

(19) World Intellectual Property Organization
International Bureau



(43) International Publication Date
17 April 2008 (17.04.2008)

PCT

(10) International Publication Number
WO 2008/045368 A2

(51) International Patent Classification:
B60K 17/14 (2006.01)

(21) International Application Number:
PCT/US2007/021470

(22) International Filing Date: 5 October 2007 (05.10.2007)

(25) Filing Language: English

(26) Publication Language: English

(30) Priority Data:
60/850,230 5 October 2006 (05.10.2006) US

(71) Applicant (for all designated States except US): **FOLSOM TECHNOLOGIES INTERNATIONAL LLC** [US/US]; 3261 Old Washington Road, Suite 2011, Waldorf, MD 20602 (US).

(72) Inventors; and

(75) Inventors/Applicants (for US only): **FOLSOM, Lawrence, R.** [US/US]; 47 Malibu Hill, Rensselaer, NY 12114 (US). **TUCKER, Clive** [GB/US]; 86B Housatonic Street, Lenox, MD 01240 (US).

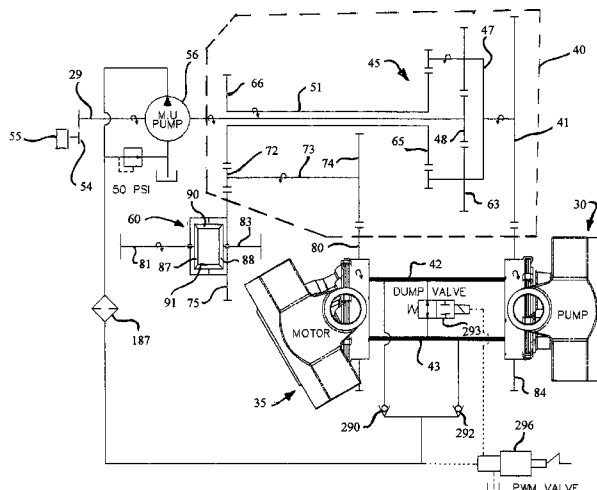
(81) Designated States (unless otherwise indicated, for every kind of national protection available): AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BH, BR, BW, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DO, DZ, EC, EE, EG, ES, FI, GB, GD, GE, GH, GM, GT, HN, HR, HU, ID, IL, IN, IS, JP, KE, KG, KM, KN, KP, KR, KZ, LA, LC, LK, LR, LS, LT, LU, LY, MA, MD, ME, MG, MK, MN, MW, MX, MY, MZ, NA, NG, NI, NO, NZ, OM, PG, PH, PL, PT, RO, RS, RU, SC, SD, SE, SG, SK, SL, SM, SV, SY, TJ, TM, TN, TR, TT, TZ, UA, UG, US, UZ, VC, VN, ZA, ZM, ZW.

(84) Designated States (unless otherwise indicated, for every kind of regional protection available): ARIPO (BW, GH, GM, KE, LS, MW, MZ, NA, SD, SL, SZ, TZ, UG, ZM, ZW), Eurasian (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European (AT, BE, BG, CH, CY, CZ, DE, DK, EE, ES, FI, FR, GB, GR, HU, IE, IS, IT, LT, LU, LV, MC, MT, NL, PL, PT, RO, SE, SI, SK, TR), OAPI (BF, BJ, CF, CG, CI, CM, GA, GN, GQ, GW, ML, MR, NE, SN, TD, TG).

Published:

— without international search report and to be republished upon receipt of that report

(54) Title: HYDROMECHANICAL CONTINUOUSLY VARIABLE TRANSAXLE TRANSMISSIONS



(57) Abstract: A hydromechanical continuously variable power transmission for converting rotating mechanical power at one combination of rotational velocity and torque to another combination of rotational velocity and torque over a continuous range, includes a hydraulic pump, operatively driven by an input shaft, and a hydraulic motor operatively driving an output shaft. The hydraulic pump and hydraulic motor are coupled together mechanically through an epicyclic gear set, and are coupled together hydraulically through a manifold, such that hydraulic fluid pressurized by said pump drives the motor, and spent fluid from the motor is cycled back to the pump where it is re-pressurized. The epicyclic gear set includes first and second planet gear sets coupled together such that the first planet gear set is coupled to the pump, and the motor is coupled to the second planet gear set and the output shaft, and the input shaft is drivingly coupled to the second planet gear set.

HYDROMECHANICAL CONTINUOUSLY VARIABLE TRANSAXLE TRANSMISSION

This is related to U.S. Provisional Application No. 60/850,230 filed on October 5, 2006, Hydromechanical Continuously Variable TransaxleTransmission.

5 This invention pertains to hydro-mechanical power transmissions, and more particularly to a continuously variable hydromechanical power transmission for use especially in front wheel drive (FWD) transaxle applications where an underdrive final ratio is desired, and where the performance of the hydrostatic units and the overall packaging are optimized to achieve a small and lightweight transmission able to
10 accommodate the highest power engines currently available for the FWD automotive application.

BACKGROUND OF THE INVENTION

Continuously variable transmissions have become recognized in recent years as
15 particularly desirable as vehicle power transmissions because of the operational efficiencies and economies that they can potentially afford. However, conventional prior art hydrostatic transmissions are known by experts in the art to be noisy and inefficient, and mechanical CVT's have power and durability limitations. To achieve significant usage throughout the potential vehicle transmission market, significant improvements in
20 cost, durability, performance, and power capacity will be needed.

One such improvement would be in the area of leakage from rotating interfaces, particularly those where working fluid is commutated between the differentially rotating pump and motor.

Another improvement would be in the area of dynamic balancing. The difficulty of
25 balancing rotating equipment to preclude vibration induced by rotating eccentric masses becomes worse exponentially with increasing speed of rotation.

Yet another improvement would be in reducing the losses caused by "windage" and fluid drag associated with the rotating elements inside the transmission housing. In applications having a prime mover with a high rotating speed, such as an electric motor,
30 turbine engine or high performance spark ignition gasoline engine, the input elements would rotate at the prime mover output speed unless a gear reduction unit were interposed between the prime mover and the transmission. Gear reduction units add undesirable cost and weight. The windage and fluid drag losses can be greatly reduced by reducing the speed of rotation of those rotating elements.

35 Still another desirable improvement would be in the area of manufacturability, simplicity, and cost. Prior art continuously variable hydromechanical transmissions have

tended to be excessively complicated and costly to build. It would be a welcome development to original equipment manufacturers to have a continuously variable hydromechanical transmission available that is efficient, small and light weight, and is easily and economically manufactured and maintained.

5 One approach for achieving these improvements is shown in an international patent application No. PCT/US98/24053 filed on November 12, 1998 by Folsom and Tucker entitled "Hydraulic Machine", now issued as U.S. Patent No. 6,358,174. A variation of this approach in a tandem hydromechanical transmission using low cost conventional components is shown in International Patent Application No. PCT/US99/28,083, now
10 issued as U.S. Patent No. 6,530,855 issued March 11, 2003, and pending in Application No. 10/386,874 entitled "Parallel Hydromechanical Underdrive Transmission" provides improvements that this technology available for smaller vehicles requiring more compactness and lower cost, such as outboard motors for boats, motor scooters, motor cycles, RV's and snowmobiles and small cars. The greatest benefit for use of a high
15 efficiency continuously variable power transmission would be for automobiles and light trucks, as well as other larger vehicles, particularly those in which a transaxle with an underdrive final ratio is desirable. This class of vehicles consumes by far the most gasoline and diesel fuel and where significant improvements in fuel economy could benefit vehicle owners and make a profound contribution to the betterment of the planet.

20

Summary of the Invention

A hydromechanical continuously variable power transmission for converting rotating mechanical power at one combination of rotational velocity and torque to another combination of rotational velocity and torque over a continuous range, includes a
25 hydraulic pump, operatively driven by an input shaft, and a hydraulic motor operatively driving a differential that operatively drives a left and right output shaft. The hydraulic pump and hydraulic motor are coupled together mechanically through a planet set, and are coupled together hydraulically through a manifold, such that hydraulic fluid pressurized by said pump drives the motor, and spent fluid from the motor is cycled back
30 to the pump where it is re-pressurized.

The planet set is arranged co-axially with the input shaft, and the output shaft is offset from the input shaft, and the hydraulic pump and hydraulic motor are arranged in series with each other on opposite sides of the manifold, and parallel to the input and output shafts, thereby optimizing the use of space and keeping the overall length of the
35 transmission to a minimum, and minimizing required lengths of said input and output shafts.

Description of the Drawings

The invention, and its many attendant features and benefits, will become better understood upon reading the following description of the preferred embodiment, in conjunction with the following drawings, wherein:

5

Fig. 1 is a schematic diagram of a continuously variable hydromechanical power transmission in accordance with this invention;

Fig. 1A is a perspective view of a continuously variable hydromechanical power transmission embodying elements of the schematic diagram of Fig. 1;

10 Fig. 1B is a plan view sketch of a front wheel vehicle having a transverse engine and a transaxle transmission in accordance with this invention;

Fig. 2 is perspective view of the transmission shown in Fig. 1A, from the opposite side;

15 Fig. 2A is a sectional elevation from the front of the vehicle shown in Fig. 1B showing the transmission shown in Fig. 2 attached to the vehicle engine;

Fig. 3 is an elevation from the front end of the transmission, with the front housing removed;

Fig. 4 is an end elevation from the front of the transmission shown in Fig. 2 with the front housing attached and showing section lines for some of the following drawings;

20 Figs. 4A and 4B are perspective views of the middle housing of the transmission shown in Fig. 2, showing the attachment of the manifold in the housing;

Figs. 5 and 6 are sectional views along lines A-A and B-B, respectively, in Fig. 4;

Fig. 6A is a laid-out view of the gearing and hydrostatic units in the transmission shown in Figs. 5 and 6;

25 Figs. 7, 8 and 9 are sectional views of the transmission shown in Fig. 2 along lines C-C, E-E, and F-F, respectively, in Fig. 4;

Figs. 10 and 11 are bottom plan views of the hydrostatic unit assembly of the transmission shown in Fig. 2;

30 Fig. 11A is a perspective view of one of the links on which the yokes are pivotally mounted;

Figs. 12 and 12A are perspective views from the bottom of the hydrostatic unit assembly and the displacement control mechanism of the transmission shown in Fig. 2 at two different displacement settings of the units;

35 Fig. 13 is an end elevation from the rear of the pump hydrostatic unit and the displacement control mechanism shown in Fig. 12;

Figs. 14 and 15 are plan views from the top of the hydrostatic unit assembly and displacement control mechanism in two different positions of the units

Figs. 16 and 17 are sectional views along lines V-V and X-X, respectively, in Fig. 4;

5 Figs. 18 and 18A are perspective and plan view, respectively, of one of the hydrostatic units of the assembly shown in Fig. 12;

Figs. 19A-B are perspective views of the pump torque plate;

Figs. 19C-D are elevations of the socket side face and the manifold side face, respectively, of the torque plate shown in Figs. 19A and B;

Fig. 19E is a sectional elevation along lines 19E-19E;

10 Fig. 20 is an enlarged sectional elevation of the torque plate socket shown in Fig. 19E; and

Fig. 21 is a sectional elevation of a dump valve.

Description of the Preferred Embodiments

15 Turning now to the drawings, wherein like reference numerals designate identical or corresponding views, and more particularly to Fig. 1 thereof, a hydromechanical continuously variable transmission is shown schematically. One embodiment of the transmission shown in Fig. 1 is shown at 20 in Figs. 1A-9. It will be understood that the design shown in Fig. 1 and the physical device shown in Figs. 1A-9 are separate
20 illustrative embodiments of the invention, and that other embodiments within the scope of the invention can and will occur to those skilled in the art in light of this description.

The transmission 20 has a housing 25, shown in Figs. 1A, 2 and 2A, including a bell housing 25A at the front of the unit 20, shown in detail in Figs. 29, and a main housing 25B in the middle, shown in detail in Figs. 30, and a rear housing 25C at the rear of the
25 unit 20. Note that the relative terms "front, middle, and rear" are arbitrary since the unit sits cross-wise in the engine compartment of a front wheel drive vehicle 53 with a transverse engine, as indicated in the sketch of Fig. 1B. The housings 25A-C are connected together by bolts 27 along mating flanges at their adjacent edges and together form a fluid-tight housing 25 for the transmission 20. The bell housing 25A has a front
30 flange 28 that is configured to mate with and be bolted to a transverse mounted engine 55 for the front wheel drive vehicle 53, with the crank shaft or output drive shaft 31 of the engine, shown in Fig. 2A, coupled to an input shaft 29 of the transmission by a spline 26, discussed in more detail below, although the transmission 20 is suitable for use in numerous other vehicle and engine configurations, including four-wheel drive vehicles.

35 As shown in Figs. 1, 5, 6 and 6A, the transmission 20 includes an input hydrostatic unit or pump 30 at the rear of the unit 20, and an output hydrostatic unit or motor 35 at

the front of the unit 20. The hydrostatic units 30 and 35 are similar to the hydrostatic unit shown in Patent No. 6,874,994, entitled "Hydraulic Pump and Motor", with some differences noted below. The hydraulic pump and hydraulic motor 30 and 35 are coupled together mechanically through a gear set 40, shown in Figs. 1 and 6A, including a
5 double-sun planet set 45, pump and motor drive gears 41 and 74, and an idler shaft 73, and are coupled together hydraulically through flow passages 42 and 43 directly through a manifold 50, shown in Figs. 5-6A, 17 and 38, such that hydraulic fluid pressurized by the pump 30 drives the motor 35, and spent fluid from the motor 35 is cycled back to the pump 30 where it is re-pressurized. The manifold block 50 is supported in the main
10 housing 25B and attached thereto with a bolts 32 at each corner, as shown in Figs. 4A and 4B.

The transmission is shown in Fig. 1 at neutral: the pump 30 is at zero displacement, and the motor 35 is at maximum displacement. Both hydrostatic units can be controlled either together or independently, depending upon the application. However, in the
15 preferred embodiment, the hydrostatic units 30 and 35 are controlled together via a mechanical linkage mechanism 37, shown in Figs. 12-15 and described in detail below.

As shown in Figs. 1 and 5-7, the pump 30 is operatively driven by the input shaft 29, acting through the gear set 40. The input shaft drives an input sun gear 48, which is in geared engagement with the planet gears 63 mounted in the planet carrier 47. The
20 pump drive gear 41 is geared to the planet carrier 47 of the epicyclic gear set 45. The input shaft 29 is driven by a prime mover, such as a vehicle engine 55, by way of a coupling 54, shown in Fig. 2A. The coupling 54 in this CVT application is a torsion damper bolted to the flywheel that dampens the engine torque spikes by allowing the input shaft 29 to rotate a few degrees relative to the crankshaft into which the end of the
25 input shaft 29 protrudes. This minor relative motion between the input shaft and crankshaft is accommodated by a needle bearing 57 between the end of the input shaft 29 and a socket in the end of the crank shaft into which the CVT input shaft 29 protrudes. This coupling 54 is just one way of coupling the transmission to the engine. Numerous other coupling structures will occur to those skilled in the art in view of this disclosure.

30 A makeup pump 56 is housed in the rear housing 25C, as shown in Figs. 2A and 5. The end of the input shaft 29 remote from the input end is splined to and drives a drive shaft 58 of the make-up pump. The separate make-up pump drive shaft is used for two reasons a) it keeps the overall length of the input shaft shorter, making it easier to manufacture, and b) having a separate splined shaft to drive the make-up pump offers
35 some compliance, and since the make-up pump 56 is located at one end of the CVT and the input end of the input shaft 29 is located at the other end, there could be some

eccentricity between the two; this separate shaft 58 and the splined connection to the input shaft 29 accommodates this eccentricity. The planet carrier 47 is concentric with but slightly spaced radially from the make-up pump drive shaft 58, and is splined to the pump drive gear 41 at 59, as shown in Fig. 6A (the make-up pump drive shaft 58 is not shown in Fig. 6A, but the spline at the end on the input shaft 29 is shown in Fig. 6A).

The double-sun planet set 45 is housed in the main housing 25B, which is fastened to the bell housing 25A with bolts 27. The double sun planet set 45 has an input sun gear 48 that is driven by the input shaft 29 by way of a spline on the input shaft 29 engaged with a spline in the bore of the input sun gear 48.

Input power from the engine 55 acting through the input shaft 29 drives the input sun gear 48 of the double sun planet set 45. The planet carrier 47 carries a plurality of double planet gears 63, which are engaged with both the input sun gear 48 and an output sun gear 65 of the double sun planet set 45, as shown in Figs. 1 and 6A. The output sun gear 65 is connected drivingly by mating splines to a sun gear shaft 51 that is disposed concentrically around the input shaft 20 and is supported on bearings 68, 69 fixed in a bearing housing 70 that is mounted in the bell housing 25A. The sun gear shaft 51 has an integral sun shaft gear 66 that is in geared engagement with a gear 72 on an idler shaft 73 and delivers reaction torque from the double sun planet set to the idler shaft 73 via the sun shaft gear 66. The idler shaft 73 is mounted in bearings 78 and 77 mounted in the housings 25A and 25B, respectively, as shown in Fig. 6. The gear 72 is also in geared engagement with and drives the differential gear 75, which in turn drives a bevel differential 60 via the differential gear. The other end of the idler shaft 73 has a gear 74 that is engaged with a motor gear 80 around the torque plate 82 of the motor hydrostatic unit 35. The bevel differential has two output bevel gears 87 and 88, two drive bevel gears 90, 91, and two output axles 81, 83 in this transaxle transmission embodiment.

The double sun planet set planet carrier 47 is connected drivingly to a pump drive gear 41 via a spline 59, and the pump drive gear 41 is supported for rotation on bearings 93, 94 located in the main housing 25B, as shown in Figs. 5 and 6A. The pump drive gear 41 drives the pump hydrostatic unit 30 via gear 84 that is drivingly connected to the pump hydrostatic unit torque plate 85 on a spline 86 (shown in Figs. 19A and B) so that reaction torque from the double sun planet set planet carrier 47 drives the pump hydrostatic unit 30.

The motor torque plate 82 is connected drivingly to the motor torque plate gear 80 via a spline; the motor torque plate gear 80 meshes with motor drive gear 74, which is connected drivingly to the idler shaft gear 72 via a spline, so that torque generated from

the motor hydrostatic unit 35 is transmitted to the differential gear 75 (and hence the transmission output) via the motor drive gear 80 and idler shaft gears 72, 74.

The pump torque plate 85, shown in detail in Figs. 17, 18, 18A, and 19A-E is supported for rotation about a longitudinal axis 112 through the manifold block 50 on
5 needle bearings 115 running inside a bearing race 116 and running around a supporting shaft 114 projecting through the manifold 50 coaxial with the axis 112. The torque plate 85 serves as a commutating fluid flow interface between spherical heads 120 of pistons 125 in bores 130 of a cylinder block 135, and the face of the manifold block 50, as well as the means for transmitting mechanical power to and from the hydrostatic unit. The
10 orientation of the hydrostatic unit in Figs. 18 and 18A corresponds to the orientation of the motor 30 in Fig. 6A, but in fact both the pump and motor hydrostatic units 30 and 35 are identical, so for purposes of this description, only one hydrostatic unit will be described, with the understanding that this description applies to both hydrostatic units 30 and 35.

15 Pressure in the cylinders acting on annular areas 121, 122 in bores 118 of the hollow pistons 125 places an axial force on the pistons, pushing the spherical piston heads 120 into torque plate sockets 126, shown in Figs. 17, 18 and 18A. To react the load and reduce wear on the piston heads 120 and the torque plate sockets 126, an underbalance hydrostatic bearing between the spherical piston head 120 and the torque plate socket
20 126 is used. The hydrostatic bearing that is in the torque plate socket 126 is comprised of an internal annular spherical area 127 that is subjected to full pressure from the respective cylinder bore. The bottom of this area 127 terminates in a blind hole 128 that communicates with a kidney slot 123 of that socket on the manifold-side face 129 of the torque plate 85.

25 The separating forces from these two annular areas are such that there is enough clamping force to hold the piston head 120 seated in the torque plate socket 126 to seal working fluid from escaping past this interface while keeping the contact force low enough so as to avoid appreciable wear at this interface. The spherical sockets 126 have a cylindrical section 119 at the opening of the sockets 126 before the spherical
30 section that is close fitting to the outside diameter of the piston head ball 120 to reduce leakage past the piston ball if it were to become unseated from the socket.

