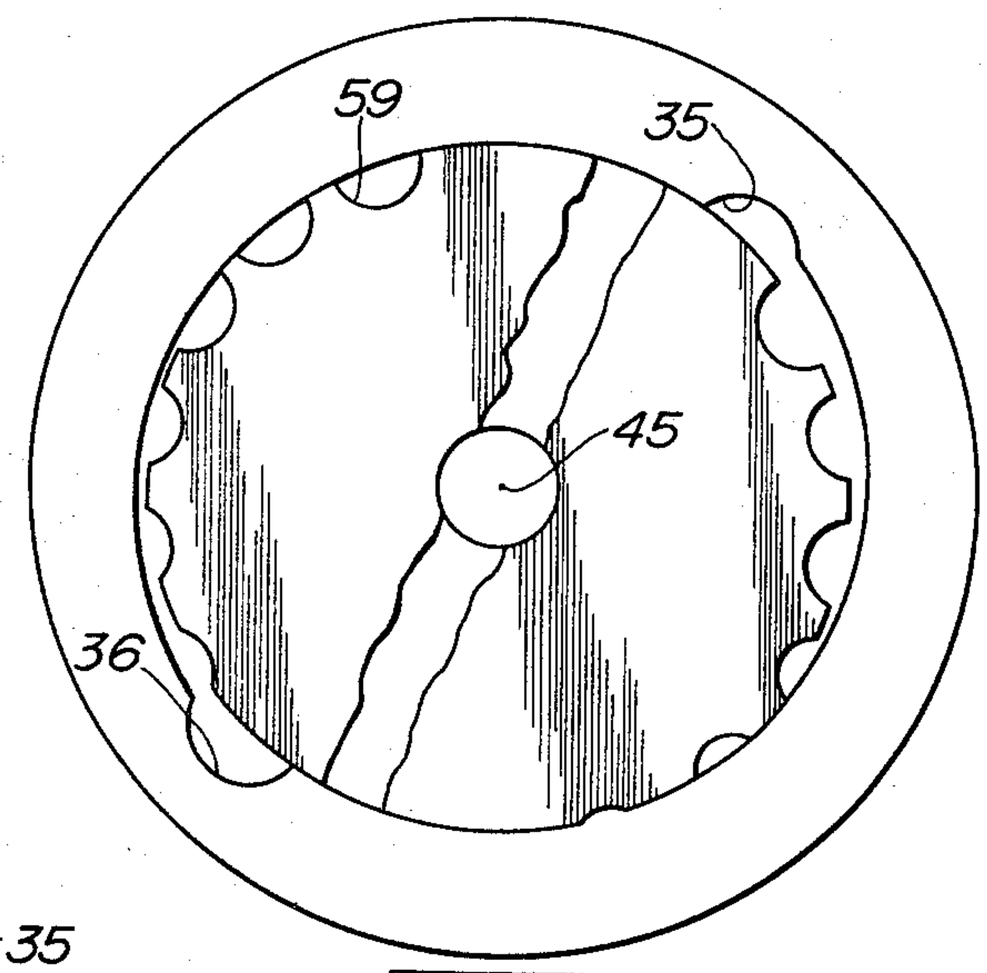
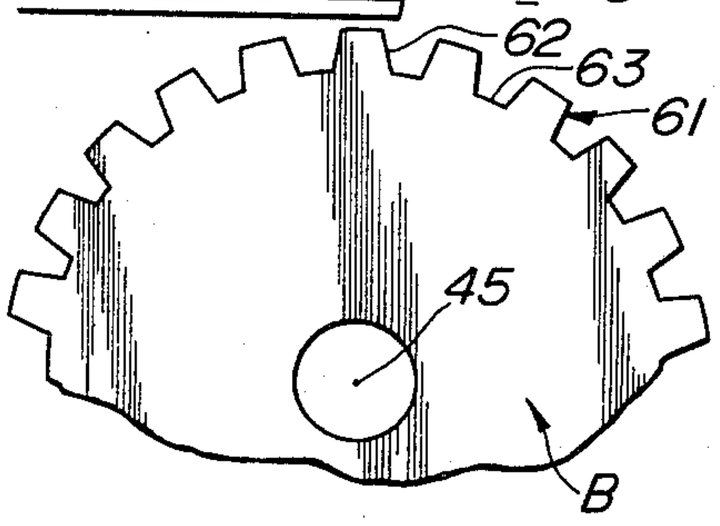
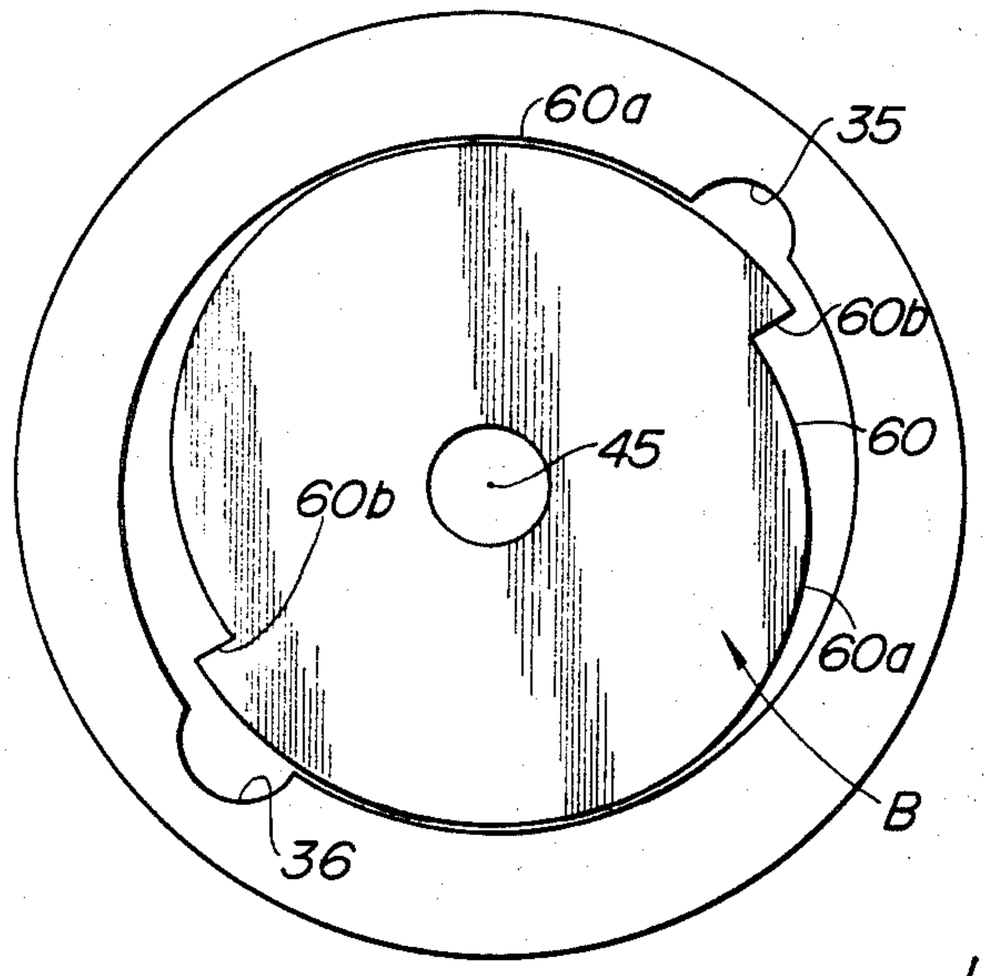


FIG. 9





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Sept. 24, 1968 A. C. MAMO 3,402,669

CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 3

FIG. 10

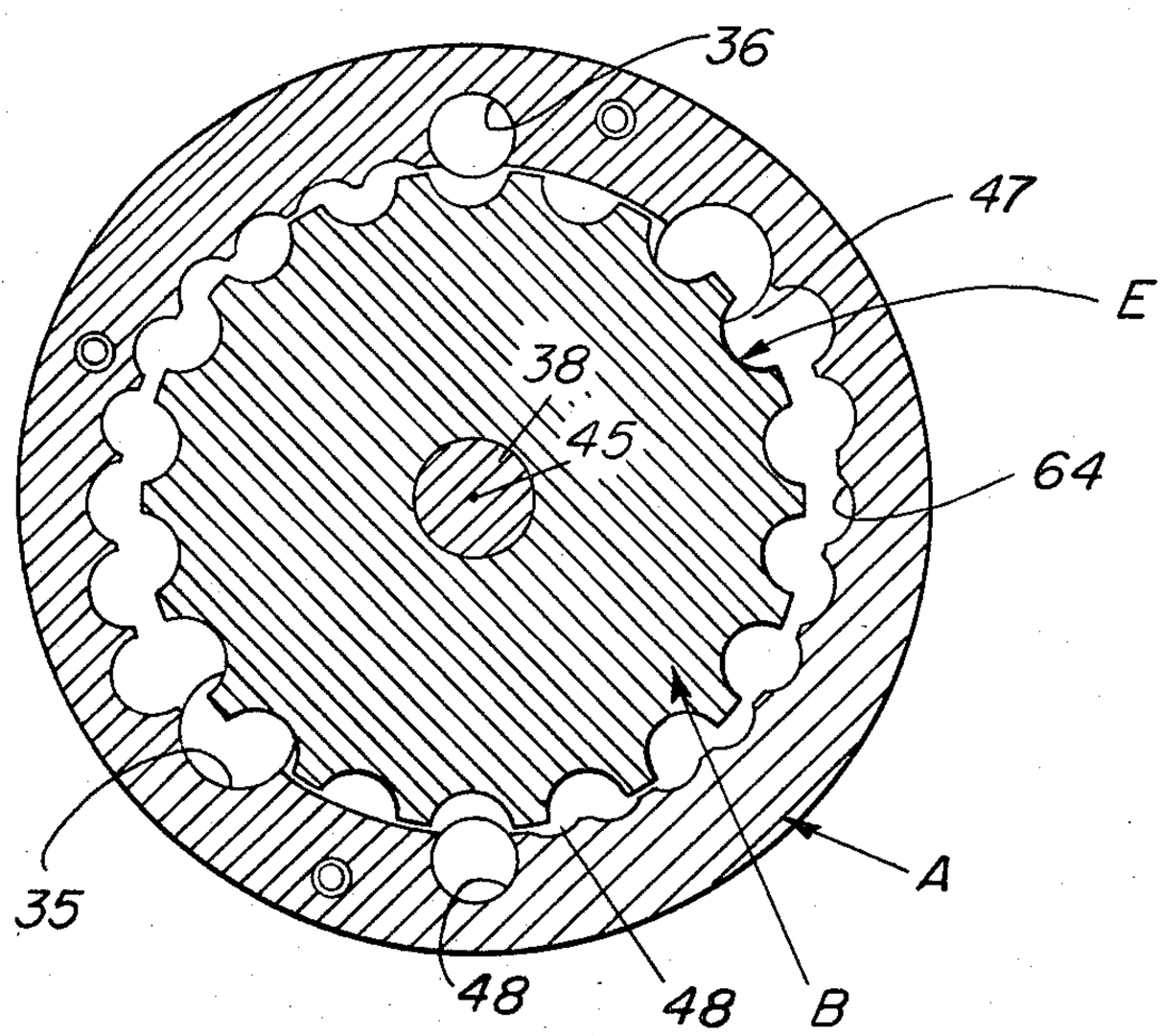
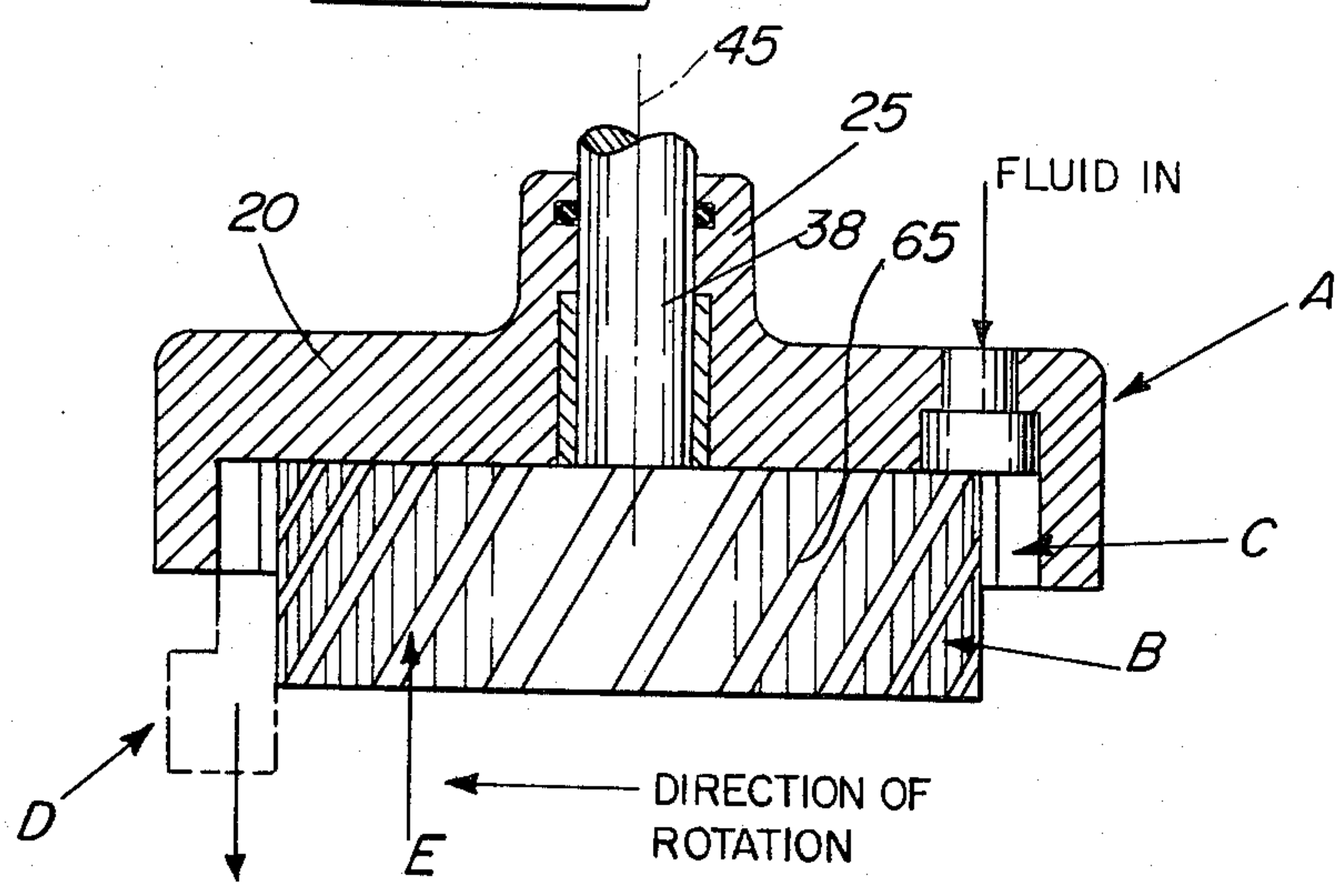