To ensure that full pressure from the respective cylinder bore 130 acts over the internal spherical area 127, a small annular groove 131 is made into the torque plate socket 126, as shown in Fig. 20. The respective kidney slot 123, opening in the bottom
35 of the torque plate socket 126, communicates with the blind hole 128, which breaks into this annular groove 131 at 132, so that any pressure that exists in the torque plate kidney

slot 123 is communicated to this groove 131. The internal spherical area 127 is now subjected to full pressure from both outside by this groove 131 and from inside through the blind hole 128 that communicates with the kidney slot 123.

To change the balance of this hydrostatic bearing, the diameter of the piston spherical head 120 can be increased or decreased and/or the position (and hence diameter) of the annular groove 131 can be changed.

As the annular groove 131 is subjected to full cylinder pressure, a small hole 124 is used to feed pressure from this groove 131 to an orifice 129 that is inserted in an enlarged portion of the hole 124 and is used to feed overbalance grooves on the manifold-side face of the torque plate 85.

There are counterbores 133 around sockets 126 in the torque plates for receiving piston retainer rings 134. The piston retainer rings 134 have a spherical section 136 on one side inside the rings that rides against the spherical surface of the spherical piston heads 120. On the other side of the inside of the piston retainer rings 134, there is a chamfered section 137 that allows the piston 125 to articulate thru its maximum range of motion. The rings have a small split 138 in them to allow them to be installed over the piston spheres 120. The rings are held in the counterbores of the torque plate 85 by a retainer plate 139, shown in Figs. 17 and 18A, that axially locates the retainer rings such that the piston retainer rings accurately locate and axially secure the pistons in their sockets in the torque plates. The retainer plate 139 is fastened to the torque plate 85 with fasteners 141. Although the piston retaining rings 134 are shown as individual pieces, the rings may be cast or molded as a one piece unit with each ring connected to each other by a small flexible arm that will allow a small amount of radial motion between each ring so as to allow for positional tolerances between the piston retainer rings and their respective counterbores 133 in the torque plates 85. An annular groove in the face of the torque plate 85 may be made to accommodate the arms of the one piece retainer ring.

If the piston retainer ring 134 is manufactured from a compliant material such as a plastic etc, it is possible to have a small amount of interference between the piston sphere and piston retainer ring when installed. This will ensure that the piston sphere remains seated firmly against the spherical socket 127 in the torque plate 85, thereby reducing leakage at this interface.

The retainer plate 139 that locates the piston retainer rings 134 is located radially by the needle roller bearing race 116 of the needle bearing 115 that locates the torque plate 85 on the manifold shaft 114, thereby ensuring the retainer plate rotates about the same axis 112 as the torque plate. The retainer plate 139 has a spherical protrusion 142

coaxial with the axis of rotation 112; the axial center of the protrusion 142 is located at the axial centerline 112 of the spherical sockets in the torque plate 85.

As shown best in Figs. 17 and 18A, a center piston 145 is fitted into a central stepped bore 144 in the cylinder block 135 to radially locate the cylinder block 135 on the center piston 145 via a precision fit. The center piston 145 is supported for rotation on a bearing 140 located in the bore of a thrust plate 146 that is located in the base 148 of a supporting yoke 150 on one end and supported by a spherical socket 147 that is formed into the center piston 145 and rides on the spherical protrusion 142 that is formed on the retainer plate 139.

As shown in Figs. 11B, 11C and 18, the yokes 150 each have arms 155 and 155' that are mounted for swiveling about two parallel lateral pivotal axes in bearings on pins 160 mounted in links 165 attached to both lateral sides of the manifold block 50. The arms 155' are thickened to provide a mounting pad for control arms 303 and 306 of a displacement control device, to be disclosed below. A link 165 is shown in detail in Fig. 11A and the attachment of the bottom link is shown in Figs. 10 and 11, and again in Figs. 12 and 13.

The cylinder block cylinders 130 are through bores; the piston heads 120 of the piston 125 protrude from inwardly facing open ends of the bores 130 and seat in the torque plate sockets 126. Pucks 170 seal the opposite ends of the cylinders 130 and ride against the thrust plate 146 that is seated in the base 148 of a supporting yoke 150. The pucks 170 each have a back side with a shallow recess surrounded by a peripheral land. A central restricted fluid orifice 175 communicates through the pucks 170 between the cylinders 130 and the recess to allow a low volume flow of fluid pressurized in the cylinders 130 into the region on the back side of the pucks 170 to create a fluid cushion, acting as a hydrostatic bearing, to lubricate and support the cylinder block 135 as it rotates against the inner face 182 of the yoke thrust plate 146, as explained in more detail below.

As shown in Fig. 18, the yokes 150 each have arms 155 that are mounted for swiveling about two parallel lateral pivotal axes in bearings on pins 160 mounted in links 165 attached to both lateral sides of the manifold block 50. A link 165 is shown in detail in Fig. 11A and the attachment of the bottom link is shown in Figs. 10 and 11, and again in Figs. 12 and 13.

The working pressure of the hydraulic fluid inside the cylinders 130 acting on the area at the bottom of the bore creates an axial load. This axial load acts in the opposing direction to that of the axial load created by the torque plates 82, 85. This load is then reacted by the yokes 150 via the thrust plates 146. To support the axial load between the

cylinder blocks and the yokes, whilst allowing for the rotational speed between the cylinder blocks 135 and the yokes 150, the hydrostatic bearing between the outside face of the puck 170 and the thrust plate 147 is preferred. It is of course possible to use rolling element bearings, but their size and life ratings make them less desirable in this application. The shallow recess and peripheral land on the outside face of the pucks 170 produce an active area and a sealing land. The active area is designed such that, when oil from the cylinder bore flows to this area via the restricted orifice 175, the pressure of this oil acting over the active area within the land will place the puck in balance with the axial load placed upon it. This balance can be less than, equal to, or more than 100% depending on the geometry of the features used and the size of the passage that allows oil to flow from the piston bore. If the balance is less than 100% (i.e. underbalanced) then there will be a resultant axial load that will force the puck in direct mechanical contact with the thrust plate. If the balance is more than 100% (i.e. overbalance) there will be a force that will try to separate the puck from the thrust plate, in this case the lubrication hole will be sized such that oil leaking past the separated puck will cause a pressure drop as it flows through the lubrication hole, therefore reducing the separating force until the puck comes to a equilibrium state. In this state the puck will be floating on a thin film of oil, whose thickness is determined by the leakage rate of the oil, this leakage rate being determined by the pressure drop of the leaking oil flowing thru the small lubrication hole (orifice). Therefore, by changing the diameter of the orifice 175, it is possible to vary the film thickness and the leakage rate. One benefit of using the overbalance design is that the puck will "float" on a film of oil and will have little or no metal-to-metal contact; this will reduce the wear at this interface and result in higher allowable rotational speeds. The orifice 175 will need to be sized such that there will be no failure of this bearing under the harshest of operating conditions whilst keeping the leakage rate to a minimum.

A compression spring 180 is positioned in the center bore of the cylinder block, surrounding the center piston 145. The compression spring 180 places an axial force on the shoulder of the center piston and on the bottom of the bore in the center of the cylinder block. The axial spring force has the effect of pushing the cylinder block 135 towards the thrust plate face and away from the torque plate 85 as well as pushing the center piston 145 against the protrusion 142 on the retainer plate 139, applying a force that pushes the torque plate 85 away from the thrust plate 146 towards the manifold 50. This applies a pre-load on the torque plate 85 clamping it against the manifold 50 to give an initial seal between the torque plate 85 and manifold 50 before hydraulic pressure can act on this face to seal as described in detail later on. The pucks 170 have individual

puck springs 185 placed between them and the cylinder blocks 135 that react the load from the center compression spring 180 placing an axial load on the pucks 170 so that they are held firmly against the face 182 of the thrust plate until hydraulic pressure can properly balance the forces placed upon them. By using individual springs 185 around
5 each puck 170, it is possible for the center compression spring 180 to place a near constant load on each puck 170 whilst allowing for manufacturing tolerances of each puck thickness and accommodating any movement of each puck 170 due to deflections under load. Although individual springs are shown, it is possible to employ a one piece part that will offer the same advantages as the individual springs at reduced cost.

10 By using individual pucks 170 as opposed to one large hydrostatic bearing plate the hydrostatic bearing is more compliant to deflections and out-of-flat running surfaces. This is because the individual puck can pivot slightly so that it can follow the form of its running surface. Any deviations in flatness acts over the circumference of the relatively small diameter of the puck 170. If the hydrostatic bearing were formed as one large
15 component (such as if it were formed directly on the back of the cylinder blocks) even if it were allowed to pivot so that it could follow the form of its running surface, any deviations in flatness would be acting over the circumference of a much larger diameter and hence would have a greater effect on the bearing. This larger hydrostatic bearing would then require stiffer (and hence larger and heavier) running surfaces so as to keep the leakage
20 and performance of the bearing at the same level as that of the individual puck type hydrostatic bearings.

The pucks 170 have a protrusion on the opposite end from the hydrostatic bearing face to balance the puck 170, so that the axial position of the center of gravity of the puck 170 coincides with the supporting cylindrical diameter of the puck that contacts the
25 cylinder bore. This eliminates any tilting forces that will arise from centrifugal forces that will tend to cause the puck to tilt and ride on an edge of the sealing land, if the axial center of gravity is not in line with the supporting section of the puck.

The pistons 125 are used to drive the cylinder block in synchronous rotation with the torque plate 85. This is done by means of the tapered outside diameter of the piston 125
30 running against the cylinder bore 130. The angle of this taper is made large enough to allow for the piston to articulate freely as the cylinder block articulates about the pivot axis, as well as to allow for positional mis-alignment of the cylinder block rotating and pivotal axis relative to the rotational axis of the torque plate 85. However the taper on the piston also allows the cylinder block to 'lag' the torque plate in rotation by a few degrees,
35 and this places an opposing torque on the cylinder block 135 from the torque plate 85. This opposing torque will add to the bearing drag torque on the cylinder block 135,

increasing the side load on the piston taper, which will result in increased wear at this interface as well as reduced torque efficiency. It is therefore desirable to reduce the angle of the taper to reduce the lag angle as much as possible, which necessitates that the centerline and pivotal axis of the cylinder block 135 be accurately located to intersect the rotational axis 112 of the torque plate 85. In order to accurately locate the rotational and pivotal axis of the cylinder block 135 relative to the torque plate rotational axis 112, the cylinder block 135 is provided with the central bore 144 in which the center piston 145 is located with a precision fit, and the center piston 145 is accurately located between the bore of the thrust plate 146 and the retainer plate axis 112 as previously described. The torque plate 85 is supported for rotation against the face of the manifold, and against radial forces acting on it, by the radial bearing 115 mounted in a bearing recess in a central bore 222 through the torque plate 85. The bearing 115 supports the torque plate 85 on the outside of the protruding end of the support shaft 114. This ensures that, when the yoke 155 pivots in its bearings on the pins 160, the centerline of the cylinder block 135 will articulate about the ball and socket centerline and therefore accurately locate the cylinder block 135 to the torque plate 85 regardless of in which direction the cylinder block 135 is articulated.

The controlled hydrostatic bearing provided on the manifold-side face of the torque plates 82, 85 is shown in Figs. 18 and 19a, B and D. The manifold-side face of the torque plates 82, 85 is the face of the torque plates 82, 85 that is in fluid engagement with the face of the manifold block 50. This hydrostatic bearing provides a fluid interface between the rotating torque plates 82, 85 with the stationary manifold face, allowing the torque plate to run freely against the face of the manifold block 50 while minimizing fluid leakage out of the interface and transferring fluid at high pressure from the pump 30 through the manifold 50 to the motor 35, and spent fluid back from the motor 35 to the pump 30. The hydrostatic bearing has an overbalance hydrostatic bearing in the form of shallow individual wedge recesses 255 radially inside an underbalance hydrostatic bearing in the form of the kidney-shaped ports 123 which communicate full fluid pressure through the torque plate 85 from the piston head sockets 127 on the other side of the torque plate 85. The wedge recesses 255 are defined by surrounding land frames 265 which in turn are delineated by a shallow annular groove 270 having holes 275 that communicate with the piston-side face of the torque plates 82, 85. An orifice 129 in the hole 124 extending from the center of each wedge recess 255 through to the rear side of the torque plate communicates with the spherical sockets 126 in which the piston heads 120 are seated to supply fluid under system pressure to the wedge recesses 255 to provide the fluid pressure to support the torque plates 82, 85 on a fluid cushion on the

manifold faces. The excess load carrying capacity of the controlled hydrostatic bearing separates the torque plates 82, 85 from the manifold faces to the extent that leakage flow around the land frames 265 into the groove 270 exceeds the flow capacity through the orifices 280 and creates a fluid pressure drop across the orifices between piston head
5 spherical sockets and the wedge recesses 255. This pressure drop reduces the axial force exerted by the controlled hydrostatic bearing until the axial spacing between the torque plates 82, 85 and the manifold face reaches an equilibrium where the axial force exerted by the two hydrostatic bearings just balances the axial force exerted by the pistons 125. The leakage from this hydrostatic bearing can be limited to an acceptable
10 rate by correct choice of the orifice diameter so that the desired balance of leakage through the bearing and reduced torque loss is achieved.

The configuration shown in Figs. 1 and 2 has been designed to optimize both the hydrostatic unit performance and the packaging requirements, to achieve a small lightweight transmission able to accommodate the highest power engines currently
15 available for front wheel drive passenger vehicle applications.

In order to achieve maximum hydrostatic unit efficiencies (and hence overall transmission efficiency) it is desirable to minimize the size of the rotating interfaces of the hydrostatic units as this will reduce both the torque loss and the leakage loss from the hydrostatic units as well as give size and weight advantages. To achieve this, it is helpful
20 to reduce the size of, or eliminate, any shafts that pass through the center of the hydrostatic units as this will help to minimize the piston bore circle and hence the torque plate land diameters. For the same reason, it also helps to reduce the number of pistons used for a given displacement. Obviously as the number of pistons used decreases, so the smoothness of operation of the hydrostatic units decreases, therefore a compromise
25 between acceptable operation and efficiency is sought. In this design seven pistons are used, as this has proven to offer a good compromise.

To reduce the rotating diameter of the hydrostatic units, the input and output shafts 29 and 81,83 that transmit power to and from the hydrostatic units 30, 35 have been located at a position transversely offset from the center of the torque plates 82, 85, unlike
30 the conventional bent axis design, and power to and from the hydrostatic units is transmitted via the outside diameter of the torque plates 82, 85 by means of a sprocket or gear. In the disclosed designs, a geared transfer has been used, although a silent chain sprocket could be used instead. The radial bearing 115 is placed in the center of the torque plate 110 for location as well as to support the radial load placed upon the
35 torque plate 110. This radial bearing 115 is supported by the shaft 114 that is secured in the manifold 50. The axial center of the radial bearing 115 and the torque plate gears 80,

84 is located coincident with the axial position of the center of the spherical piston heads 120 in torque plates so that there is no moment produced on the radial bearing 115 and torque plates 82, 85 from any radial loads placed upon it from either the hydrostatic unit pistons or the gears 80, 84. Taking power from the outside as opposed to the inside of the torque plate not only gives the advantage of being able to keep the hydrostatic unit torque plate size to a minimum, but by careful angular orientation of the line of force from the gear it is possible to use the radial and tangential force induced by the gear to reduce the radial force induced by the hydrostatic unit pistons. This will reduce the resultant radial forces that are transmitted to ground which will enable the use of a smaller radial bearing to support this load, as well as reduce noise transmitted to ground from the hydrostatic units. A combination hydrostatic bearing shown in Fig. 18 supports the axial load on the torque plate.

The device that is used to transmit power to and from the hydrostatic units 30, 35 via the torque plates 82, 85 is shown as a separate component from the torque plates. This enables the torque plates to be made from a different material from that of the torque transmission gears 82, 85, therefore making it possible to manufacture these components from their ideal material, taking into consideration performance, durability, manufacturability and cost etc. In the design shown in Figs 18 and 18A, the gear 84 is splined onto the torque plate 85 and held axially between the retainer plate 139 and a shoulder on the torque plate 85. The retainer plate 139 is held onto the torque plates via screws 141. It is also possible to have the gear 84 be directly formed to the outside of the torque plate if material selection allows.

In order to keep the transmission as small and light as possible, it is best to reduce the loading that all of the hydraulic components impart on their supporting structures, thereby reducing the required size and weight of these structures. By placing the hydrostatic units so that the torque plates 82, 85 face each other across the manifold block 50, in a series configuration, the large axial force from the torque plates 82, 85 cancel each other out and place the manifold block 50 mainly in compression. As the manifold block 50 is mainly under a compressive load, the manifold structure is inherently strong and stiff thereby reducing the size required to keep the manifold faces flat and deflection free, which affords the best performance of the combination hydrostatic bearing. The axial load placed upon the yokes 150 that support the hydrostatic units 30, 35 can be reacted from the pump yoke to the motor yoke by connecting the two yokes 150 together through the links 165 fastened to top and bottom sides of the manifold block 50. These links 165 are placed mainly in tension where they are inherently strong and stiff, thereby reducing the size of the structure taking this load. These links 165 are

rigidly connected to the manifold block 50, but the only loads that are placed upon the manifold block 50 from the links are due to the imbalance of axial forces when the pump and motor hydrostatic units 30, 35 are at different displacements, and the radial loads that are induced from the yokes 150 when the hydrostatic units are at angle other than zero degrees.

To further decrease the size of the hydrostatic units and reduce the size weight of the transmission, the hydrostatic unit displacement is reduced, so that under maximum transmission output torque conditions a maximum operating pressure of 5000 psi is reached.

The flow to and from the hydrostatic units is passed through the hollow pistons 125 and the torque plates 82, 85 to the manifold 50. An added benefit of placing the hydrostatic units in a series configuration is that the passages that carry the fluid in the manifold to and from the hydrostatic units can now be relatively short and straight, thereby minimizing the flow losses and fluid noise through the manifold and increasing transmission efficiency.

When locating the power shaft apart from the center of the hydrostatic units, the best location for the axis of the hydrostatic units is in parallel to the input/output axis as opposed to concentric to the input/output axis. To achieve the optimum packaging shape and size for the intended front wheel drive application, the series hydrostatic units have been placed above and to the right (when viewed from the input end) of the input/output centerline. This has the benefit of keeping the rotating elements of the pump/motor hydrostatic unit assembly above the level of the oil reservoir which greatly reduces the aeration of the reservoir oil. In order to separate the reservoir oil from the oil that is expelled from the rotating components from lubrication and HSU leakage, a separate oil containment volume is created within the main and bell housings, as best shown in Figs. 2A and 3. This volume is created from a pocket that is formed into both housings so that when they are bolted together this volume becomes sealed from the rest of the transmission internal volume except for strategically placed openings to allow oil to flow in and air to flow out. This effectively creates two separate oil volumes within the transmission housing – a sealed oil volume protected from the rotating components and an 'open' oil volume that is exposed to the rotating components. The open oil volume will be replenished from oil that leaks from the hydrostatic units and from the lubrication supply. Oil is taken by the makeup pump 56 from the sealed oil volume thru a suction filter to feed the hydrostatic units and lubrication circuit. The openings that connect the two oil volumes are positioned such that oil will flow into the sealed volume from the bottom of the open volume after the oil has been de-aerated. This will ensure that the oil

taken by the makeup pump will be de-aerated and therefore the oil supplied to the hydrostatic units to replenish oil lost thru leakage will also be de-aerated. There is a baffle 186 that partially surrounds the differential gear to minimize the churning effect that this large gear has on the oil in the open oil volume, in order to aid de-aeration of the oil in the open oil volume.

Make up pressure oil is fed to the manifold through a filter 187 in a fluid line 188 from a make up pump driven from the input shaft 26. Make up pressure is used to replenish system oil that leaks from the pump 30 and the motor 35 to the transmission sump via the various hydraulic interfaces, as well as to keep a positive pressure on the low pressure side of the flow passages to prevent cavitation, as well as to supply the lubrication circuit and oil cooler supply (if required). The makeup pressure is fed to the main flow passages 42, 43 in the manifold block 50 via the check valves 290 and 292 so that this oil will flow to the flow passage that is at the lower pressure. There are two ports in the manifold body that connect from the main high and low pressure flow passages inside the manifold to a dump valve body 293 that is connected to the manifold 50. The valve body 293, shown in detail in Fig. 21, that contains a spool valve that will either block one of these ports from the other or connect both ports to each other depending upon the spool position. If both ports are connected to each other then oil on the high pressure side of the manifold can free flow to the low pressure side of the manifold thereby limiting the pressure that can be produced by the pump 30 so as to provide a neutral on the transmission even if the pump is not at zero displacement. The spool itself can be actuated by a number of means - mechanical, electrical or hydraulic etc. In the embodiment shown the spool is spring loaded in one direction and hydraulically actuated in the other by directing fluid pressure to an operating area on the end of the spool which is at the opposite end to the spring. To activate the spool against its spring force, makeup pressure is tapped off from the make-up supply inside the manifold and fed to a standard electro hydraulic PWM valve 296. The PWM valve 296 is hydraulically connected to the spool valve 293 so that it will either block the make up pressure and vent the operating area of the spool to atmosphere or direct the makeup pressure to the operating area of the spool, depending upon an electrical signal fed to the PWM valve from the vehicle controller. The operating area and spring force are such that when make up pressure is fed to the operating area, the force generated is enough to overcome the spring force and move the spool valve overcoming any hydraulic forces acting on the spool valve as it actuates. This enables the vehicle controller to be able to activate the dump valve as and when required.

Control Operation:

The operation of the displacement controls for the hydrostatic units will now be described in connection with Figs. 12-15. These controls are in accordance with a control regime, as follows, that is envisioned to govern operation of transmissions in accordance with the invention:

To achieve the maximum output torque from the transmission, the motor is set at maximum displacement under maximum pressure to generate maximum hydraulic torque, whilst having maximum input torque reacted to the output via the planet set arrangement. To achieve this it is beneficial to have a control regime that will hold the motor at its maximum displacement as the pump is stroked from zero displacement until the pump reaches a displacement where it can generate maximum pressure whilst reacting maximum input torque. Once this position has been reached, the pump and motor can be stroked simultaneously (the motor at a slightly faster rate) so that the pump and motor reach their final displacements (pump at max disp, motor at zero disp) at the same time. The advantage of this control regime is that this will minimize the maximum flow rate in the transmission and hence reduce flow losses and noise generation.

Although it is possible to control the pump 30 and motor 35 individually or simultaneously by various methods, in the embodiment shown the pump 30 and motor 35 are controlled simultaneously via a mechanical linkage and bellcrank mechanism that is actuated by an electric control motor via a gear mesh. The electric motor is controlled via the transmission controller (not shown) that is connected electrically to an electrical control motor 300. The control mechanism, shown in Figs. 12-15, includes a motor control arm 303 and a pump control arm 306 rigidly connected to the arms 155' of the yokes 150 of the motor and pump hydrostatic units, respectively. Each of these arms 303 and 306 have a bore 310 and 312 at the operating end of arms to pivotally locate one end of a motor control link 314 and pump control link 316 respectively via pins. The other end of the pump and motor links 314 and 316 are pivotally connected to a bellcrank 320 that itself pivots about a shaft 322 that is rigidly located in the main housing. There is a gear form 325 on the outside periphery of the bellcrank that meshes with a control drive gear 327 that is supported for rotation in the main housing. The control drive gear 327 is drivingly connected to an electric control motor 300 that is actuated by the vehicle controller, so that when instructed by the controller, the control motor rotates the control drive gear which causes the bellcrank 320 to rotate about its pivot 322, causing the pump and motor control links 314, 316 to move so that the pump and motor yokes 150 articulate about their pivots 160 to change the displacements of the pump 30 and motor 35.

The relative position between the pivot points of the yokes, control arms, control links and bellcrank is such that when the control drive gear is rotated, the displacements on the pump and motor are changed in accordance with the desired displacement schedule for the particular transmission ratio. The relative position of the motor yoke, control arm, control link and bellcrank pivot points are such that when the control drive gear initially rotates (starting from 0 degrees) the motor displacement remains at or close to its maximum displacement and changes very little with respect to the control drive gear rotation. The relative position of the pump yoke, control arm, control link and bellcrank pivot points are such that when the control drive gear initially rotates (starting from 0 degrees) the pump displacement is at a negative displacement and changes rapidly with respect to the control drive gear rotation. This enables the controller to stroke the pump from a negative to positive displacement when the motor is held at, or close to its maximum displacement. This offers the preferential control regime as described above to enable maximum torque multiplication in both forward and reverse.

There are large control forces generated by the pump and motor hydrostatic units that try to pivot the hydrostatic units to a positive and negative angle. These forces are generated from the center of pressure of the hydrostatic unit's pistons under high pressure moving above and below the HSU pivotal axis causing a fluctuating, equal and opposite control force, and although the average control force is zero the instantaneous force can be very high. When the transmission is at its max torque multiplication in forward and reverse, the HSU pressure is at its highest, and therefore the instantaneous control forces are high; as the motor is at its slowest speed these instantaneous forces will be reacted back to the control linkage and will try to rotate control motor. However one advantage of employing the control mechanism arrangement as described above, is that due to the geometry of the pivot points, the motor has little mechanical advantage when at the max torque multiplication ratio, and therefore the high control forces from the motor HSU has little effect on the control motor. Although with this regime the pump HSU has a high mechanical advantage, the rotational speed of the pump is at its maximum in relation to engine speed when at this ratio, and the frequency of the high instantaneous control forces is too fast for these forces to get reacted back to the control motor. Hence it is possible to control the CVT with a relatively small electric motor.

With the linkage arrangement shown the mechanical advantage of the motor HSU increases as the motor speed increases and the mechanical advantage of the pump HSU decreases as the pump speed decreases, therefore the control forces that get reacted back to the control motor stay well within acceptable limits throughout the entire ratio range.

As the control motor rotation angle moves from 0 – 125 degrees the motor HSU remains at or close to it's maximum displacement as the pump HSU goes from a small negative to positive displacement so as to achieve maximum output torque in forward and reverse; then the motor HSU displacement starts to decrease as the pump HSU displacement starts to increase as the control motor angle goes from 125 – 240 degrees where the pump HSU reaches its maximum displacement as the motor HSU reaches zero displacement and therefore obtaining the optimum control regime as described previously. This arrangement offers a very simple and cost effective method of controlling both the pump and motor HSU with one small electric motor and simple easily manufactured components. Because the control forces seen by the electric motor are relatively small the control motor is able to change the ratio of the CVT with required speed and accuracy to offer extremely good vehicle control. In the embodiment shown an electronic rotary position sensor is connected to the pump to supply a signal of pump HSU displacement to the transmission controller. Because of the fixed (and calculable) ratio of pump to motor displacements it is possible for the transmission controller to know the theoretical transmission ratio by measuring just one of the HSU displacements, although the actual transmission ratio will vary from the theoretical ratio because of manufacturing tolerances, deflections and slippage, etc. The pump HSU has been chosen for displacement measurement because although the knowing the exact pump/motor relationship is not critical (because the controller will ultimately use input and output speed to determine transmission ratio) it is critical to know when the pump is at zero displacement so the controller can accurately command neutral when required, as well as offer good low speed vehicle controllability around neutral in both forward and reverse.

Operation The operation of the transmission shown in Fig. 1 will be described in its several drive modes, using a shorthand notation of the planet gearsets 40 and 45 to indicate the number of teeth on meshing gears. This notation is explained on the following list:

- Si – input sun gear 65
- So – output sun gear 70
- Pi – input planet gear
- Po – output planet gear
- Cp – planet set planet carrier 85
- Gp – pump drive gear 71
- Gm – motor drive gear
- GTP – pump torque plate gear 95

GTm – motor torque plate gear 105

Gi – idler shaft gear 105'

Go – output sun shaft gear

Gd – differential gear 225

5

Neutral:

When the transmission is at neutral and the vehicle is stationary, the differential, and hence the differential gear, idler shaft gear, motor drive gear, motor torque plate gear and the output sun shaft gear are all stationary. The dump valve is open. As the input shaft 29 and input sun gear 48 are rotating at input speed, the planet carrier 47 will rotate and hence cause the pump to rotate at the ratio of:

$$\text{Input Speed} / (1 - So/Po \times Pi/Si) \times (Gp/GTp) \quad eq1$$

If the pump and motor hydrostatic units 30, 35 are at some displacement, there will be some pumping from the pump 30, but as the dump valve is open any flow from the pump will short circuit from the high pressure side to the low pressure side and will therefore not be able to generate any appreciable pressure to generate any torque on the motor, hence it will not cause the output differential to rotate.

Whilst the transmission is at neutral and before the dump valve is closed, the transmission controller will ensure that the pump hydrostatic unit is at zero displacement and the motor hydrostatic unit is at maximum displacement. This will ensure that there will be no flow from the pump and hence no rotation from the motor. This can be achieved by several ways including (but not limited to): A speed sensor on the motor or the motor drive gear that will detect speed and rotation direction, and hence determining the pump HSU displacement; or, an angular position sensor on the pump hydrostatic unit – the design shown incorporates latter of these methods.

Low Ratio (high torque multiplication):

When the transmission controller is signaled to go to Drive mode the dump valve will be closed (once the controller has determined the pump HSU is at zero displacement).

Due to the planet set configuration, as described above, the input torque is split into two parallel paths, these being a direct mechanical path fed continually to the output shaft at the ratio of:

$$\text{Input torque} \times (Pi/Si) \times (So/Po) \times (Gd/Go) \quad eq2$$

and a 'hydraulic' torque path fed continually to the pump at the ratio of:

$$\text{Input torque} \times (1 - So/Po \times Pi/Si) \times (GTp/Gp) \quad eq3$$

When instructed by the transmission controller, the pump is stroked to give a small displacement. As the pump is rotating at the speed as described by eq1, pumping and

hence fluid flow will take place. This fluid flow passes directly thru' the manifold to drive the motor. As the dump valve is now closed, this motor rotation will cause the motor torque plate gear and hence the differential gear to rotate to give an output speed. Due to the fact that the pump is at a small displacement, a small amount of torque to the pump results in a high pressure and small flow rate, and as the motor is at a large displacement this high pressure and small flow rate results in a high output torque and low output speed at the differential gear in the ratio of:

$$\text{Motor Torque} \times (G_m/G_{Tm}) \times (G_d/G_i) \quad eq4$$

Due to the fact that the pump is at a small displacement it will react a torque from the input shaft as described in eq3. This torque on the input shaft will then be reacted directly to the differential gear shaft as described in eq2.

The high 'hydraulic' output torque as described by eq4 is added directly to the mechanical output torque as described by eq2. Therefore the total output torque can be expressed as:

$$[\text{Input torque} \times (P_i/S_i) \times (S_o/P_o) \times (G_d/G_o)] + [\text{Input torque} \times (1 - S_o/P_o \times P_i/S_i) \times (G_{Tp}/G_p) \times (G_m/G_{Tm}) \times (G_d/G_i) \times (\text{motor disp/pump disp})] \quad eq5$$

It can therefore be seen that there is a total output torque comprising of a fixed mechanical torque plus a variable hydraulic torque, and as the motor displacement to pump displacement ratio decreases, the amount of hydraulic torque decreases, and if the pump or motor displacement equals zero then there is no hydraulic torque, just mechanical torque.

As pump displacement increases, flow rate from the pump increases, and this increased flow causes the motor and hence the double sun planet output gear and differential gear to increase in speed. As the differential gear increases in speed relative to the input shaft speed, the output sun gear speed increases relative to the input sun gear speed; this causes the planet carrier speed to decrease. This has the effect of causing the pump speed to decrease as the transmission ratio increases and the differential gear increases relative to input speed. This has the effect of reducing the total system flow rate, (when compared to a conventional hydrostatic transmission of the same capacity), to approximately $\frac{1}{3}$ to $\frac{1}{4}$ depending on planet set ratios used. This reduces the flow losses and noise levels normally associated with pure hydrostatic machines.

With the mechanical control linkage, as the transmission approaches final ratio, the motor approaches zero displacement whilst the pump approaches maximum displacement. Therefore the pump speed will approach zero as motor speed approaches its maximum.

Final ratio:

When the motor reaches zero displacement it can no longer accept fluid flow, so the pump can no longer displace fluid and therefore stops rotating. This causes the pump and planet carrier to stop rotating. The pump is now acting as a reaction unit for the planet carrier and now all of the input torque is transferred thru' the double sun planet set to the differential thru the geartrain. Due to the ratio of the double sun planet set and the gear ratio from the output sun gear shaft gear to the differential gear the output speed is decreased and the output torque increased give a final underdrive ratio of:

$$\text{Input speed} \times (S_i/P_i) \times (P_o/S_o) \times (G_o/G_d) \quad \text{eq6}$$

As the pump has been stroked to its full displacement, hydraulic pressure required to react the input torque has been reduced to a minimum, thus reducing hydraulic leakage losses and hydraulic loading of bearings to a minimum.

As the motor is now at zero displacement it is no longer adding any torque to the output, and as stated above, the motor is now rotating at its maximum speed. This will create an efficiency loss, as the motor is now being 'driven' by the differential gear and will subtract torque from the transmission output.

As all of the input power is now transferred thru' the double sun planet set, and the hydraulics are acting only as a reaction unit to hold the planet carrier, the efficiency of the transmission will be high (90+%), as the only losses are the normal gear set losses (approx. 1%), slippage on the pump due to leakage and spin loss of the motor and makeup pump losses. To further increase the efficiency at this point, a brake could be applied to the pump. This will stop the pump HSU from rotating due to hydraulic leakage and hence reduce speed loss at the output shaft.

Reverse:

Starting from the same conditions as in neutral, with the motor at its maximum displacement and the pump at zero displacement: the controller will stroke the pump in the opposite direction (i.e. to a negative angle) causing fluid flow to go in the opposite direction. This will cause the motor and hence the differential to rotate in the reverse direction. Due to the planet set gear configuration, the mechanical torque (as described in eq2) is still acting in the forward direction. Therefore the total output torque, in reverse, can be expressed as:

$$[\text{Input torque} \times (P_i/S_i) \times (S_o/P_o) \times (G_d/G_o)] - [\text{Input torque} \times (1 - S_o/P_o \times P_i/S_i) \times (G_{Tp}/G_p) \times (G_m/G_{Tm}) \times (G_d/G_i) \times (\text{motor disp/pump disp})] \quad \text{eq7}$$

Maximum Torque Limitation:

Due to the fact the motor to pump displacement ratio can be infinitely large, at or around the neutral zone in forward and reverse, it is possible to generate infinitely high pressures and output torques. Obviously these have to be limited to reasonable values, as determined by the structural limitations of the transmission. This can be achieved in several ways, including but not limited to:

The use of a pressure relief valve that will limit the maximum pressure that the pump can generate, and hence the maximum output torque. Since the pump will be at relatively small displacements, the flow rate thru the relief valve will be at reasonable levels.

The system can be designed to be self regulating; by designing the pump and motor to have a leakage rate (which is necessary for hydrostatic bearing interface cooling and lubrication) which at a specified pressure is equal to the pump discharge at maximum input speed. This will prevent the pump from generating a higher pressure than this. The transmission will then reach a 'stall' torque.

With today's sophisticated electronic engine controls it is possible to limit engine torque when the pump is at small displacements so as to regulate the maximum system pressure.

There is a minimum pump angle at which the pump can react full input torque without exceeding the maximum allowable system pressure, and hence generate maximum output torque. At pump angles less than these, the output torque will not increase as the maximum pressure is limited by methods as described above, but the input to output speed ratio will continue to decrease and will approach infinity as the pump angle becomes infinitely small.

The hydromechanical continuously variable transmission disclosed in Fig. 2 has many advantages over existing transmission designs including:

Taking power to and from the hydrostatic units via the outside diameter of the torque plate enables the hydrostatic units rotating diameters to be kept as small as possible, as well as allowing for a small number of pistons to be used. This increases the hydrostatic units efficiency as both torque loss and leakage loss across the HSU increase as the rotating diameters increase.

Taking power to and from the hydrostatic units from the outside diameter of the torque plate also allows for the hydrostatic units to be placed in parallel to the input/output axis and facing each other in series. Placing the hydrostatic units in series enables the pump forces to counteract the motor forces, significantly reducing the resultant forces that are exerted to the supporting structure and transmission case. This

allows the transmission to be as small and light as possible whilst being able to handle high powers.

Using a gear (or chain) to drive the outside diameter of the torque plates enables the radial reaction load from the chain or gear to counteract the radial load from the pistons acting on the torque plate. This reduces the force on the bearing that radially locates the torque plate, therefore reducing the size of this bearing and its supporting structure.

By making the torque input/output members of the hydrostatic units separate from the torque plates allows for different materials to be used for the chain sprocket/gear and the torque plates. It is therefore possible to manufacture these components from their ideal material, taking in to consideration performance, durability, manufacturability and cost etc.

By connecting the pump and motor hydrostatic units in series as described above, the pump and motor rotation directions are such that both the high and low pressure flows are directly inline with each other between the pump and motor. This ensures that the flow passages are as short and as straight as possible, thereby reducing flow losses and maximizing hydraulic efficiency.

As the manifold and the hydraulic assembly are separate to the housings, all of the high pressure oil passages are contained within the hydraulic assembly and do not need to be passed into the transmission case. This reduces structural integrity requirements of the transmission case as well as reduces hydraulic noise transmitted to the housings.

By way of an overview of some of the salient features of the continuously variable transmissions disclosed herein, the following list is presented. Naturally, there will be many applications that do not need to use all of these features, or would use some modification of these features. Therefore, the inclusion of this list should not be interpreted as a limitation to the claims in which this invention is defined.

Hydrostatic Unit Related Features:

Torque plate has overbalance grooves inside of the main kidney slots

Torque plate has overbalance grooves outside of the main kidney slots

Hydrostatic units placed facing each other so that the axial force of the pump is counteracted by the axial force of the motor, placing the manifold in compression.

Individual pucks are used to support the axial load from the cylinder block to the cylinder block support structure

Individual pucks that have an overbalance hydrostatic bearing using an throttling device so that the leakage flow rate across the bearing causes a pressure drop in the recess area so that the separating force in the recess area is equal to the clamping force

from the pressure acting over the piston area, placing the puck in equilibrium or near-equilibrium.

Individual pucks that have an underbalance hydrostatic bearing where the separating area and force is less than that of the clamping area and force. The individual pucks are
5 balanced eliminating any tilting forces that will arise from centrifugal forces that will tend to cause the puck to tilt and ride on an edge of the sealing land, if the axial center of gravity is not in line with the supporting section of the puck. Power is transferred to and from the each hydrostatic unit by means of a gear or chain sprocket that is positioned around the outside of the torque plate.

10 The axial center of the gear (or chain sprocket) is positioned such that the centerline of the gear (or chain sprocket) is co-incident (or near co-incident) with the center of the piston spheres and the hydrostatic unit articulation center in the torque plate.

The gear mesh between the torque plate gear and its mating gear is orientated such that the radial force of the gear is used to counteract the radial force that is generated by
15 the pistons acting on the torque plate.

The torque plate is radially supported by a bearing that is positioned at the radial center of the torque plate.

The torque plate is radially supported by a bearing that is positioned such that the center of the bearing is co-incident (or near co-incident) with the center of the piston
20 spheres and the hydrostatic unit articulation center on the torque plate.

Geartrain Related Features

A double sun planetset is used to obtain a power split from the input power path so as to generate a parallel power path from the input to the output and to the pump hsu.

25 The power transferred to the pump is transferred to hydraulic power which is used to drive the motor which then transfers this hydraulic power to the output.

Controls Related Features

A simple mechanical link arrangement with one electric motor is used to control both
30 the pump and motor HSUs.