FIG. 11



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Sept. 24, 1968 A. C. MAMO 3,402,669

CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 4

FIG. 12

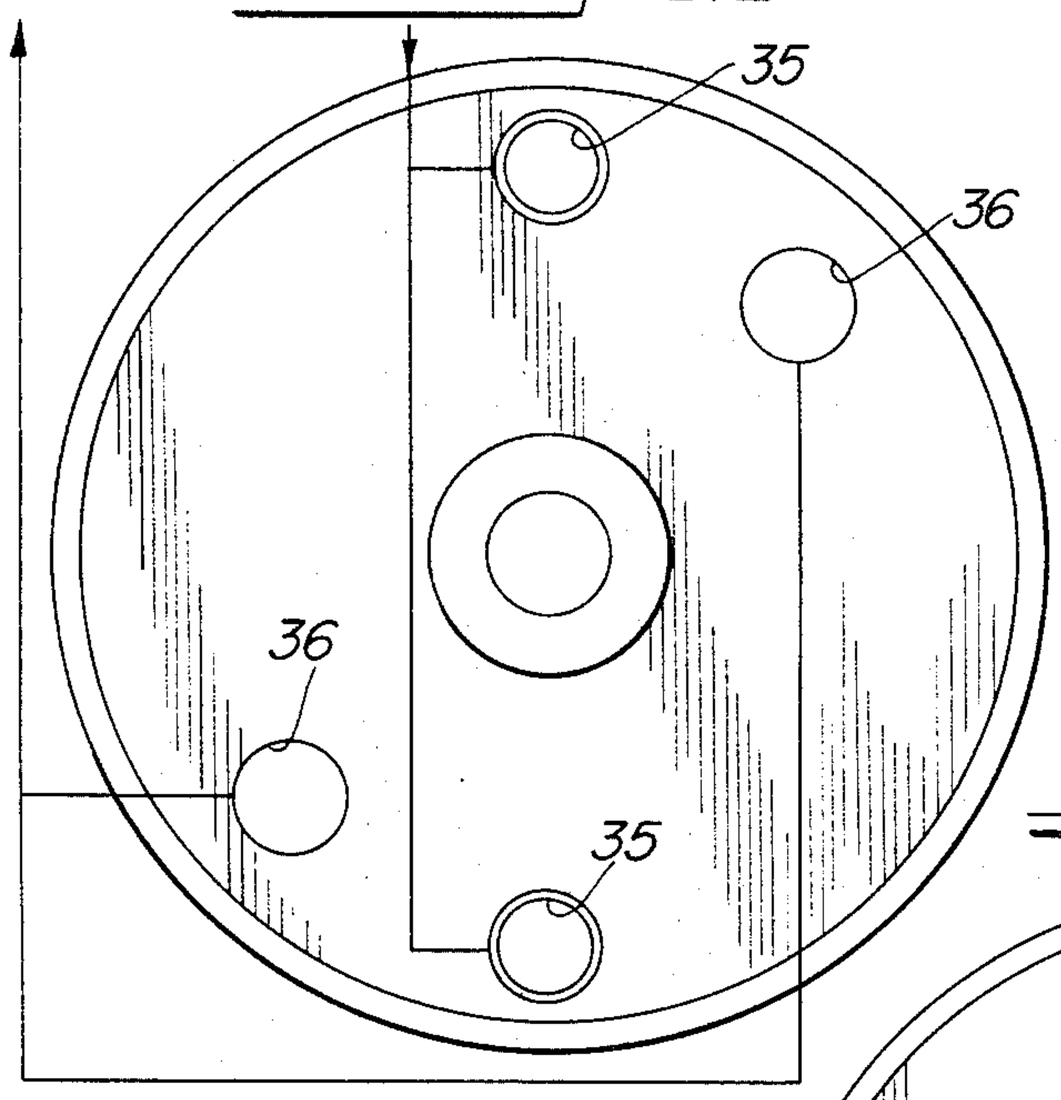


FIG. 13

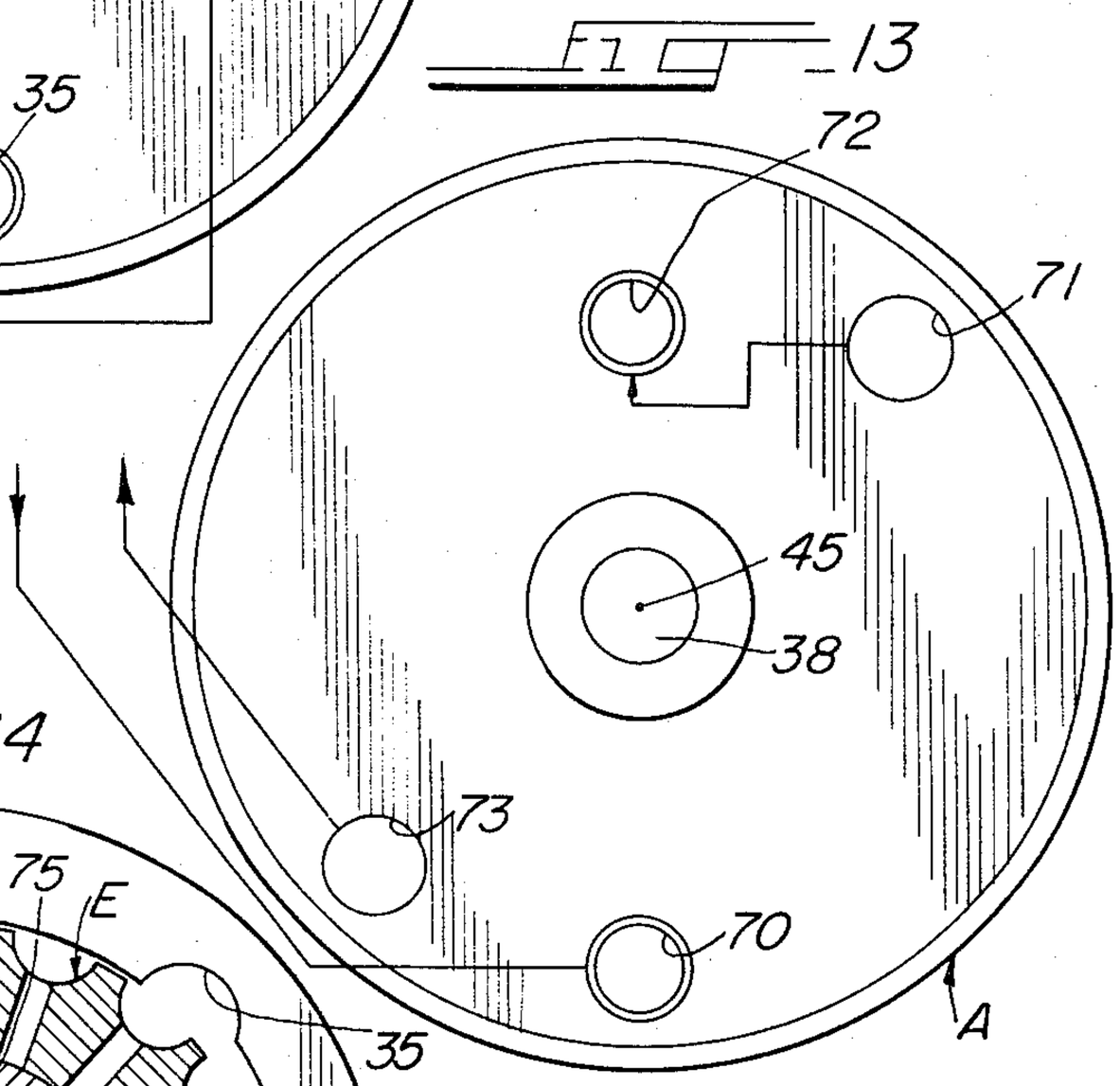