Obviously, numerous other modifications, combinations and variations of the preferred embodiments described above are possible and will become apparent to those skilled in the art in light of this specification. For example, many functions and
35 advantages are described for the three preferred embodiments, but in some uses of the invention, not all of these functions and advantages would be needed. Therefore, we

contemplate the use of the invention using fewer than the complete set of noted functions and advantages. Moreover, several species and embodiments of the invention are disclosed herein, but not all are specifically claimed, although all are covered by generic claims. Nevertheless, it is our intention that each and every one of these species and

5 embodiments, and the equivalents thereof, be encompassed and protected within the scope of the following claims, and no dedication to the public is intended by virtue of the lack of claims specific to any individual species. Accordingly, it is expressly intended that all these embodiments, species, modifications and variations, and the equivalents

10 thereof, are to be considered within the spirit and scope of the invention as defined in the following claims, wherein we claim:

1. A transaxle transmission for a front-wheel-drive vehicle, for converting rotating mechanical power at one combination of rotational velocity and torque at an input shaft to another combination of rotational velocity and torque at an output shaft over a continuous
5 range, comprising:

an input hydrostatic unit and an output hydrostatic unit hydraulically coupled together with fluid passages so that pressurized working fluid from said input hydrostatic unit drives said output hydrostatic unit, and spent fluid from said output hydrostatic unit recharges said input hydrostatic unit for being pressurized therein;

10 each of said hydrostatic units having cylinders in a cylinder block mounted for rotation about cylinder block rotation axes, and for pivoting about pivotal axes relative to a stationary manifold block through which said fluid passages extend;

said cylinders each having a piston therein, and each piston having a piston head thereon protruding from an open axial end of said cylinder on a first axial end of said
15 cylinder block;

said input and said output hydrostatic units include input and output torque plates, respectively, having sockets which receive said piston heads, each of said torque plates being supported between said cylinder blocks and input and output faces of said manifold block, respectively, to rotate with said cylinder blocks about a center of rotation of each
20 torque plate against said input face and said output face of said manifold, respectively, said torque plates each having kidney slots communicating with said sockets in said torque plates that convey fluid through said torque plates between said pistons and said fluid passages in said manifold;

an epicyclic gear train through which said hydrostatic units are mechanically
25 coupled, said epicyclic train including an epicyclic gear set that includes first and second planet gear sets coupled together such that said first planet gear set is coupled to said input hydrostatic unit, and said output hydrostatic unit is coupled to said second planet gear set and said output shaft, and said input shaft is drivingly coupled to said second planet gear set.

30

2. A transmission as defined in claim 1, wherein:

said cylinder blocks of said input and output hydrostatic units are supported for rotation on an input yoke and an output yoke, respectively;

said input yoke and said output yoke are mounted for pivoting about a pivotal axis
35 parallel to said input face and said output face, respectively, of said stationary manifold block.

3. A transmission as defined in claim 2, further comprising:
a displacement control device for adjusting and controlling displacement of said hydrostatic units, said displacement control device having a mechanical linkage and
5 bellcrank mechanism that is actuated by an electric control motor via a gear mesh.
4. A transmission as defined in claim 3, wherein:
both hydrostatic units are controlled simultaneously via said mechanical linkage and bellcrank mechanism that is actuated by said electric control motor via said gear mesh.
10
5. A transmission as defined in claim 3, wherein:
said mechanical linkage and bellcrank mechanism includes an input hydrostatic unit control arm and an output hydrostatic unit control arm rigidly connected to the yokes of said input and output hydrostatic units respectively, and links pivotally connected
15 between said control arms and a bellcrank, said bell crank being pivotally mounted on a fixed shaft;
said bell crank has an outside periphery, on which is formed a gear form that meshes with a control drive gear that is driven for rotation by said electric control motor for controlling said displacement of said hydrostatic units.
20
6. A transmission as defined in claim 1, wherein:
said epicyclic gear train includes a double sun planetset that affords a power split from said input shaft so as to generate a parallel power path from said input shaft to said output shaft and to said input hydrostatic unit.
25
7. A transmission as defined in claim 1, wherein:
said hydrostatic units are placed in a series configuration with said torque plates facing each other across the manifold block and aligned on a single axis of rotation, so that large axial forces from said torque plates cancel each other out, placing said
30 manifold block mainly in compression.
8. A transmission as defined in claim 7, wherein:
said single axis of rotation is parallel to a central axis of said epicyclic gear train.

9. A transmission as defined in claim 7, wherein:

said hydrostatic units place an axial load placed upon said yokes, and said axial load is reacted between said yokes through links that are rigidly connected to said manifold and connect said two yokes together.

5

10. A transmission as defined in claim 1, further comprising:

a housing having an upper chamber in which said hydrostatic units, said manifold, and said epicyclic gear train are supported;

10 a oil containment reservoir within said housing that is separated from said upper chamber, effectively creating two separate oil volumes within said housing – a separated oil volume protected from rotating components, and an 'open' oil volume that is exposed to said rotating components;

openings connecting said two oil volumes, positioned such that oil flows into said separated volume from said open oil volume;

15 a makeup pump for pumping oil from said separated oil volume to lubricate said hydrostatic units and said epicyclic gear train.

11. A transmission as defined in claim 2, wherein each of said hydrostatic units further comprises:

20 a center piston in a central bore of said cylinder block for radially supporting said cylinder block;

said center piston is supported in a bore of a thrust plate that is located, at one end of said center piston, in a base of said yoke, and is supported at an opposite end of said center piston, by engagement of a spherical socket in said other end of said center piston
25 on a spherical protrusion protruding from said torque plate.

12. A transmission as defined in claim 11, wherein each of said hydrostatic units further comprises:

30 a compression spring in said central bore of said cylinder block, surrounding said center piston and under compression to exert an axial force between said center piston and said cylinder block to preload said cylinder block on said torque plate and said thrust plate.

13. A transmission as defined in claim 11, wherein each of said hydrostatic units
35 further comprises:

a retainer plate fastened to said torque plate, said spherical protrusion being formed on said retainer plate, said retainer plate also having portions that retaining said piston heads in said sockets in said torque plate.

- 5 14. A transmission as defined in claim 11, wherein each of said hydrostatic units further comprises:

a split retainer ring in counterbores in each torque plate socket for holding said piston head in said sockets, said retainer rings being held in said counterbores in the torque plates

10

- 15 15. A transmission as defined in claim 11, wherein each of said hydrostatic units further comprises:

a bearing puck fitted into and sealing an open end of each of said cylinder block cylinders opposite to ends from which said piston heads protrude, said pucks each

- 20 16. having a restricted opening communicating between said cylinders to a backside of said puck;

a shallow recess surrounded by a peripheral land in said backside of said pucks is pressurized with hydraulic fluid from said cylinders through said restricted opening to act as a fluid bearing for said pucks in running against an inner face of said yoke thrust plate mounted in said yoke to provide thrust forces to counterbalance thrust forces exerted by hydraulic pressure in said cylinder.

25

17. A transmission as defined in claim 11, wherein each of said hydrostatic units further comprises:

- 25 individual springs placed between said pucks and said cylinder blocks for partially reacting load from said center compression spring placing on said puck.

17. A transmission as defined in claim 11, wherein each of said hydrostatic units further comprises:

- 30 said pucks have a protrusion extending into said piston bores to center and balance said puck, so that the axial position of the center of gravity of said puck coincides with the supporting cylindrical diameter of the puck that contacts the cylinder bore, minimizing any tilting forces that arise from centrifugal forces on said puck.

- 35 18. A transmission as defined in claim 11, wherein:

power is taken to and from said input hydrostatic unit via a separate gear or chain sprocket that is drivingly connected on said torque plate;

said separate torque plate gear or chain sprocket is held axially on said torque plate by said retainer plate.

19. A transmission as defined in claim 1, wherein:

5 said sockets which receive said piston heads are equally spaced around a socket circle on said input and output torque plates, respectively;

said gear or chain sprocket on each of said hydrostatic units has an axial center that is positioned such that the centerline of the gear or chain sprocket is substantially co-incident with the center of said socket circle and an articulation center of said hydrostatic
10 unit in the torque plate, and said torque plate is radially supported by a torque plate bearing that is positioned such that the center of said torque plate bearing is substantially co-incident with the center of said socket circle and said hydrostatic unit articulation center on said torque plate.

15 20. A transmission as defined in claim 1, wherein:

said fluid passages in said manifold block include main high and low pressure flow passages;

said manifold block has two ports that connect from said main high and low pressure flow passages inside said manifold block to a valve that is operable to either block one of
20 said ports from the other port or connect both ports to each other, depending upon the valve position, such that when both ports are connected to each other then oil on the high pressure side of the manifold flows freely to said low pressure flow passage, thereby limiting the pressure that can be produced by said input hydrostatic unit, so as to provide a neutral on said transmission even if said input hydrostatic unit is at a non-zero
25 displacement.

21. A transmission as defined in claim 20, wherein:

said valve is a spool type valve that is spring loaded in one direction and hydraulically actuated in the other direction and is operated by pressure fed from a make-up supply
30 provided inside the manifold by a make-up pump via an electro hydraulic valve.

22. A method of controlling the transmission ratio of a continuously variable transmission over a continuous range, comprising:

driving a control drive gear with an electric control motor to rotate said control drive
35 gear;

driving a bellcrank with said control drive gear to rotate about a bellcrank pivot, thereby moving an input hydrostatic unit control link and an output hydrostatic unit control link, which links are connected to yokes, on which cylinder blocks of said hydrostatic units are mounted for rotation, to articulate about yoke pivots to change
5 displacements of said hydrostatic units.

23. A method as defined in claim 22, wherein:

when said control drive gear is rotated, said displacements of said hydrostatic units are also changed;

10 when said control drive gear initially rotates, said output hydrostatic unit displacement remains at or close to its maximum displacement and changes very little during initial control drive gear rotation;

when said control drive gear initially rotates said input hydrostatic unit displacement is at a negative displacement and changes rapidly with respect to said control drive gear
15 rotation into positive displacement.

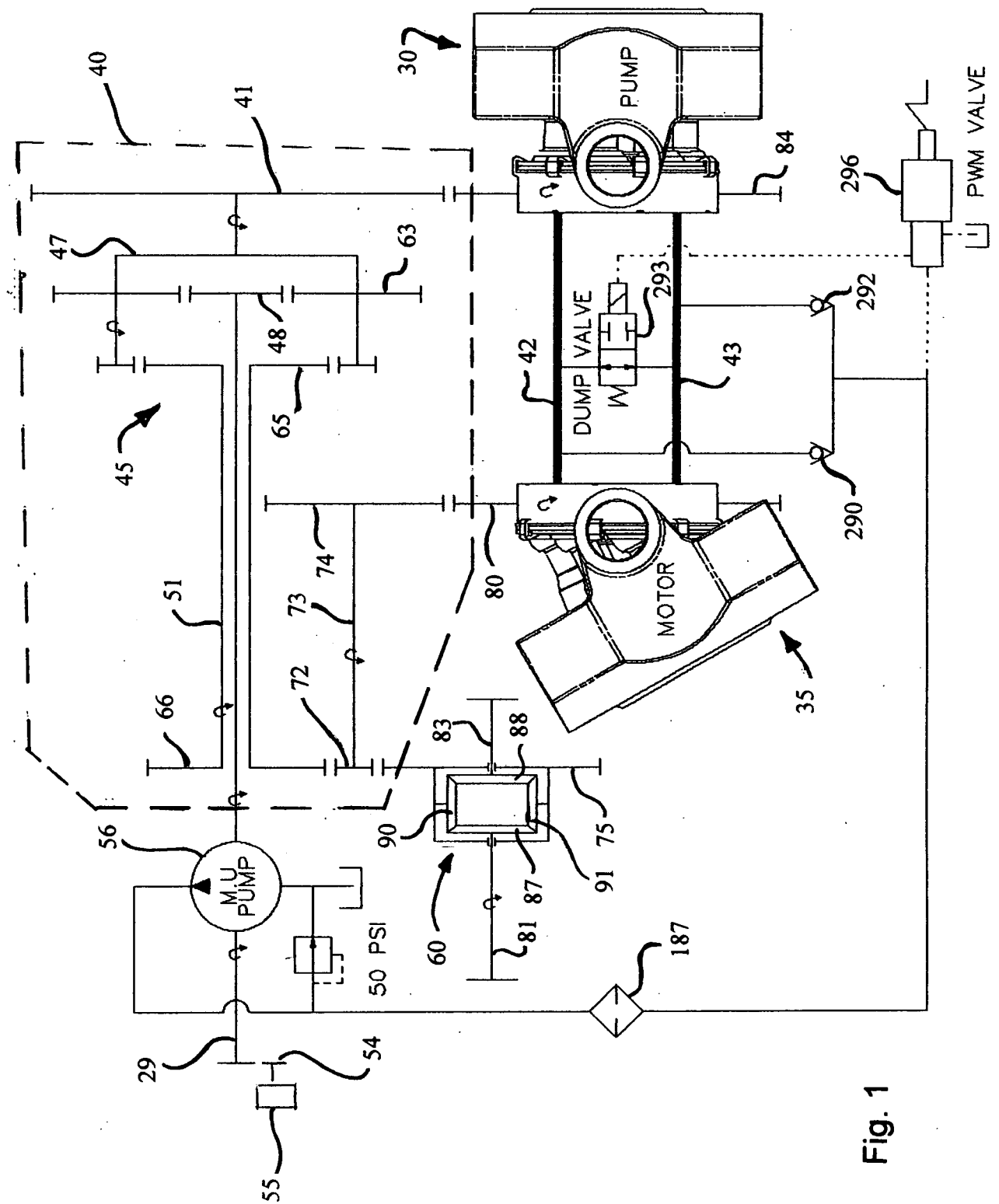


Fig. 1

2/28

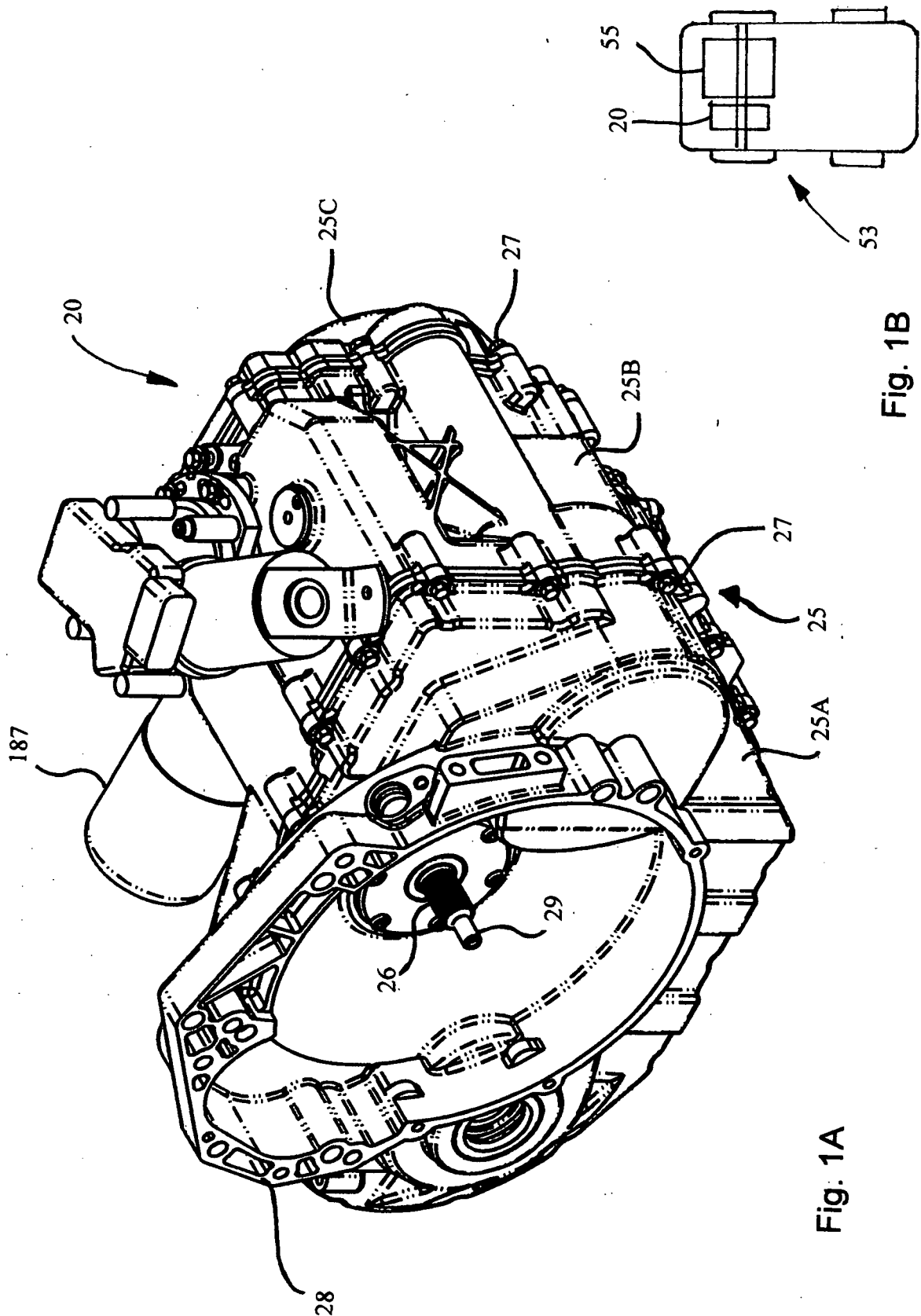


Fig. 1B

Fig. 1A

3/28

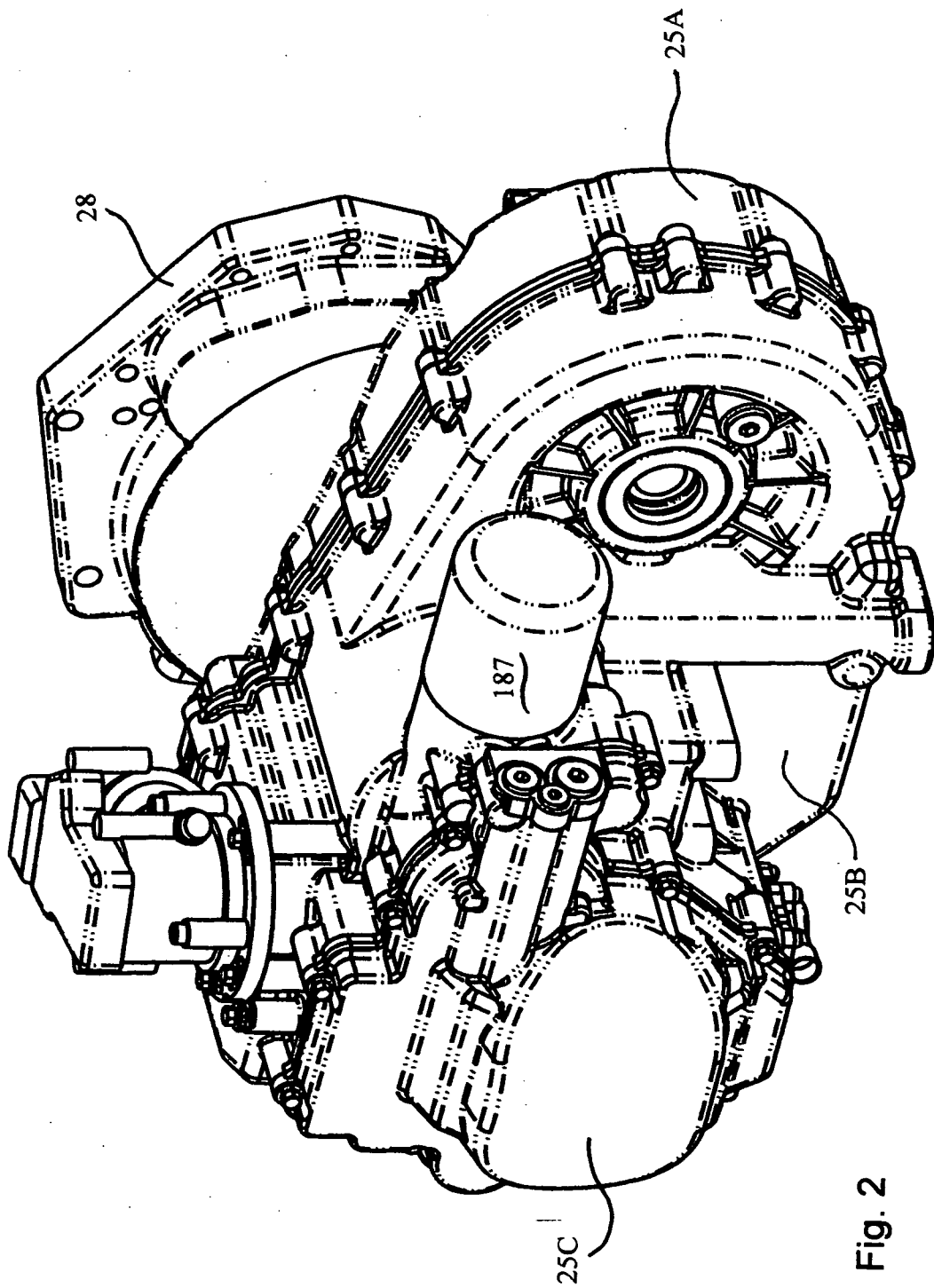
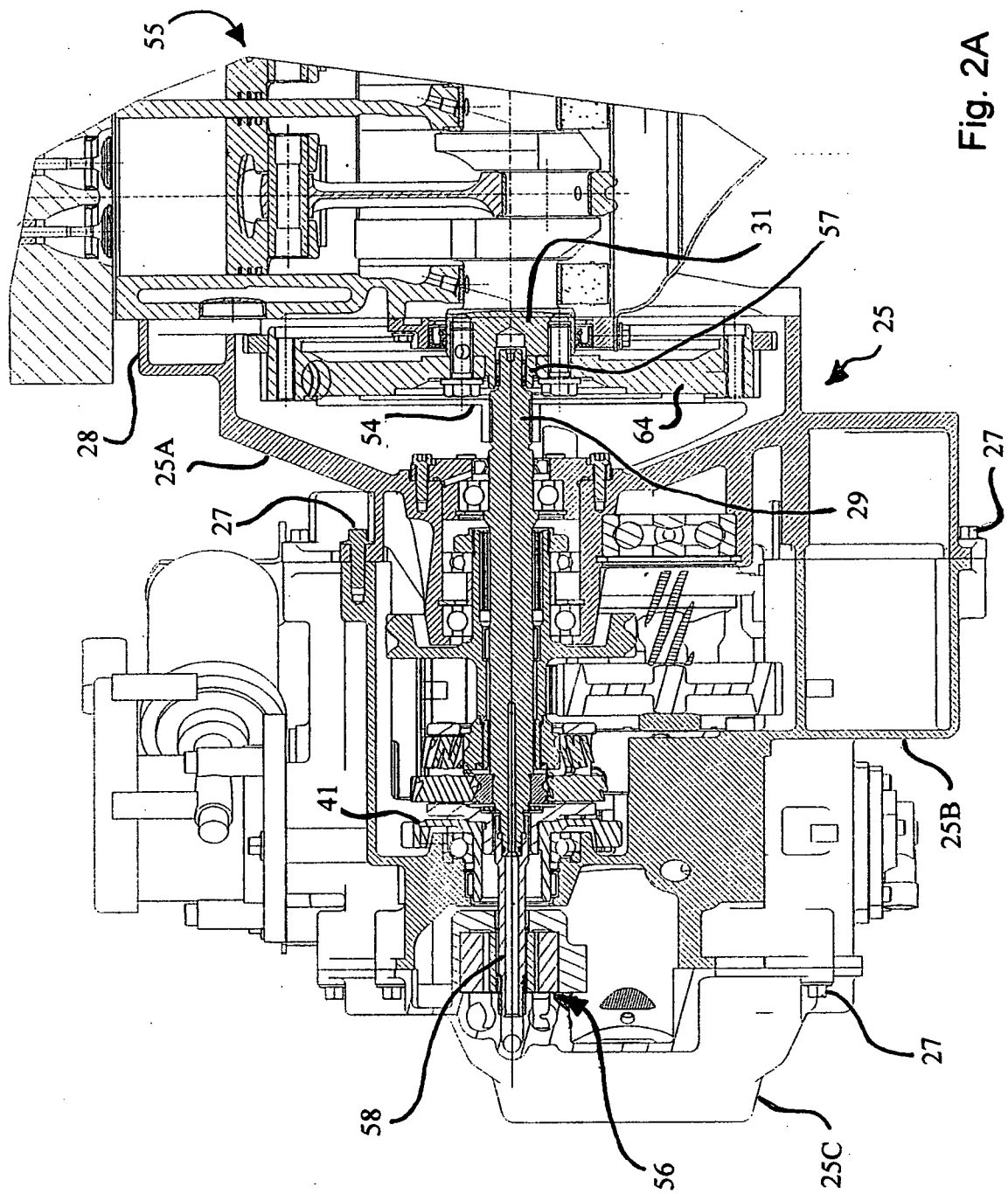


Fig. 2

4/28



5/28

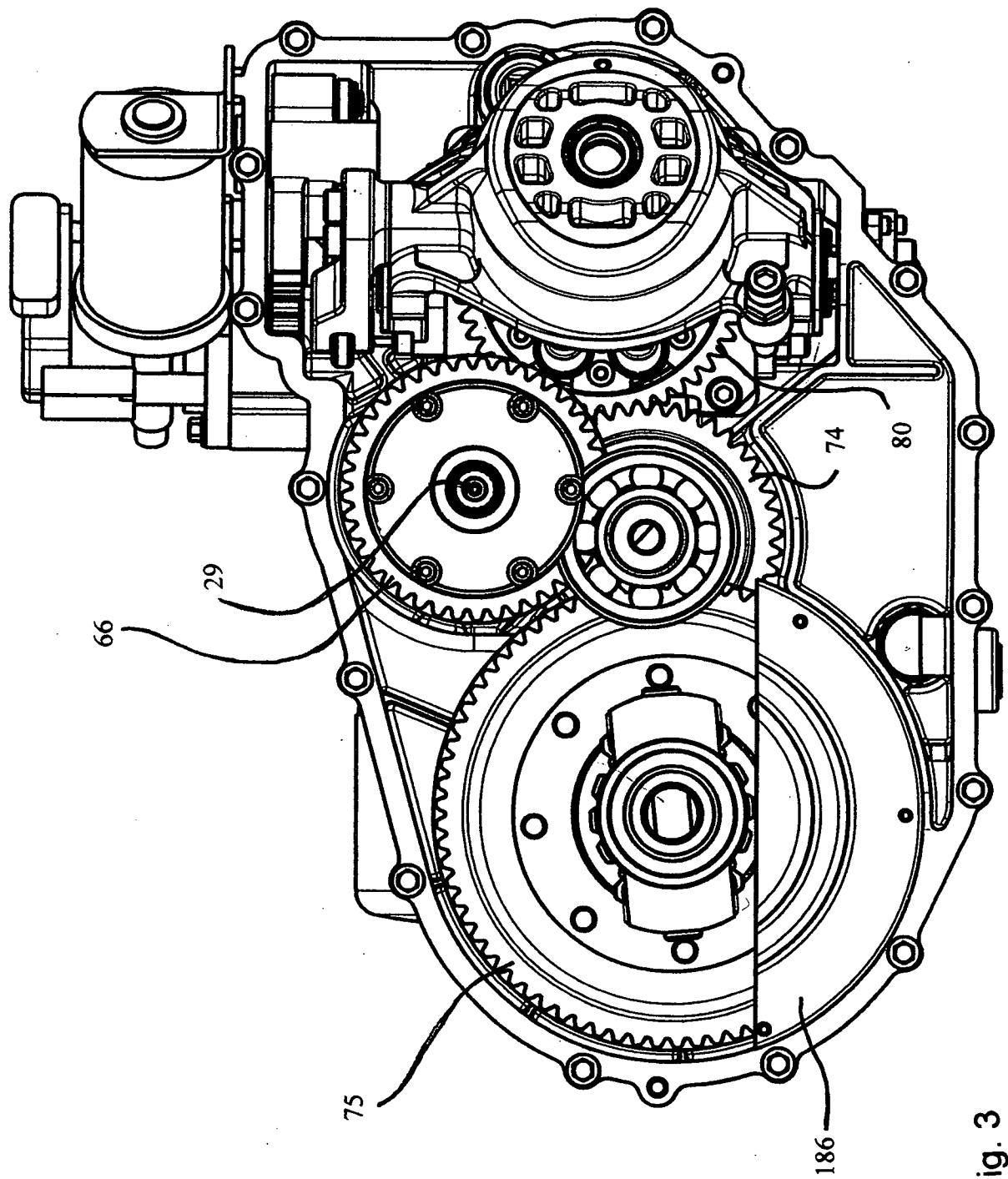


Fig. 3

6/28

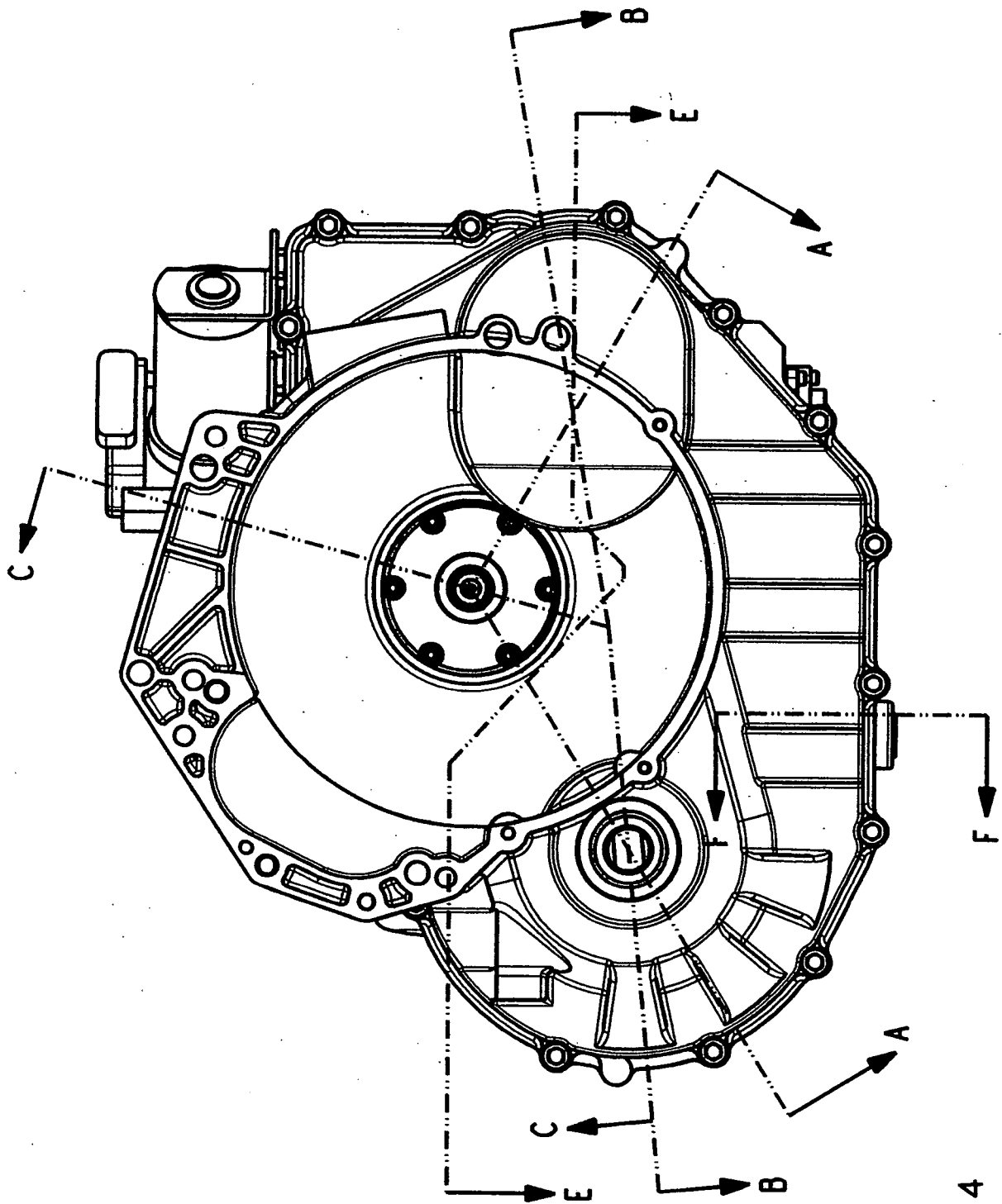


Fig. 4

7/28

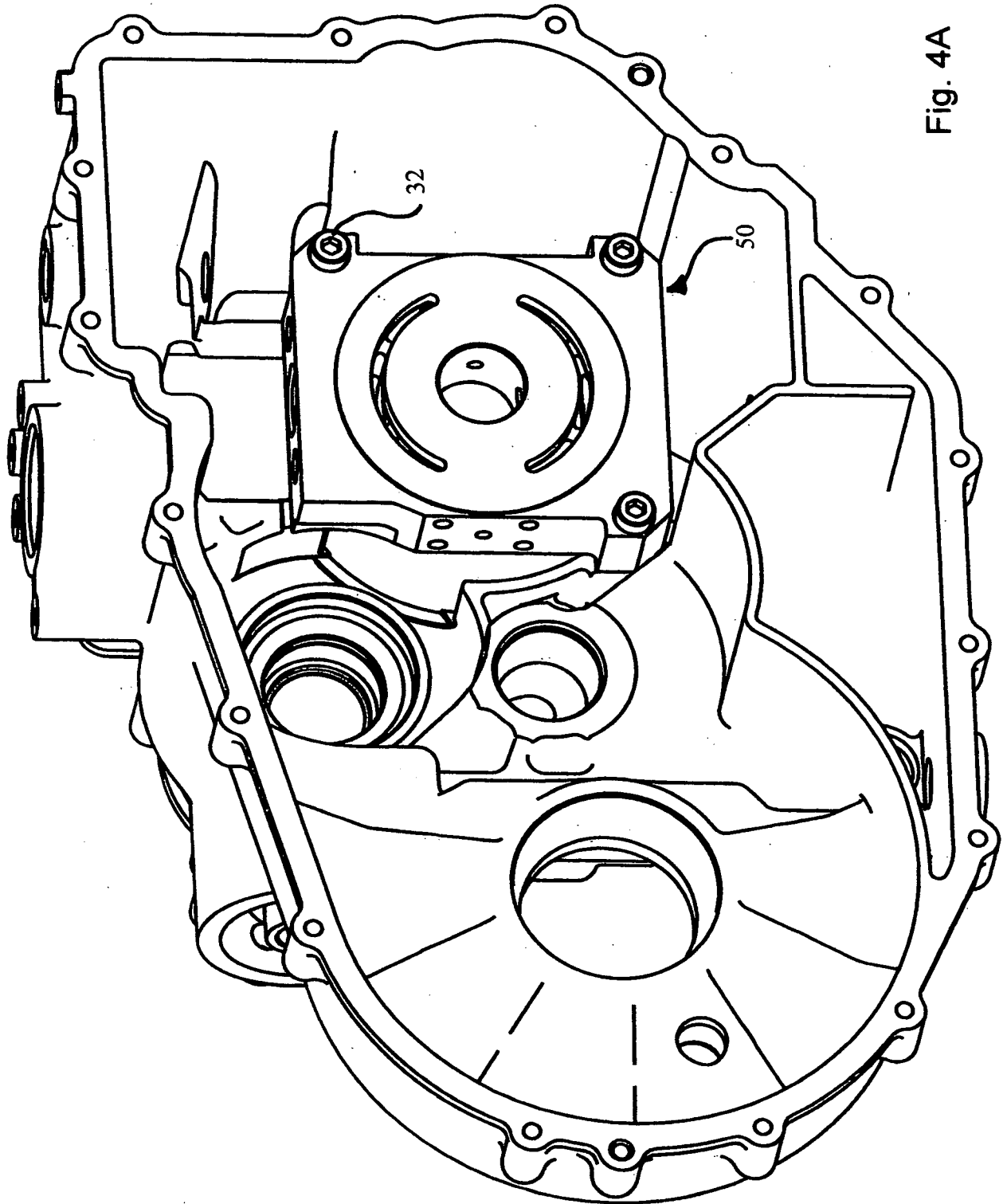


Fig. 4A

8/28

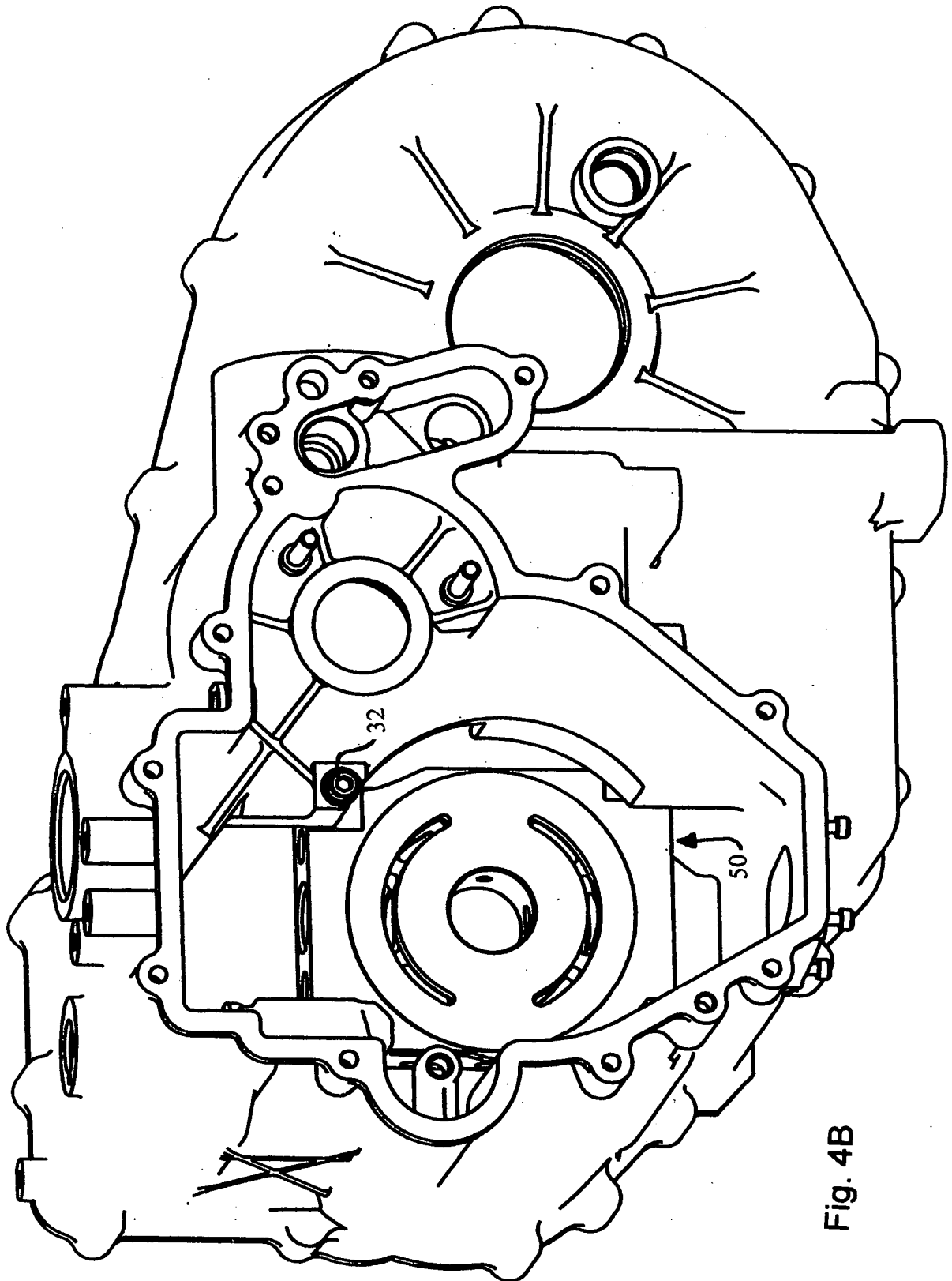
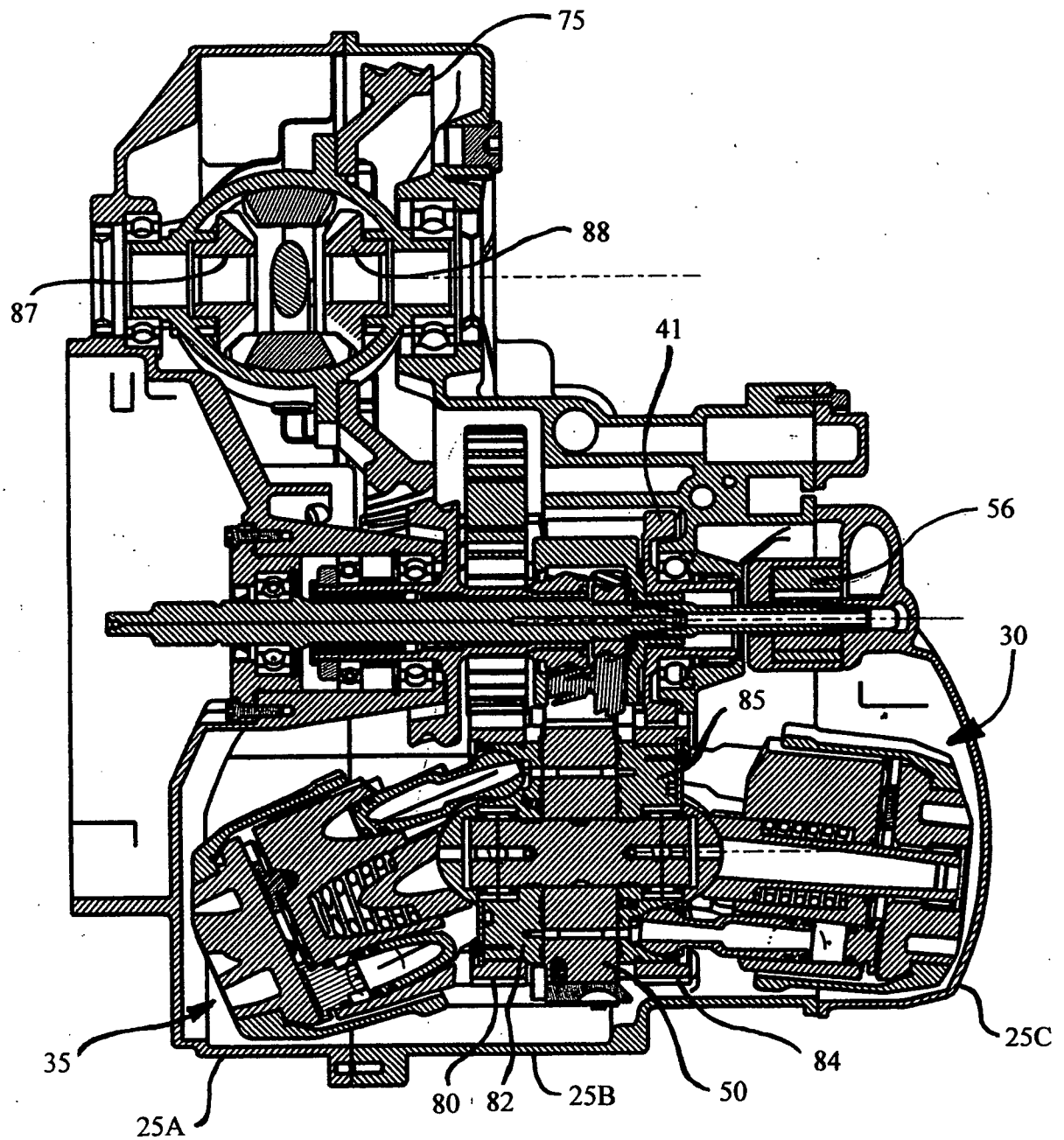