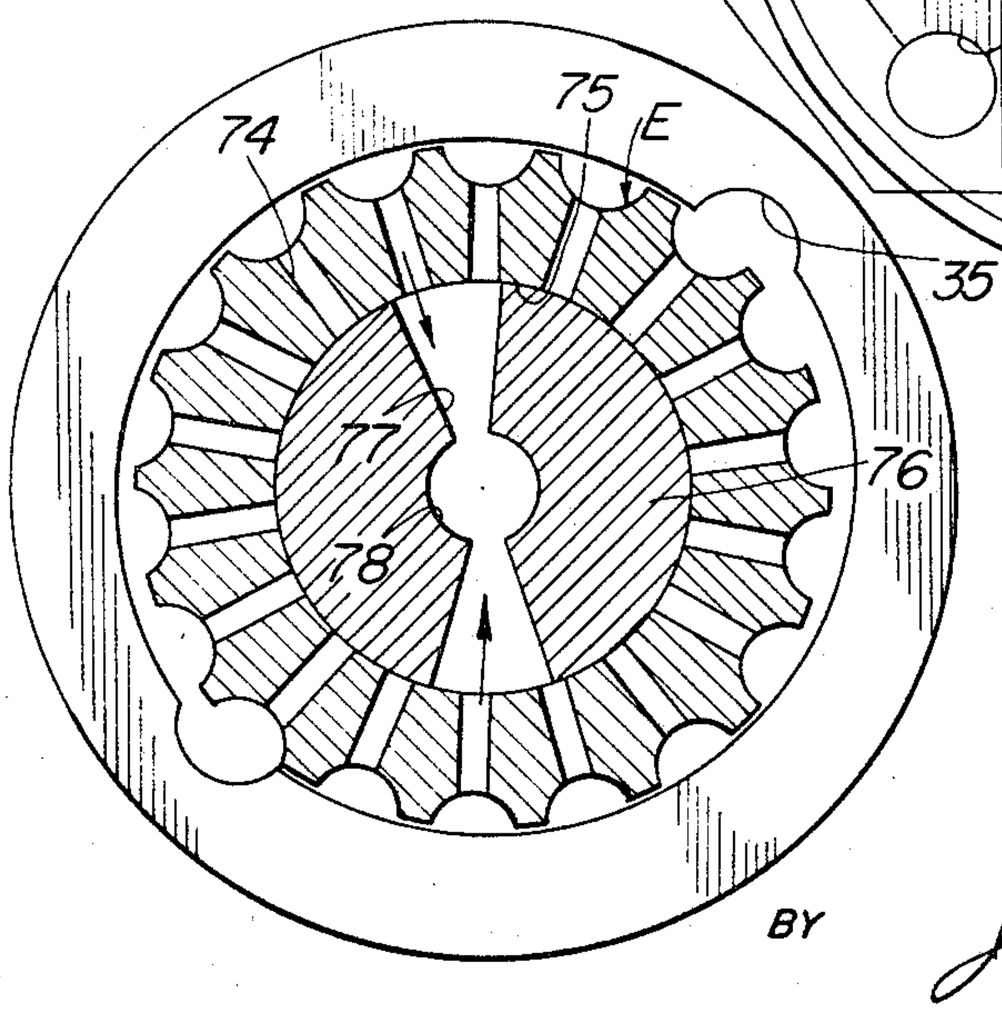


FIG. 14



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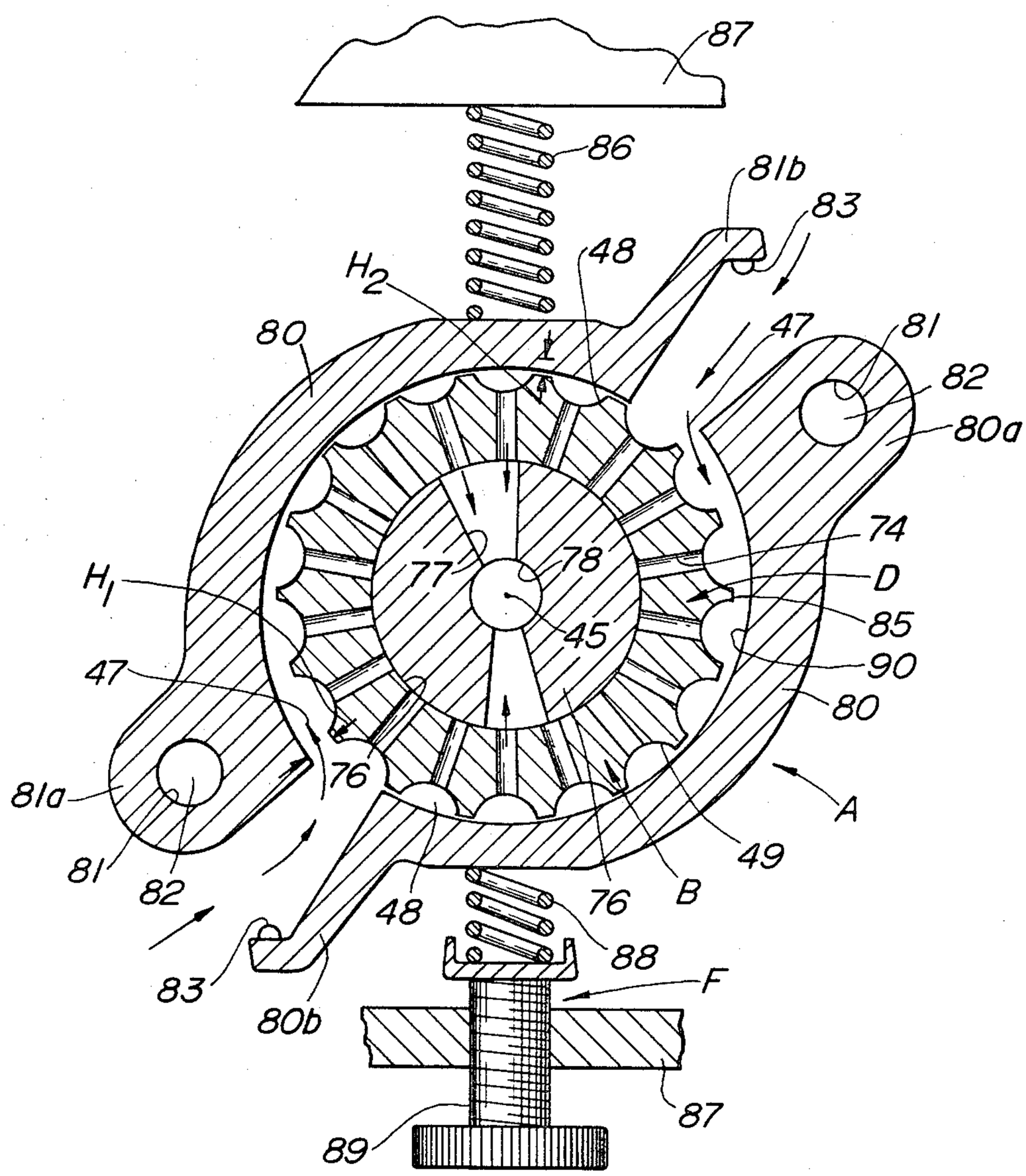
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Sept. 24, 1968 A. C. MAMO 3,402,669

CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 3

Fig. 15



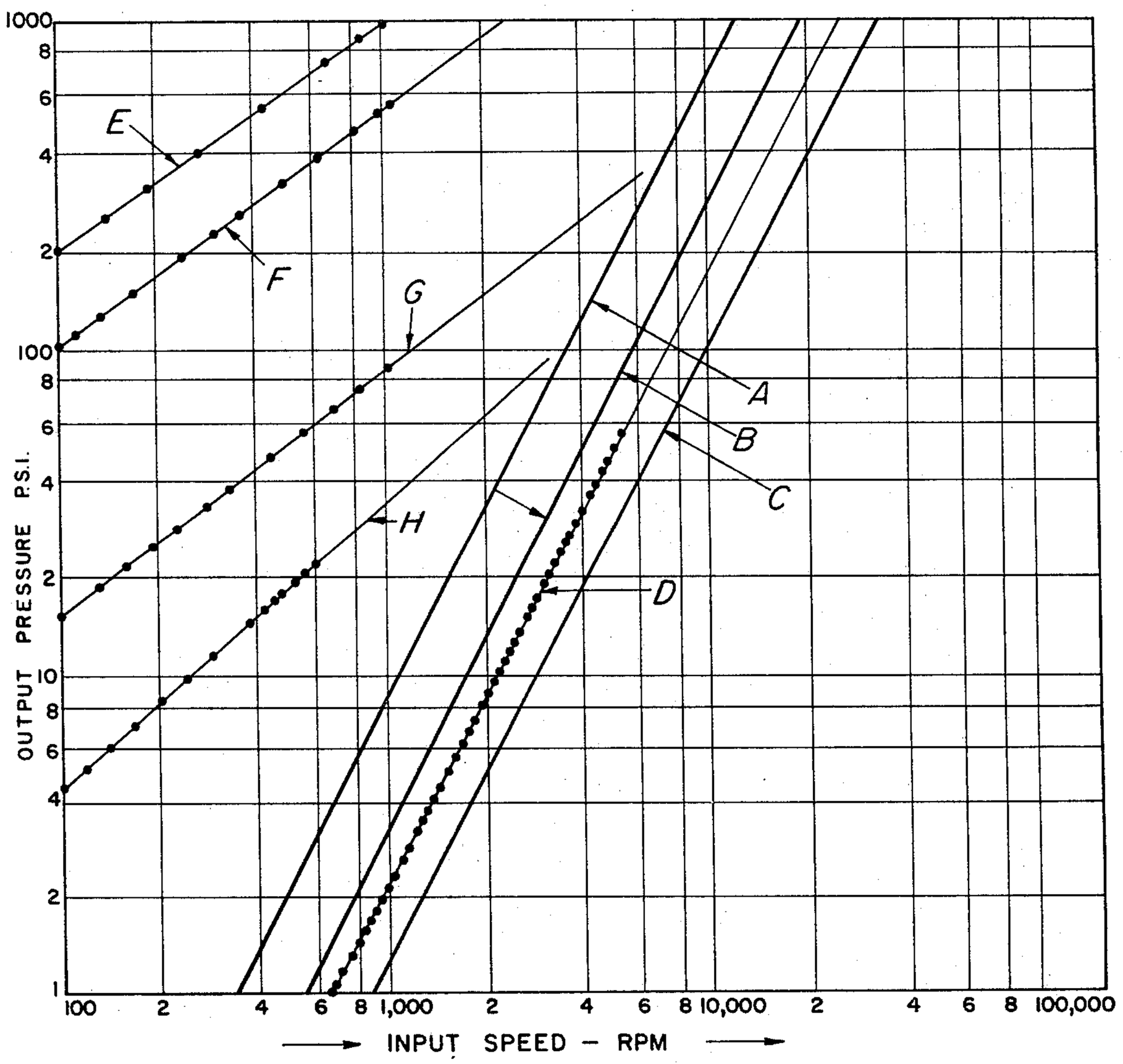
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CENTRIFUGAL INERTIA PUMP

Filed June 17, 1966 6 Sheets-Sheet 6

FIG. 16



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United States Patent Office

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CENTRIFUGAL INERTIA PUMP

Anthony C. Mamo, Arlington Heights, Ill., assignor to Borg-Warner Corporation, Chicago, Ill., a corporation of Illinois

Filed June 17, 1966, Ser. No. 558,370
18 Claims. (Cl. 103—97)

ABSTRACT OF THE DISCLOSURE

A rotary pump has a rotor spaced from the housing. Between the rotor and the housing there is a convergent working space in the direction of rotation of the rotor from the inlet to the outlet. The rotor has a series of grooves on its periphery.

This invention relates to rotary high pressure pumps and improvements therein.

In industrial fields requiring the pumping of fluids at relatively high pressures, the pump structures have typically been of the positive displacement or complex multistaging types in order to achieve such high pressures here being considered. These pump types are characterized by relatively high cost of manufacture and relatively limited durability. In the case of the positive displacement type the higher costs result from the extremely close tolerances and careful machining that is necessary for such high pressure utilization; in the case of multistaging the repetitious structure obviously adds the high cost factor. With regard to the positive displacement type, the useful life of such pumps are relatively limited in any application where the fluid might contain varying sizes of grit material which would affect the predetermined tolerances between the elements. In the multistage type the durability and life span would, of course, be limited by the enhanced sealing problems, as well as magnified axial loads, shaft deflection, and end space. Both of these pump types are noted for progressively poorer performance (efficiency) with increasing viscosity of the pumped fluid.

The single-stage centrifugal rotary type pump is free of disadvantages above noted, but is restricted in attainment of high pumping pressures. The stator and rotor elements of a centrifugal pump are not in contacting relationship and thereby contribute to a very low noise level as well as a desirable life span for the pump. However, assuming an ideal centrifugal pump structure and fluid, the mathematical maximum attained pressure is $V^2/2G$ for the pump or V^2/G for the pump and volute wherein V represents the tip velosity of the rotor.

It is accordingly a primary objective of this invention to provide an improved rotary type pump comprised essentially of two parts (a rotor and a stator housing) which is more economical and durable and can pump a variety of fluids of varying viscosity, while being able to move these fluids at relatively high pressures for long periods of continuous operation without being limited by the conventional centrifugal pump formula $V^2/2G$ or V^2/G.

Another object of this invention is to provide a high pressure type pump which is not dependent upon strict dimensional relationships between the elements, but rather is characterized by a rotor which runs free from the stator housing resulting in an improved noise level of operation and durability.

Another object of this invention is to provide an improved rotary type pump which is capable of pumping fluids at extremely high pressures and which is characterized by an operational efficiency which progressively improves with an increasing viscosity of the fluid being pumped.

Another object is to provide a pump structure in which both axial and radial loads are balanced.

A more particular object of this invention is to provide an improved type pump having a rotor and stator housing effective to define a converging pumping space provided with a fluid entrance and exit, the rotor being movable in the direction of convergence of said space and having one or more convolutions for affecting the fluid within said space.