Fig. 4B

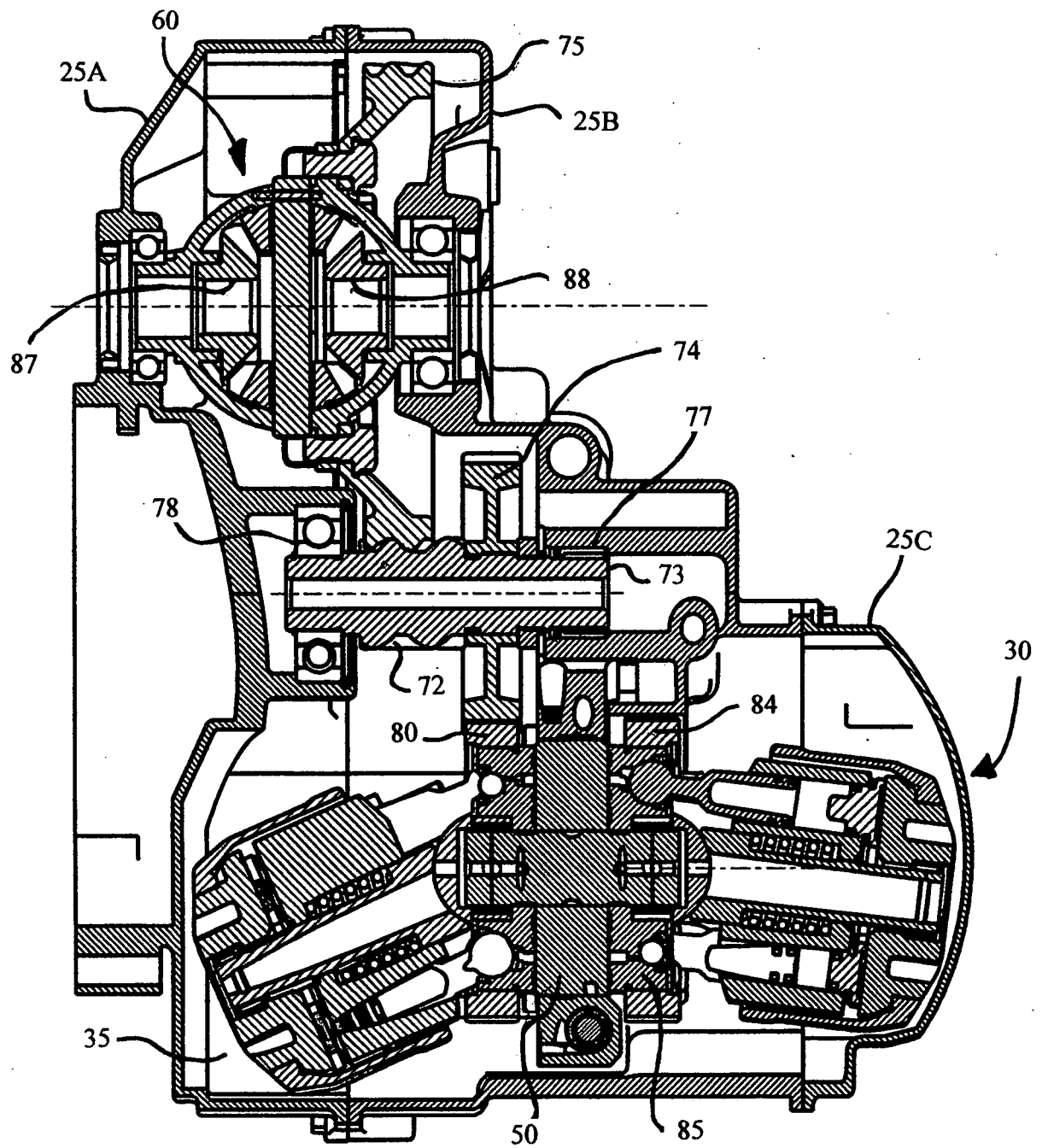
9/28



Section A - A

Fig. 5

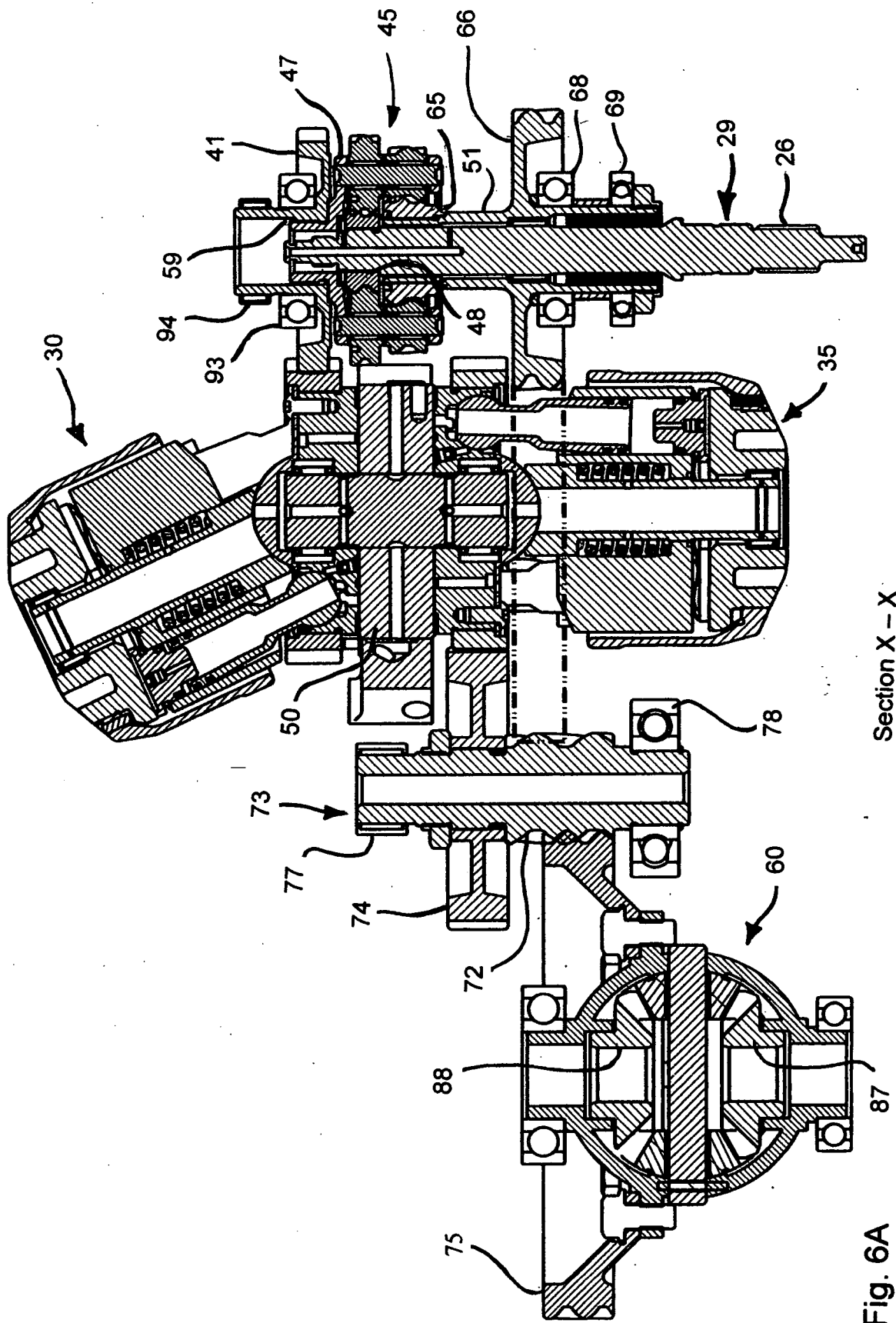
10/28



Section B - B

Fig. 6

11/28



12/28

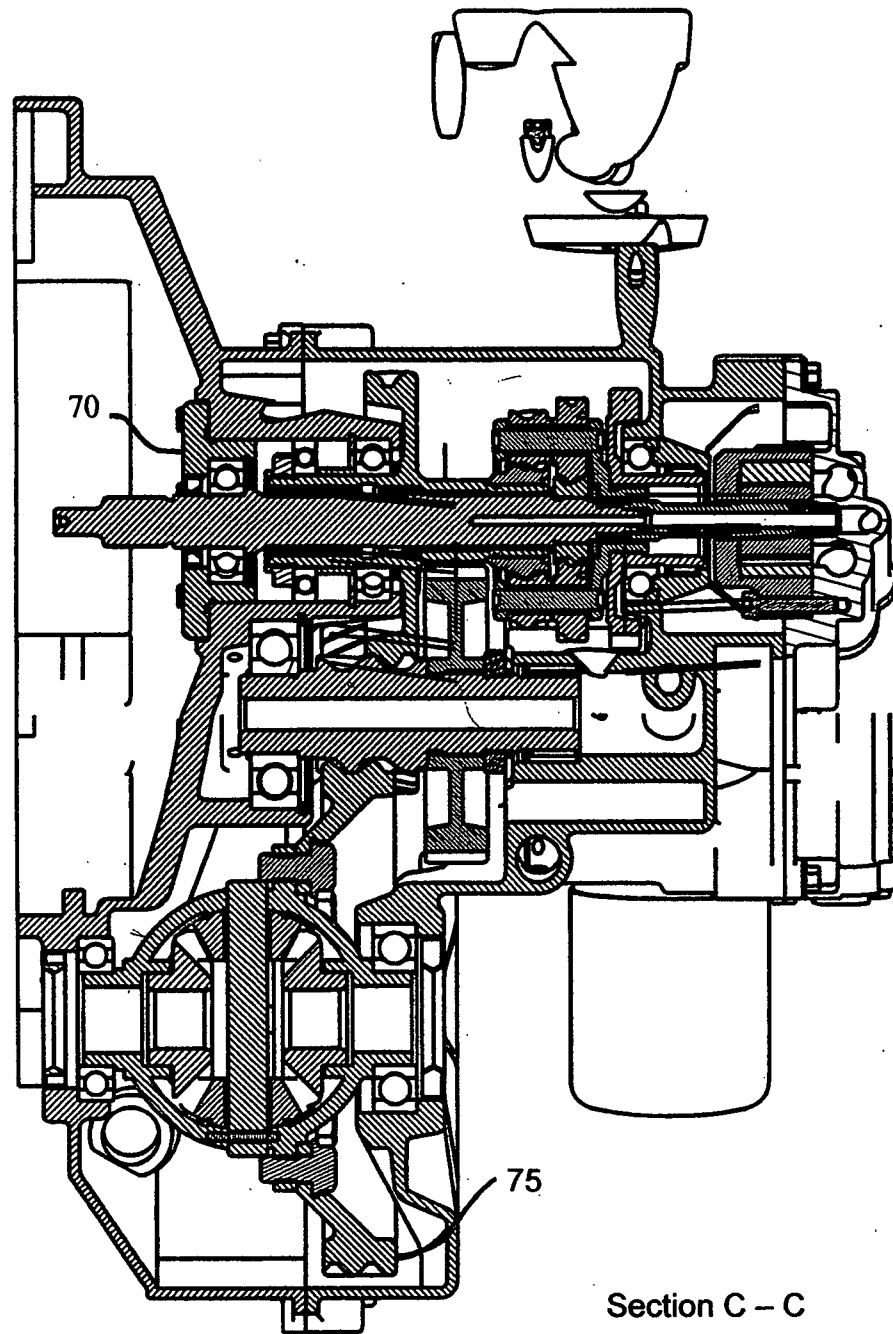
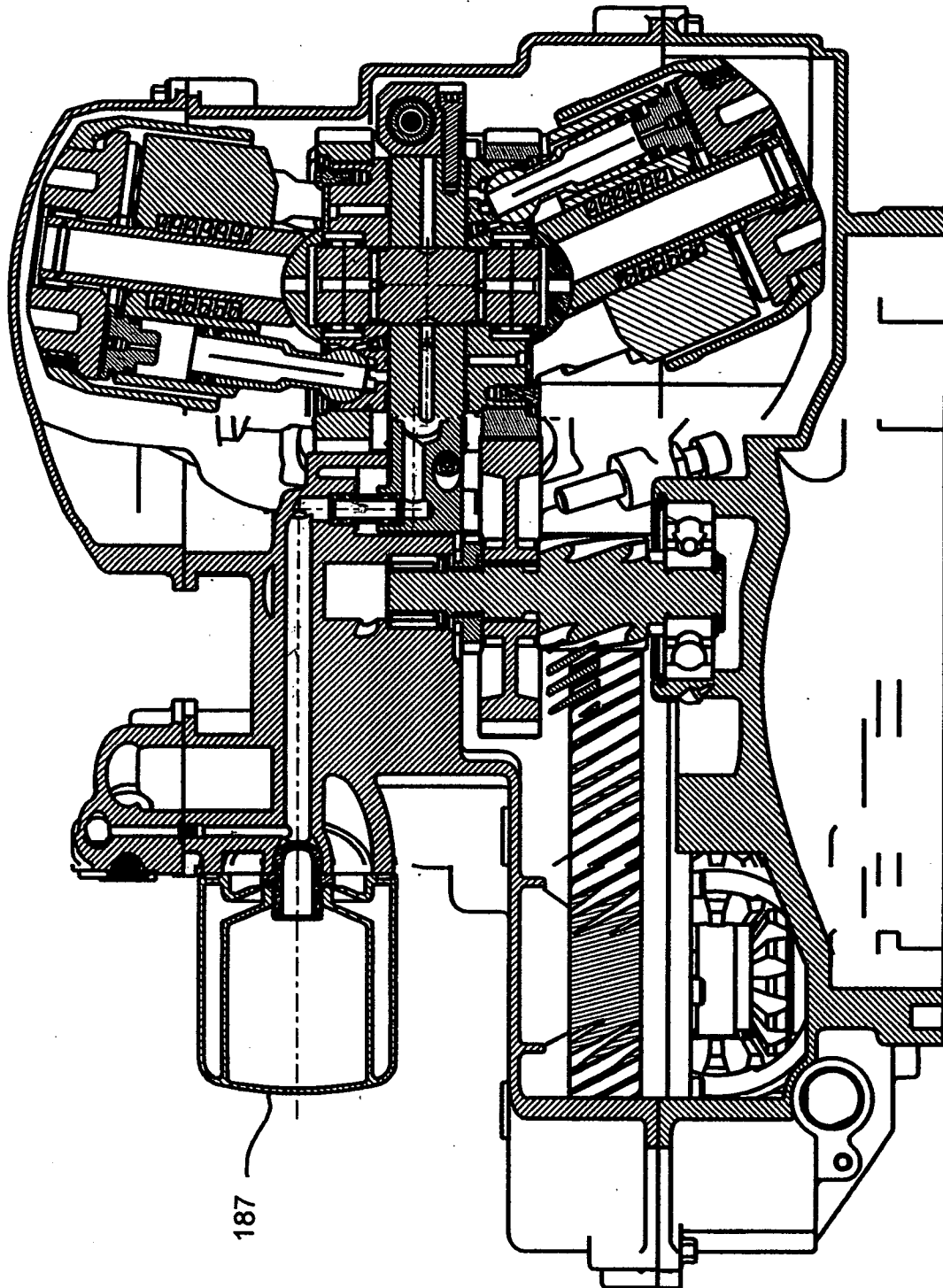


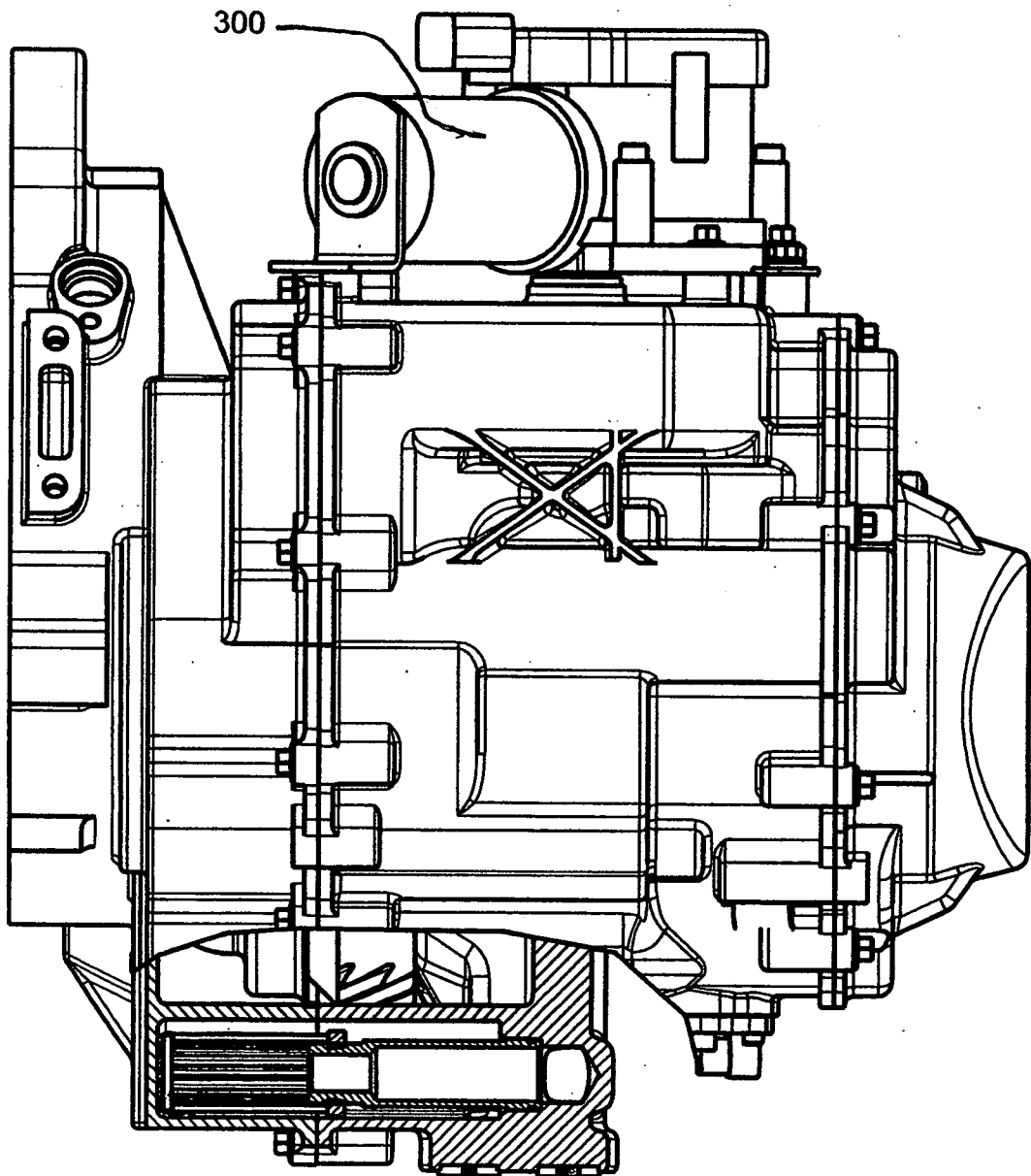
Fig. 7



Section E - E

Fig. 8

14/28



Partial Section F – F

Fig. 9

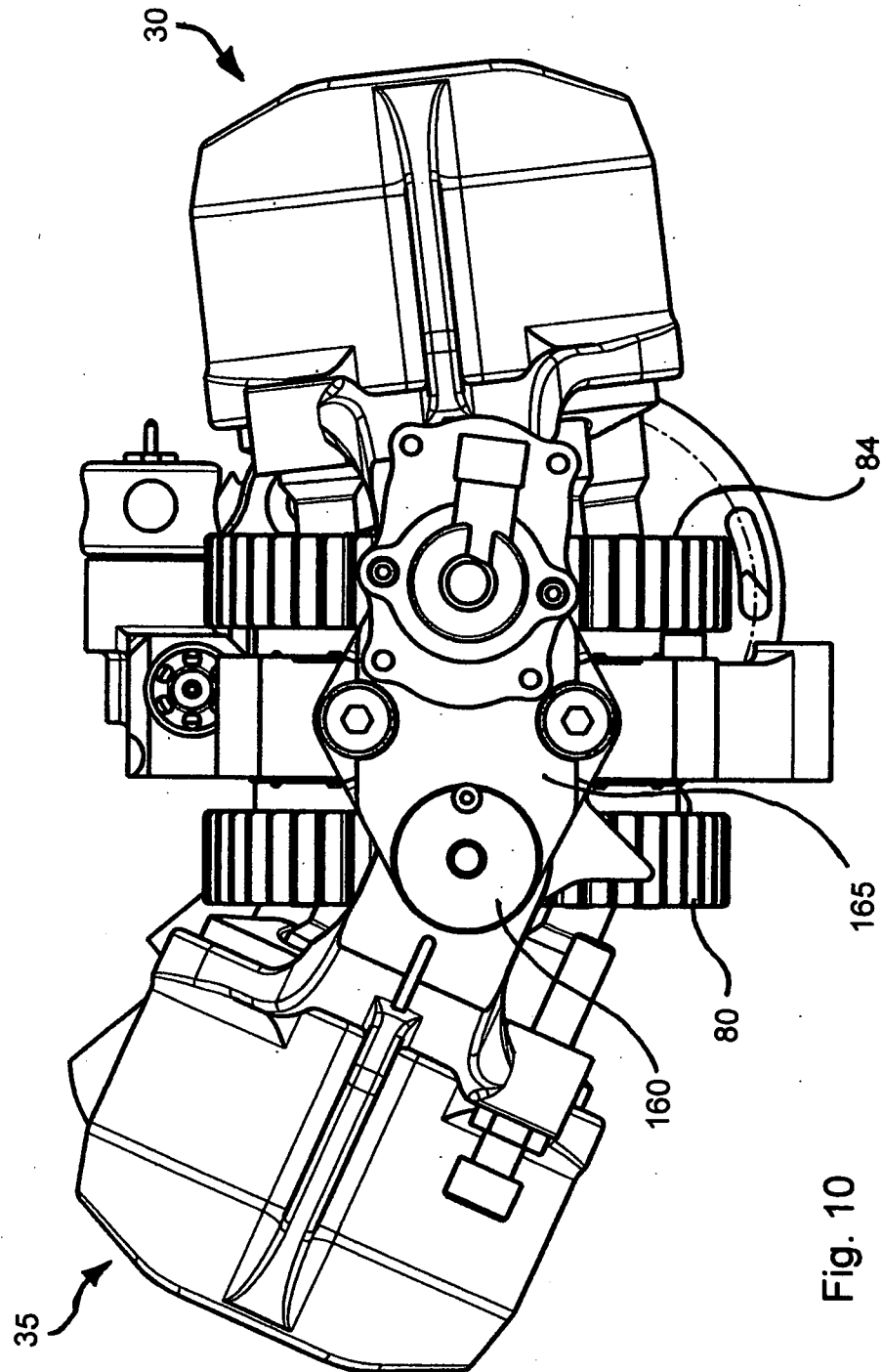


Fig. 10

16/28

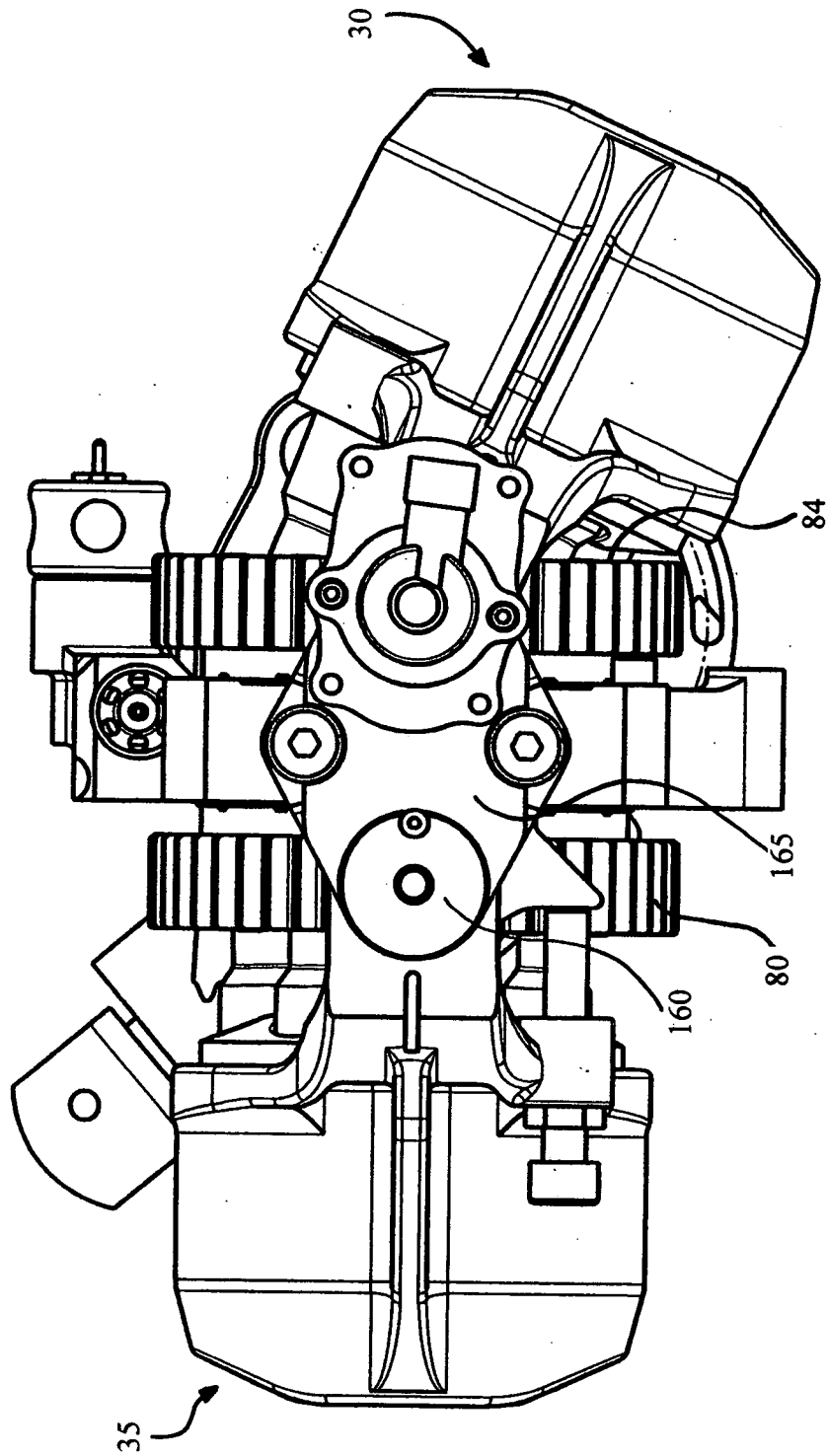


Fig. 11

17/28

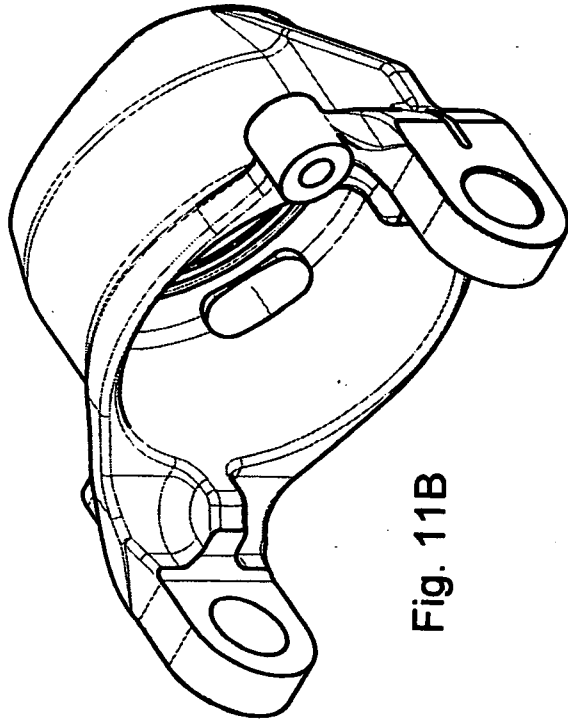


Fig. 11B

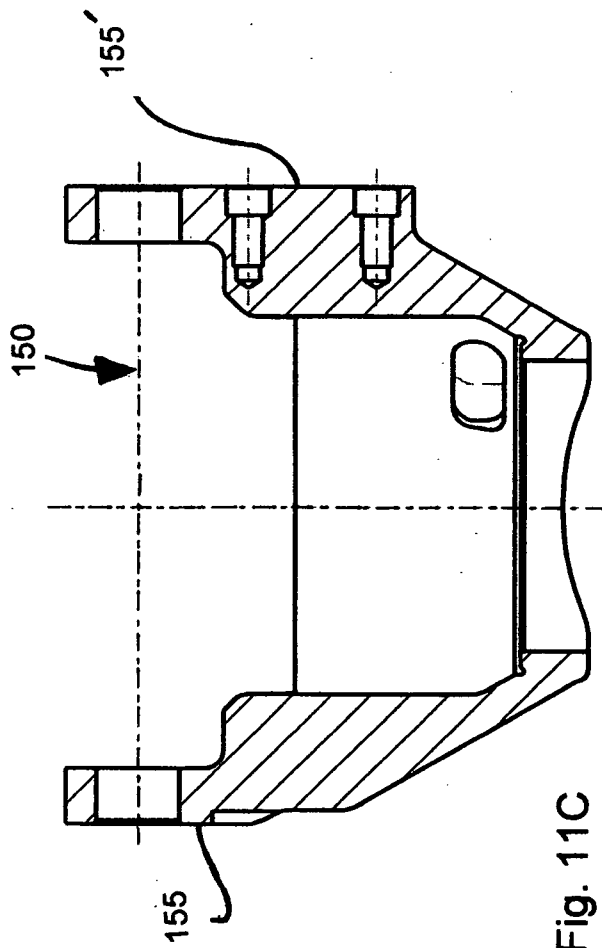


Fig. 11C

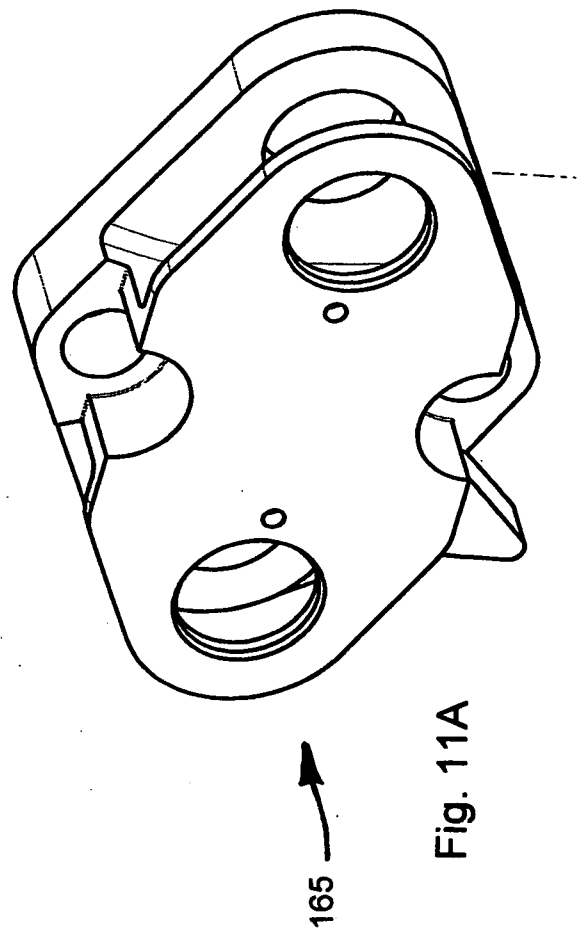


Fig. 11A

18/28