Particular structural features pursuant to the above objects and which form a part of this invention may comprise (a) the use of convolutions or transverse grooving on the rotor or stator housing which are biased in the direction of movement of the fluid and cooperate with the placement of the fluid entrance and outlet so that the kinetic energy losses of the fluid, passing through the pump, will be reduced and thereby result in optimizing the efficiency; (b) the fluid flow characteristics may be influenced by regulating the transverse dimension of the grooving or convolutions with an increase in width or dimension resulting in more flow, increasing the number of convolutions, or by varying the crescent space at the fluid entrance—all of the foregoing resulting in optimizing the flow characteristics; (c) the employment of variations to optimize the output pressure of the pump such as: shaping the grooving or pockets on the rotor or housing so as to provide a relatively large land area between the pockets while still providing sufficient trapping volume for regeneration purposes, providing multistaging without departing from the concept of the single rotor, or varying the crescent space at the fluid exit.

Other objects and advantages of this invention will become more apparent from the following detailed description taken in connection with the accompanying drawings in which:

FIGURE 1 is a central sectional view of a structure embodying the principles of this invention;

FIGURE 2 is a sectional view taken substantially along line 2—2 of FIGURE 1;

FIGURE 3 is an end elevational view of the structure shown in FIGURE 1;

FIGURES 4–9 illustrate various alternative rotor constructions within the principles of this invention;

FIGURE 10 is a sectional view of an alternative embodiment, illustrating progressive variations in the convolutions defined in both the rotor and housing;

FIGURE 11 is an elevational view of still another embodiment, having the housing shown in section, and illustrating biased groove convolutions on the rotor;

FIGURES 12–14 illustrating several embodiments ilustrating various positioning of the fluid entrances and exits, FIGURE 13 particularly illustrating multiple staging of a plurality of entrances and exits and FIGURE 14 particularly illustrating the fluid exit within the rotor shaft;

FIGURE 15 illustrates an alternative arrangement whereby the volume of the pumping spaces may be automatically varied; and

FIGURE 16 is a graphical illustration of the pump output pressure plotted against the input speed and also illustrating comparative data for pertinent prior art constructions.

Turning now to the drawings and more particularly to FIGURES 1–3, there is illustrated a preferred embodiment of this invention and broadly comprises two elements: a fixedly mounted stator or housing A which in turn carries a power driven rotor B therein. The rotor and stator cooperate to define at least one fluid working space C which converges in the direction of rotation of the rotor; means D is employed for continuously intro-

ducing fluid to and from the working spaces. The capability of generating extremely high pressure results from a hybrid of technical aspects including the "ram" effect of a progressively converging work space (the resistance to escaping fluid varies with the cube of the clearance between rotor and housing), the "regenerative" effect of convolutions E defined in at least one wall of the working space.

In more particularity, the stator is comprised of a first portion **20** and a second portion **27**; portion **20** is cup-shaped having a circular cylindrical outer wall **23** and a hollow interior space **21** opening through flat end wall **22**. Space **21** is defined by flat interior side wall **21***a*, perpendicular to the axis of rotation **45** of the rotor, and by a radially outer essentially cylindrical interior wall **21***b*. The exterior side wall **24**, opposite from that of the wall **22**, is also flat and has a central sleeve **25** projecting therefrom with a cylindrical bore **26** therethrough for journaling a rotor shaft **38**.

The housing portion **27** is comprised of a heavy plate having parallel flat side walls **27***a* and **27***b*. The plate is generally commensurate in thickness to that of the portion **20** between side walls **22** and **24**. The portion **27** is mated against the side wall **22** of portion **20** and is coupled thereto by suitable fasteners **29** extending through aligned openings **28** circumferentially spaced above the margins of portions **20** and **27** thereof.

The power driven rotor B is disposed within the hollow interior space **21** of the stator and comprises a unitary circular disc **37** and is rotatively carried by shaft **38** extending through the bore **26** of housing portion **20**. Suitable seals **39** are disposed within grooving of the portion **20** for sealing against the outer surface of shaft **38**. The rotor has a convoluted cylindrical outer wall **40** concentric about the axis of rotation **45** and has a pair of flat side walls **41** which are recessed at **41***a* to reduce the frictional contact between the sides of the rotor and the side walls **27***b* and **22**, the latter walls being in a "face-sealed" relationship (minimal clearance) or varied to permit up to .10 inch clearance depending on allowable flow requirements and economics.

To provide for pumping action of fluid, the rotor and stator cooperate to define the progressively converging working spaces C. In the preferred embodiment, such taper or convergence is brought about by irregularly shaping the interior wall **21***b* of the stator portion **20** while maintaining the outer periphery of the rotor in a generally circular shape. The irregular definition of the interior wall **21***b* is comprised of one or more (here being two) arcuate portions, each extending circumferentially between a mouth **47** and an apex **48** thereof. The mouth **47** corresponds to the widest radial spacing between the rotor and the interior wall **21***b*; the apex **48** corresponds to the closest relationship therebetween. The geometrical centers for the definition of the arcuate portions are offset from the axis **45** of the rotor; each arcuate portion thereby proceeds from a rather wide-spaced relationship with the outer wall **40** of the rotor adjacent to a mouth **47** associated therewith and progressively narrows to a bearing relationship with wall **40** which is substantially tangential to the outer wall **40** adjacent to an apex **48** thereof. The rotor B is dimensioned so that said bearing fit will be realized without the rotor actually contacting the stator. Other variations for defining the converging spaces C can be envisaged; for example, the wall **21***b* may be made circular and spaced from the rotor—a baffle mounted on the housing may then be employed to achieve the convergence between the circular walls.

In addition to the accelerating flow derived from the converging working spaces, the rotor is provided with a plurality of convolutions E on its outer periphery to aid in developing a regenerative reverse flow. Each of the convolutions or grooves E extend transversely across the other periphery thereof, and in the preferred embodiment they are generally parallel to the axis **45** of rotation of the rotor. The grooves or convolutions are regularly and

equicircumferentially spaced about the periphery of the rotor and their presence define arcuate lands **46** out of the remaining peripheral wall. The grooves form in effect a pocket for recirculating fluid in a centrarotational path as it is carried from a mouth **47** to an apex **48** of the working space. Such recirculating flow in effect forms a vortex whereby energy is added to the fluid in a number of impulses by the rotor's convolutions.

In the preferred embodiment, the pump device is supplied with fluid by means D comprising a pair of fluid inlet passages **35** extending from the outer side **24** of housing portion **20** transversely therethrough and communicating with a peripheral portion of the space **21** and thereby with the working spaces C. The passages **35** are disposed at diametrically opposite positions (see FIGURE 3) of the rotor and each have a threaded nipple **35***a* for reception of suitable conduit couplings; the inner end of the passages **35** intersect with the periphery of wall **21***b*.

Similarly, a pair of fluid outlet passages **36** are provided, each extending from the outer side **24** of the housing portion **20** transversely therethrough and communicating with a peripheral portion of the interior space **21** and thereby with a working space C. The passages **36** are also disposed at diametrically opposite positions (see FIGURE 3) of the rotor and each have a threaded nipple **36***a* for reception of suitable conduit couplings. The inner end of each passage **36** intersects with the periphery of wall **21***b* substantially in a tangential manner without extending entirely through the housing portion **20**. It should be noted that the arcuate working spaces C interposed between each mouth and apex are on directly opposite sides of the device and thereby afford a balanced hydraulic condition of the centrifugal fluid forces. (In FIGURE 31, the hydraulic forces are balanced by nonsymmetric porting.)

The cross sectional configuration of each of the convolutions E can be varied while keeping within the invention herein. In addition to the convolutions of the preferred embodiment, FIGURES 4–9 illustrate other variations, all of the varied convolutions being regularly or equicircumferentially spaced about the axis of rotation and each extending entirely transversely across the outer wall of the rotor. In FIGURE 1, the particular grooves **49** are semi-circular in cross section and project below the outer surface of the rotor a depth **50** which is generally equal to the radius defining the cross sectional shape of each groove **49**; the grooves are spaced apart a distance such that the remaining land **46** therebetween is generally equal in circumferential dimension to the radius of a groove **49**. In FIGURE 4, substantially cylindrical tunnels or T-grooves **51** are defined in the outer wall of the rotor and project to a depth **52** which is greater than the radius of the circular cross section of each groove **51** but less than the complete diameter thereof; the grooves **51** define substantially larger lands **54** which approximate the diameter of the grooves **51**. In FIGURE 5, the grooves **55** have straight side walls **55***a* interconnected by a flat bottom wall **55***b*; the circumferential width of each groove **55** is generally equal to the lands **56** therebetween and the depth of each groove being generally less than the width thereof. In FIGURE 6, the regularly spaced grooves **57** are V-shaped with one wall **57***a* considerably longer than the other wall **57***b*, the depth of the V-shaped groove being generally one half the width of the land **58** defined between the grooves **57**. In FIGURE 7, the groove **59** initially are defined similar to FIGURE 1, but have a progressive reduction of the volume of each groove such that as they proceed toward the apex of the working space the grooves **59** are reduced to a nil volume. This can be brought about by progressively moving the center of definition of each groove **59** outwardly beyond the outer wall of the rotor. In FIGURE 8, there are essentially two diametrically opposed grooves **60** each having an extremely long leading arcuate surface **60***a* which terminates against a short radial directed wall **60***b* which together define the apex of the groove. In FIGURE 9, regu-

larly spaced grooves **61**, similar to gear tooth construction, are defined; each groove has converging flat sidewalls **62** which terminate in a bottom wall **63**. The depth of each groove **61** is greater than the depth of grooving shown in FIGURE 5.

Further variations can be incorporated into the design of the device such as shown in FIGURES 10 and 11; FIGURE 10 illustrates the additional definition of convolutions **64** on the wall **21***b* of the housing as well as the rotor. These convolutions **64** have a semi-circular configuration; they are initially similar to the convolutions of FIGURE 1 adjacent to the mouth **47** and progressively become less in volume as they proceed toward the apex **48**. The convolutions E as defined on the rotor are of the type shown and illustrated in FIGURE 1 and have a slightly greater radius than the grooving **64** on the interior wall of the housing. A double inducement of local regenerative flow is provided by such radially inner and outer grooving.

In FIGURE 11, the transverse grooving **65** is given a biased path relative to the axis **45** of rotation. The bias is in the direction of rotative movement of the fluid and cooperates with the placement of the fluid entrances and outlets (see FIGURE 11) on opposite sides of the rotor so that kinetic energy losses of the fluid, passing through the pump, will be reduced and thereby result in optimizing the efficiency. To facilitate the placement of the fluid entrances and outlets, the passages **35** may be defined in the housing portion **20** as described in connection with FIGURE 1 and the fluid outlets **36** may be defined in the housing plate **27**. Further, the plate **27** may be adjustably rotatable relative to the housing portion **20** thereby providing ability to vary the circumferential dimension of the converging spaces C thereby resulting in a means for optimizing the performance characteristics of the pump for each particular application. The relative movement of the entrances and outlets may be used without the combination of biased grooving for optimizing pump operation over a wide speed range.

Turning now to FIGURES 12 and 13, systems for operatively connecting the various fluid passages of the pump are illustrated. In the preferred embodiment of FIGURE 1, the fluid entrances **35** and fluid exits **36** are independent from each other, although the plurality of entrances may be fed from a common line and similarly the plurality of fluid outlets may be communicated to a common line (see FIGURE 12). However, other arrangements may be envisioned for this type of pump wherein multistaging can be incorporated. In FIGURE 13, the schematic shows the first fluid entrance **70** connected to a principal source of fluid, the first fluid outlet **71**, instead of being communicated with a conduit leading from the pump, is connected to the second fluid entrance **72**. thereby immediately reworking the output of the first working space and sending it to a second stage of the pump by way of fluid entrance **72**. The total fluid output is taken from fluid outlet **73** and applied to a specific load. Thus, it is a distinct novel characteristic of this pump that permits it to be easily connected for multistaging with the *same rotor in the same housing* (in counterdistinction to prior art where additional rotors must be utilized) and without changing the flow radii. With this type of multistaging, the working spaces may be irregularly spaced about the axis **45** to balance a constantly increasing pressure through the several spaces C.

In FIGURE 14, each of the convolutions, comprised in the series E, have independent radial passage **74** which communicate with an interior periphery **75** of the rotor; stationed within the hollowed interior is a hub **76** having at least two radially diverging channels **77** in communication with a central axially directed fluid exit **78**. Fluid is drawn into the pump working spaces by way of typical fluid entrances **35** and leave the working spaces by way of the centripetal action of the fluid tending to work radialy inwardly through passages **74**, channel **77** into central outlet **78**.

FIGURE 15 illustrates another embodiment which provides for governing the output pressure in a uniform manner while varying the flow characteristics through the device. To this end, the embodiment of FIGURE 15 schematically incorporates two housing parts **80** and **81** each arcuately shaped cooperating with side walls (not shown) which are generally fixed. The arcuate parts each have one end **80***a* and **81***a* respectively provided with a journal opening **81** adapted to fit about a fixed pivot **82**; the opposite end of the arcuate parts have an arm **80***b* and **81***b* respectively effective to engage one of a pair of stops **83**. The fixed pivots **82** are positioned such that they are at diametrically opposite positions relative to the rotor B and permit the arcuate parts to be moved to and from the axis of rotation **45** of the device thereby regulating the clearances adjacent the fluid exits and thereby regulate the apex **48** of each working space. Each of the arcuate parts have an inner cylindrical wall **90** which is defined relative to the fixed pivot so that it will be nonconcentric with respect to the axis of rotation **45**.

To achieve a regulated or governed outlet pressure a biasing means F is employed which incorporates a pair of coiled compression springs **86** and **88** effective to normally resiliently urge the ends **80***b* and **81***b* of the arcuate parts toward a closed or decreasing clearance position. The degree to which the springs exert their resilient force may be adjusted by a thumb screw **89** carried in a reaction wall **87**. The conveyance of fluid to the working spaces may be as shown in FIGURE 1 and conveyance of the fluid away from the working spaces may be as that shown in FIGURE 14 providing for a centripetal action. By adjustment of the thumb screw, the exit clearances can be changed and thereby the governed pressure may be varied to achieve an optimum outlet pressure for the given conditions of the pump and the type of fluid being carried. The adjustment of the clearance between the arcuate housing parts and the rotor gives rise to a divergence ratio H_2 over H_1 (H_2 represents the clearance adjacent the mouth of each working space and H_1 represents the clearance adjacent the apex of each working space). This ratio can be influenced by varying H_2 as in the embodiment of FIGURE 14 or H_1 can be varied to provide a variable displacement pump. The pump can readily be converted into a variable displacement type by arranging the pivots **82** to be adjacent the fluid exits so that the clearance adjacent the entrance may be more readily adjusted. In this way the slope of the working space at its mouth can be changed and the degree of flow passing through the pump can be varied as opposed to the ultimate outlet pressure.

To illustrate the comparative characteristics of this invention as related to typical single rotor pumps that have been used for achieving the high pressures, reference is made to FIGURE 16. In this chart, pumps of the same working diameter have been utilized (here being 3 inches). Pump pressure is plotted against the input speed of the pump rotor on logarithmic paper to define relatively straight line plots. Plot A represents the variation of output pressure for a typical regenerative type centrifugal turbine pump such as that illustrated in the U.S. Patents 1,665,687 or 2,724,338; plot B represents the output pressure variation for a typical pitot tube pump; plot C represents the output pressure variation for a typical centrifugal pump. The first two of these pumps are characteristically known for their high output pressure from a single rotor. Each of these typical prior art pumps were rated with water as the pumped fluid except that a subplot D was provided for the regenerative type pump to show the performance drop when a viscous fluid is substituted for water, the viscous fluid having a rating of 3,000 SUS. Plot D shows how increase of viscosity of the pumped fluid reduces output performance.

Various embodiments of the pump of this invention

were tested and all showed plot variations considerably higher in output pressure for any given input speed than that shown for the prior art centrifugal, or one moving part structures. For example, a rotor with 40 convolutions along its outer periphery with lands **46** dimensioned with a width of .125 of an inch gave the plot E (pump construction similar to FIGURE 1). Similarly a rotor with 18 equicircumferentially spaced convolutions and a land dimension of .375 of an inch was also tested to give plot F (construction also similar to FIGURE 1). Both plots E and F were derived from a pump using a fluid with a viscosity of 3,000 SUS at 100° F. Plots G and H were respectively for rotors having the same characteristics as plots E and F but tested with a fluid medium having a viscosity of 200 SUS at 100° F.

It is clear from this comparative chart that the output pressures from the construction of this invention were outstandingly greater than that for the closest known prior art constructions and that pump efficiency rises with an increase of viscosity of the pump fluid contrary to the prior art. The pump output flows are consistently steady, the noise level comparatively low, and cavitation was not observed or induced in any combination of flows and pressures for the range of embodiments illustrated. The nonexistence of sliding parts or rubbing parts in the pump permits this device to be used at very high rotative speeds, even encompasing gas turbine speeds. The operating speeds would not necessarily be limited by the strength of materials used for the rotor as the generated pressures subtract from the stress of the rotating parts.

The various alternative suggestions show the complete versatility of this type of pump whereby optimum efficiency may be achieved by varying the type of grooving and particularly biasing the grooving relative to the direction of flow; the optimum flow characteristics may be achieved by varying the number of convolutions, the divergence ratio of the working space at the mouth thereof, and by varying the width of the pump taken in a direction parallel to the axis of rotation; the optimum pressure characteristics may be achieved by varying the working space (the divergence ratio) at the apex thereof, varying the volume of the convolutions and the land area as well as the particular shape of the grooving, and multistaging of the fluid entrances and outlets by connecting them in series.

The extreme economy of this two part pump and its exhibited characteristics would be relatively useful in such applications as oil furnace pumps where its greater economy, longer life and lower noise level, would be suitable. Other areas of application might include means for pumping plastic, molten metal, blood, liquid rubber, paints and viscous cosmetics. This pump may also be used in areas such as variable capacity torque transmitting devices comparative to fluid couplings and fluid sheer mechanisms because of its exceptionally high K factor.

While I have described my invention in connection with one specific embodiment and other alternative suggestions, it is to be understood that this is by way of illustration and not by way of limitation and the scope of my invention is defined solely by the appended claims which should be construed as broadly as the prior art will permit.

I claim:

1. A pump device comprising a stator and a power driven rotor cooperatively related to define at least one fluid working space therebetween, said space being characterized by a progressive convergence between a mouth portion and an apex portion thereof, said progressive convergence being in the general direction of movement of said rotor and proceeding from said mouth portion to said apex portion said rotor having at least one convolution formed therein to define a pocket for inducing regeneration of fluid as said pocket is exposed to the working fluid space during operation of said rotor, and means for continuously introducing fluid to the mouth of said working space and for continuously exhausting pressurized fluid from the apex portion of said working space.

2. A pump device comprising: a housing defining an interior fluid chamber having an annular wall bounded by a pair of spaced flat side walls; a power driven rotor mounted within said chamber for rotation about a central axis, said rotor having side walls in face-sealed relation with the side walls of said housing chamber, said rotor having an outer annular wall in close contiguous relationship with only a portion of the housing chamber annular wall while other portions of said housing are spaced therefrom, means defining a fluid entrance and a fluid exit for said space, said rotor and housing cooperating to define at least one progressively convergent working space between said fluid entrance and said fluid exit, said rotor having at least one transverse groove defined in the rotor annular wall effective to stimulate a secondary compressible flow of fluid as said rotor turns; and means for continuously introducing fluid to said entrance and means for continuously exhausting pressurized fluid from said exit.

3. A pump device, as in claim **2**, in which said rotor is provided with a plurality of uniformly shaped grooves in the annular wall thereof, said grooves being regularly spaced circumferentially thereabout.

4. A pump device, as in claim **2**, in which said housing has means for automatically adjusting the volume of the working space defined between said housing and rotor to compensate for variations in output pressure thereby governing the output pressure at a predetermined value.

5. A pump device, as in claim **2**, in which said housing and rotor cooperate to define a plurality of progressively convergent working spaces each having an independent entrance and exit.

6. A pump device, as in claim **3**, in which said spaces are irregularly arranged about the circumference of said devices.

7. A pump device, as in claim **2**, in which said housing has a pair of movable parts, each part thereof being arcuate and having a fixed pivot at one end for rockably adjusting the spacing between said annular walls to vary the volume thereof, and resilient means normally biasing said parts to one position for governing the output pressure thereby.

8. A pump device, as in claim **3**, in which each groove has a silhouette, taken along a plane transverse to the axis of rotation, which is generally a sector of a circle.

9. A pump device, as in claim **3**, in which the grooves progressively vary in shape along the circumference of said device.

10. A pump device, as in claim **4**, in which said fluid entrance and exit are on opposite axial sides of said housing, and the transverse centerline of said grooves extend on a bias with respect to the axis of rotation of the device, said bias being generally in the direction of movement of fluid from the entrance to the exit thereby augmenting efficiency.

11. A pump device, as in claim **2**, in which said means for introducing fluid and exhausting fluid comprise transverse passages in said housing effective to respectively communicate with the entrance and exit of said working space.

12. A pump device, as in claim **2**, in which said means for introducing and exhausting fluid comprises radial passages in said rotor communicating with central passages extending along the axis of rotation of the device.

13. A pump device, as in claim **1**, in which said rotor is provided with a plurality of uniformly shaped grooves in the annular wall thereof, said grooves being regularly spaced circumferentially thereabout.

14. A pump device, as in claim **1**, in which said housing has means for automatically adjusting the volume of the working space defined between said housing and rotor to compensate for variations in output pressure thereby governing the output pressure at a predetermined value.

15. A pump device, as in claim **13**, in which each groove has a silhouette, taken along a plane transverse

to the axis of rotation, which is generally a sector of a circle.

16. A pump device, as in claim **13**, in which the grooves progressively vary in shape along the circumference of said device.

17. A pump device, as in claim **1**, in which said means for introducing fluid and exhausting fluid comprise transverse passages in said housing effective to respectively communicate with the entrance and exit of said working space.

18. A pump device, as in claim **1**, in which said means for introducing and exhausting fluid comprises radial passages in said rotor communicating with central passages extending along the axis of rotation of the device.

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