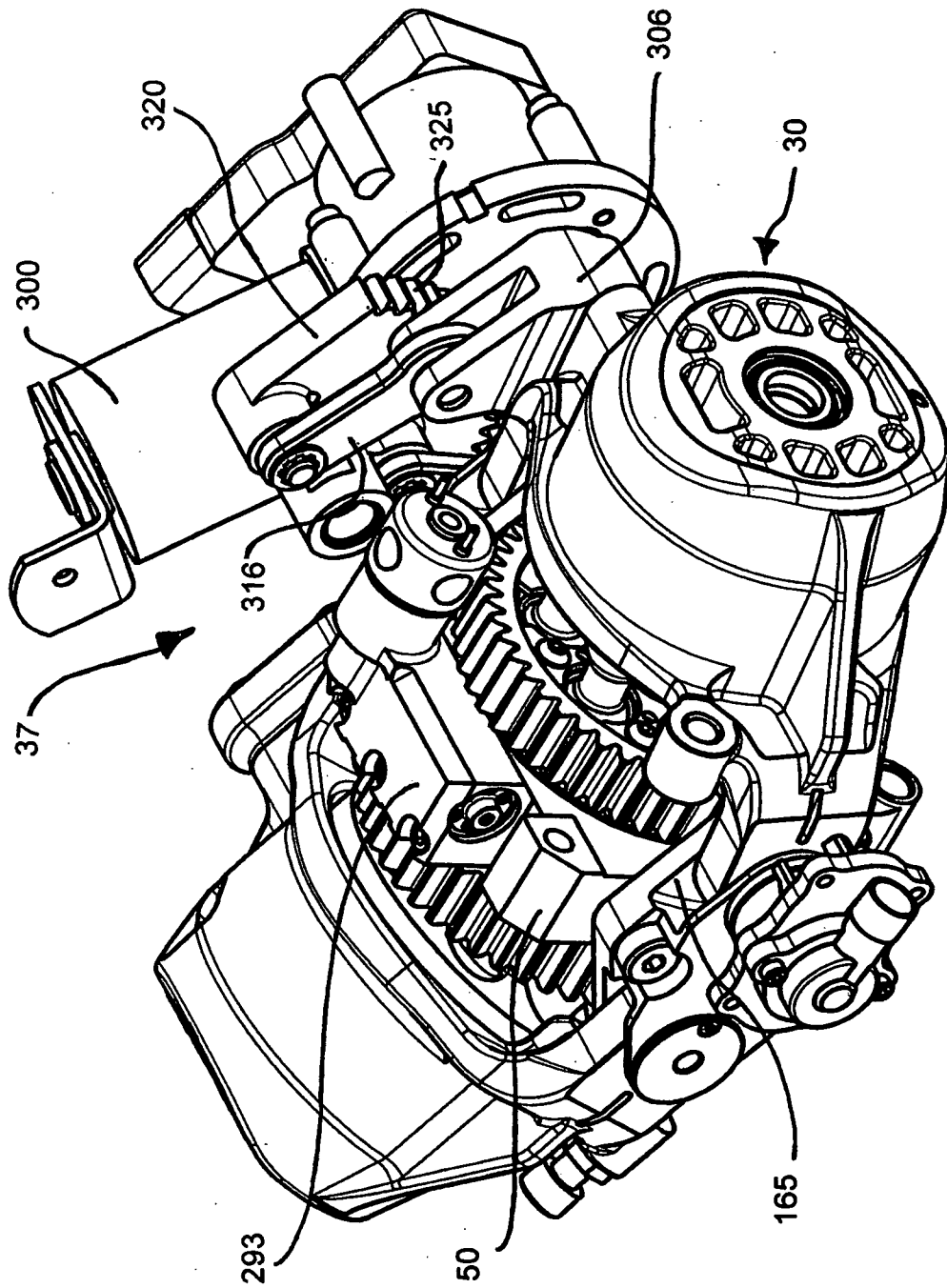


Fig. 12

19/28

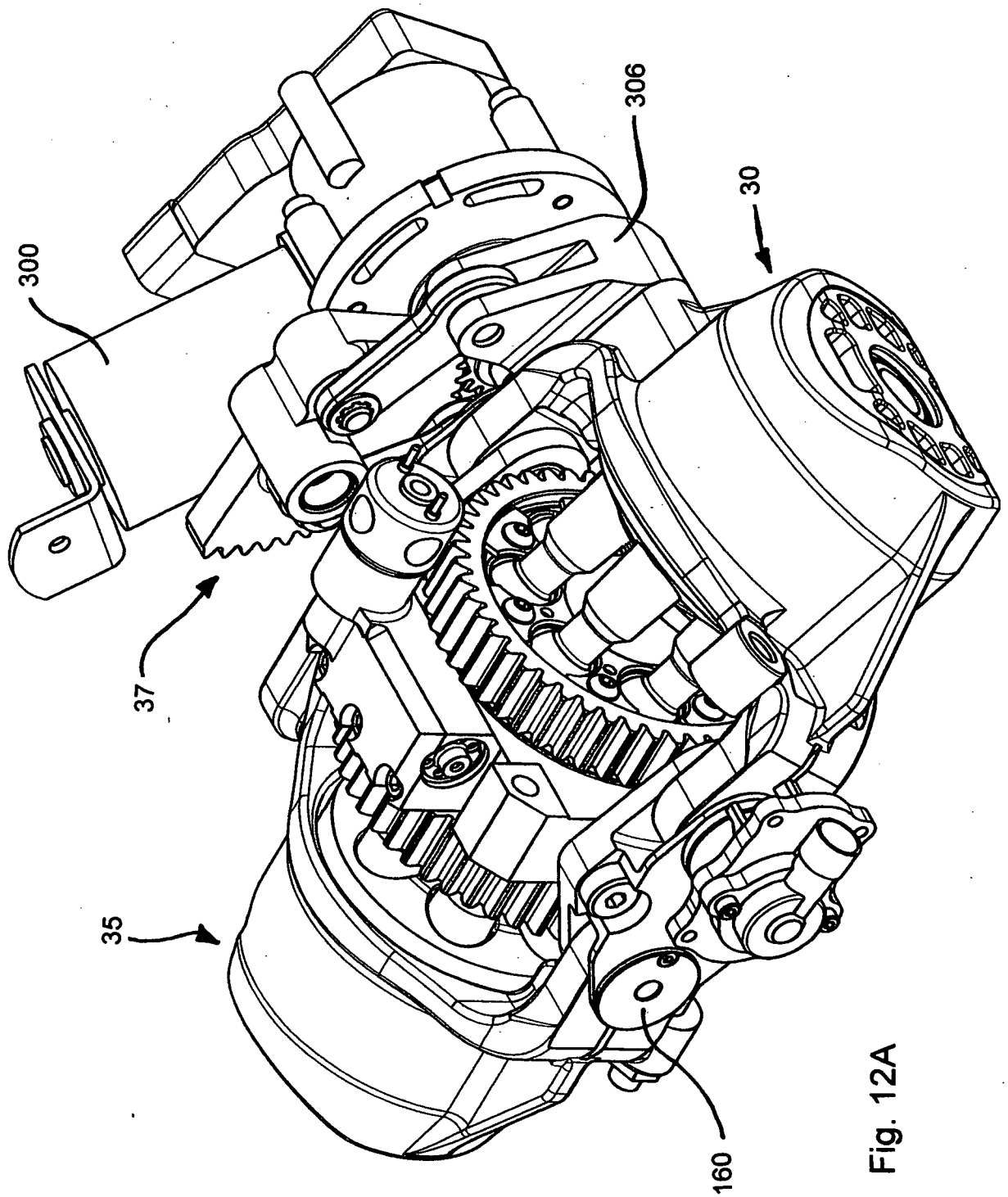


Fig. 12A

20/28

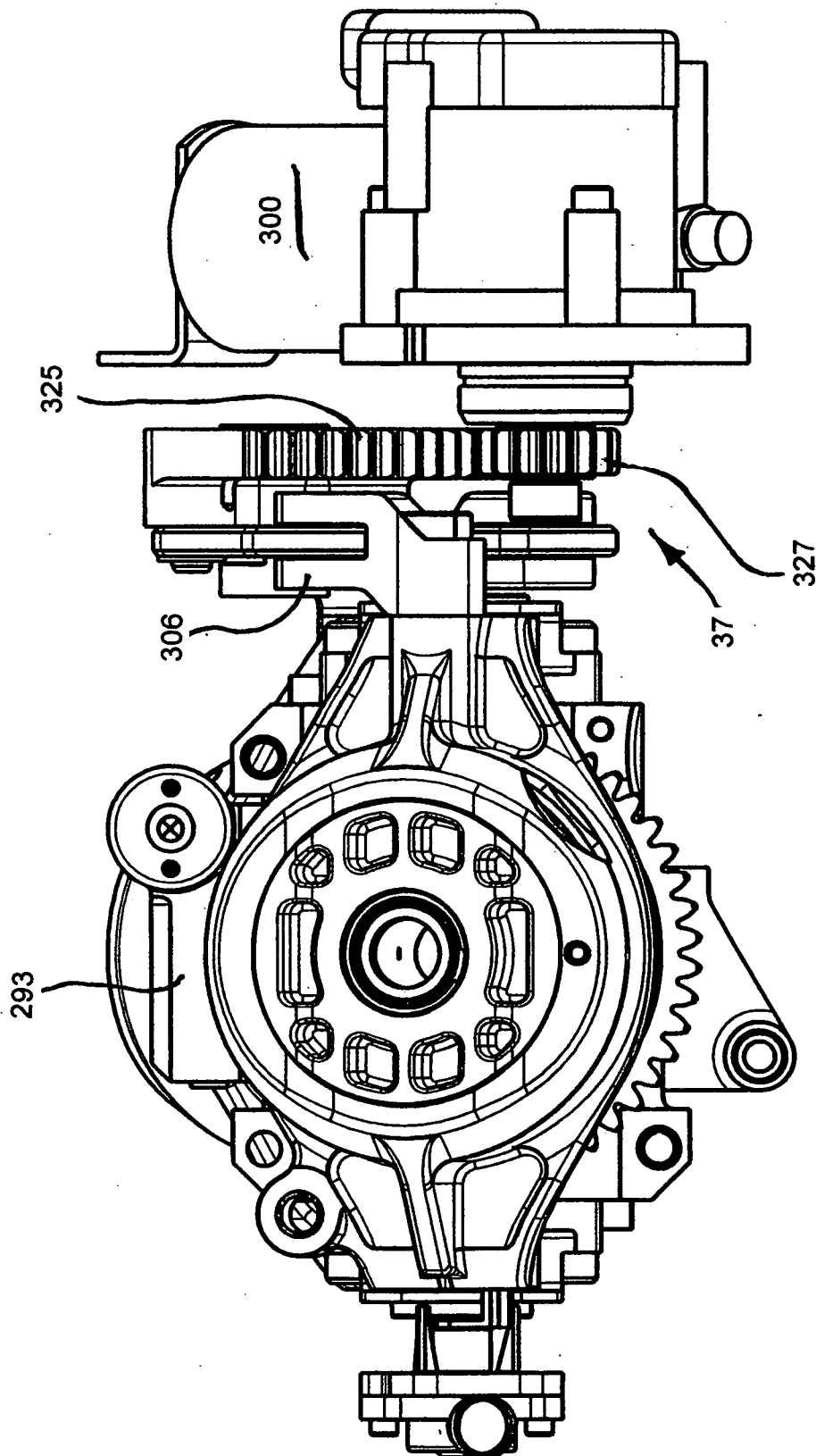


Fig. 13

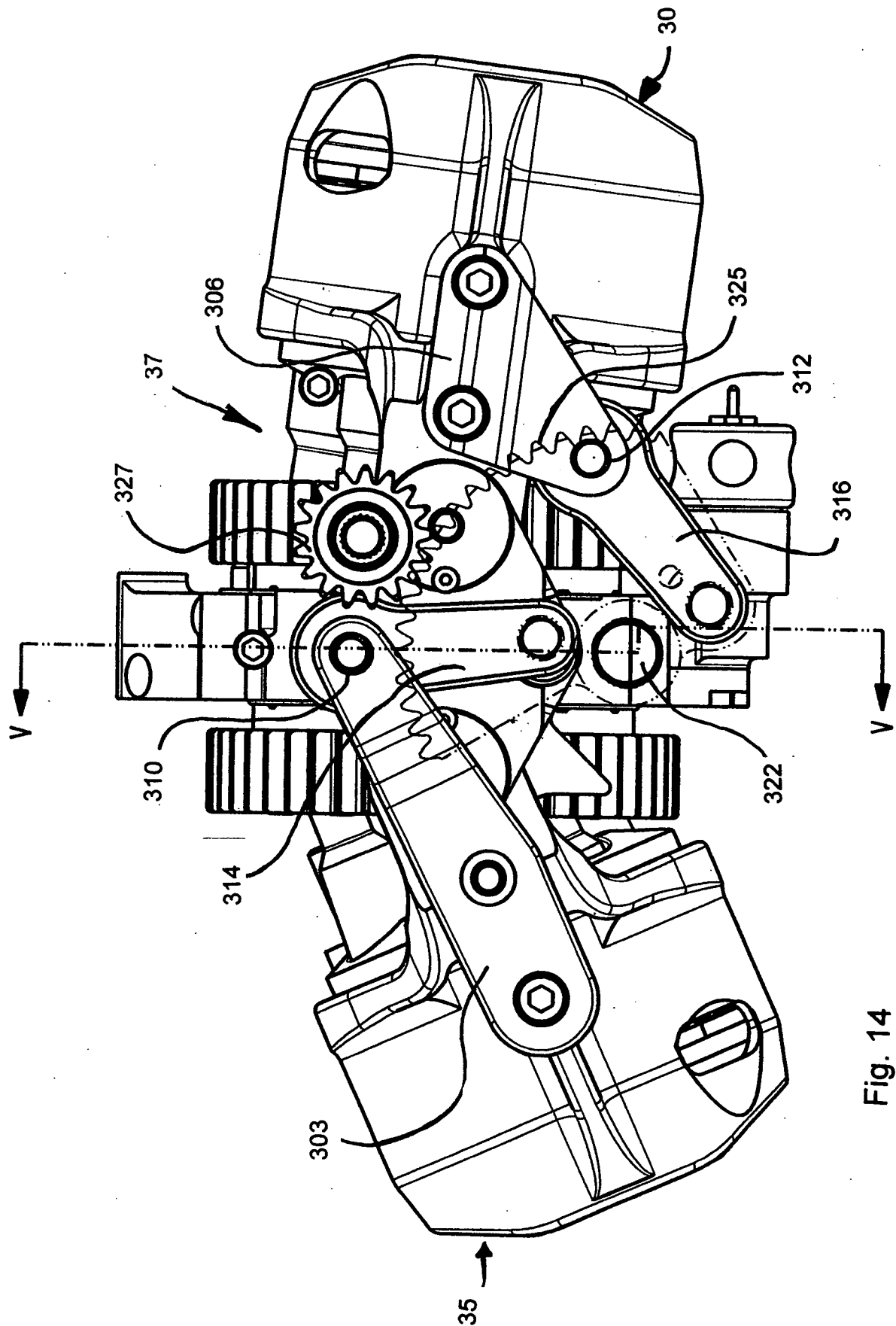


Fig. 14

22/28

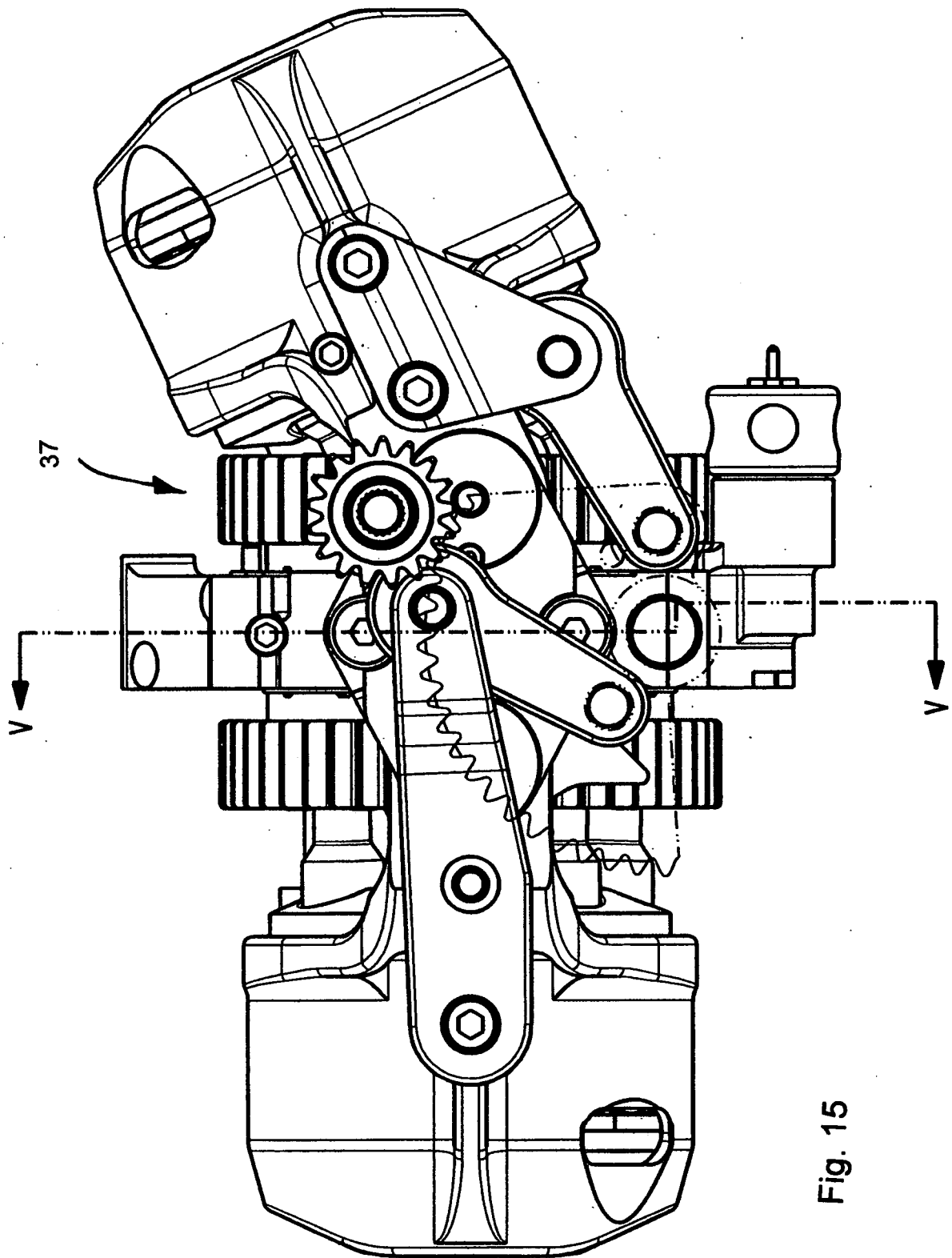


Fig. 15

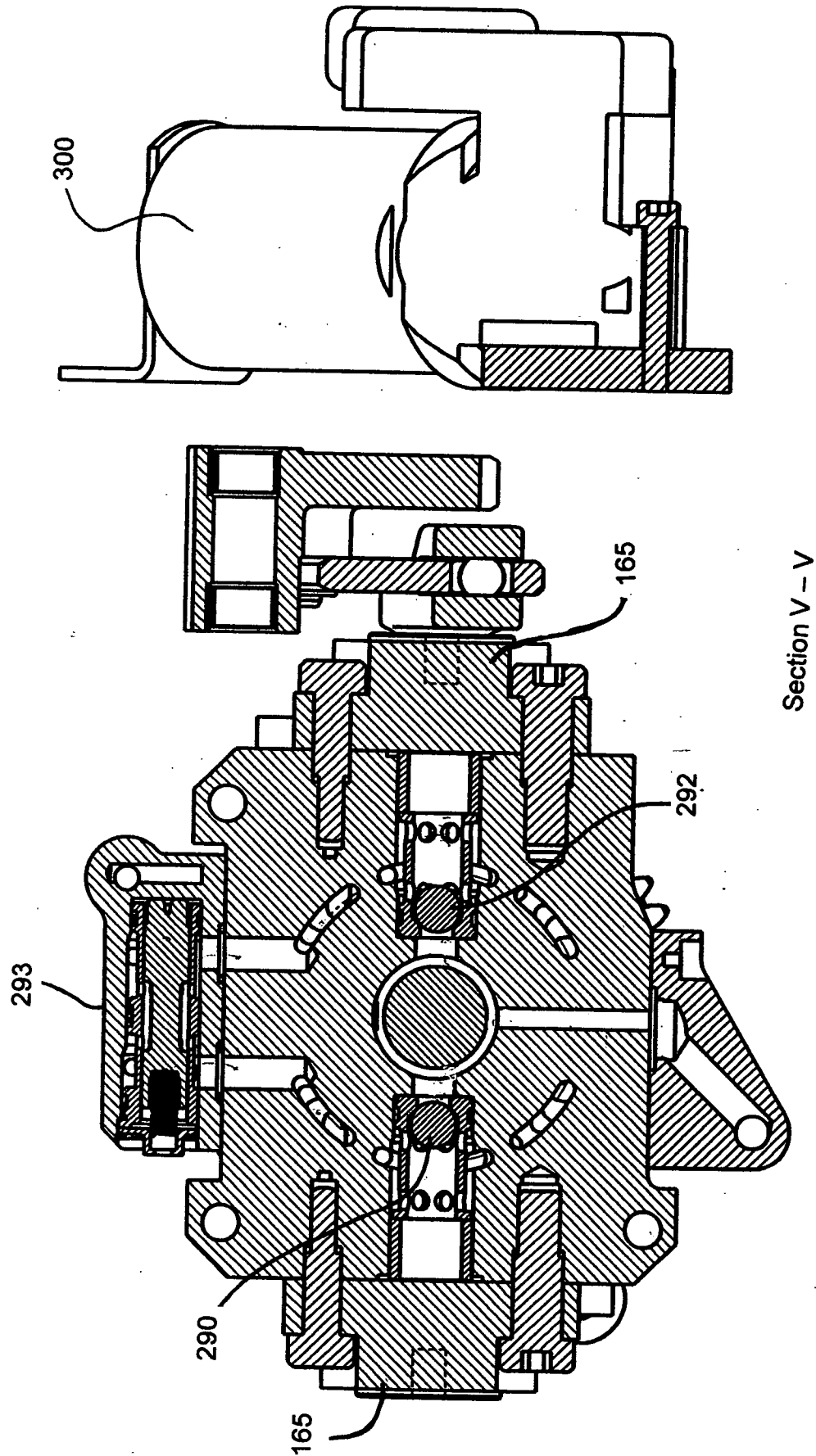


Fig. 16

24/28

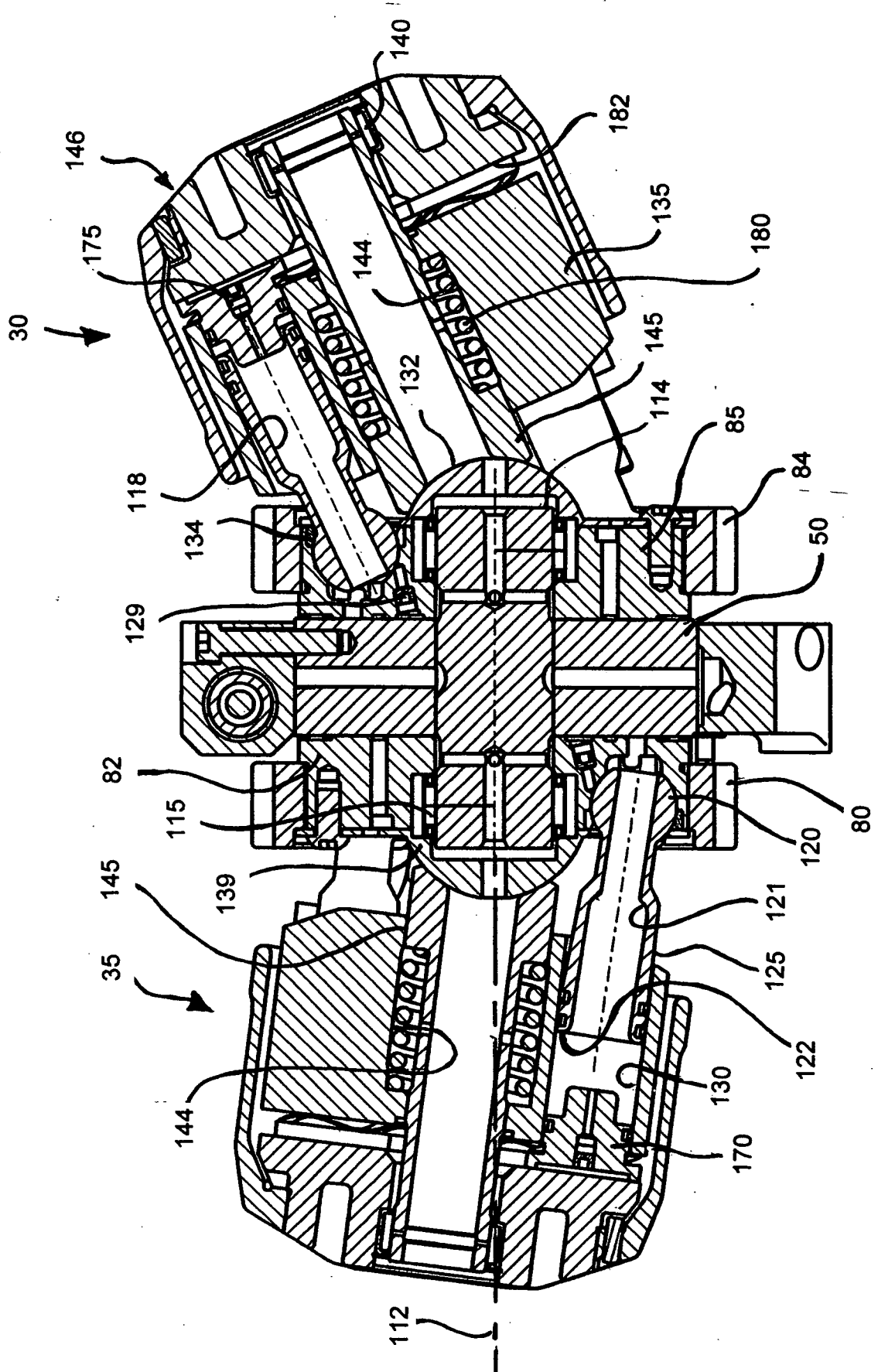


Fig. 17

25/28

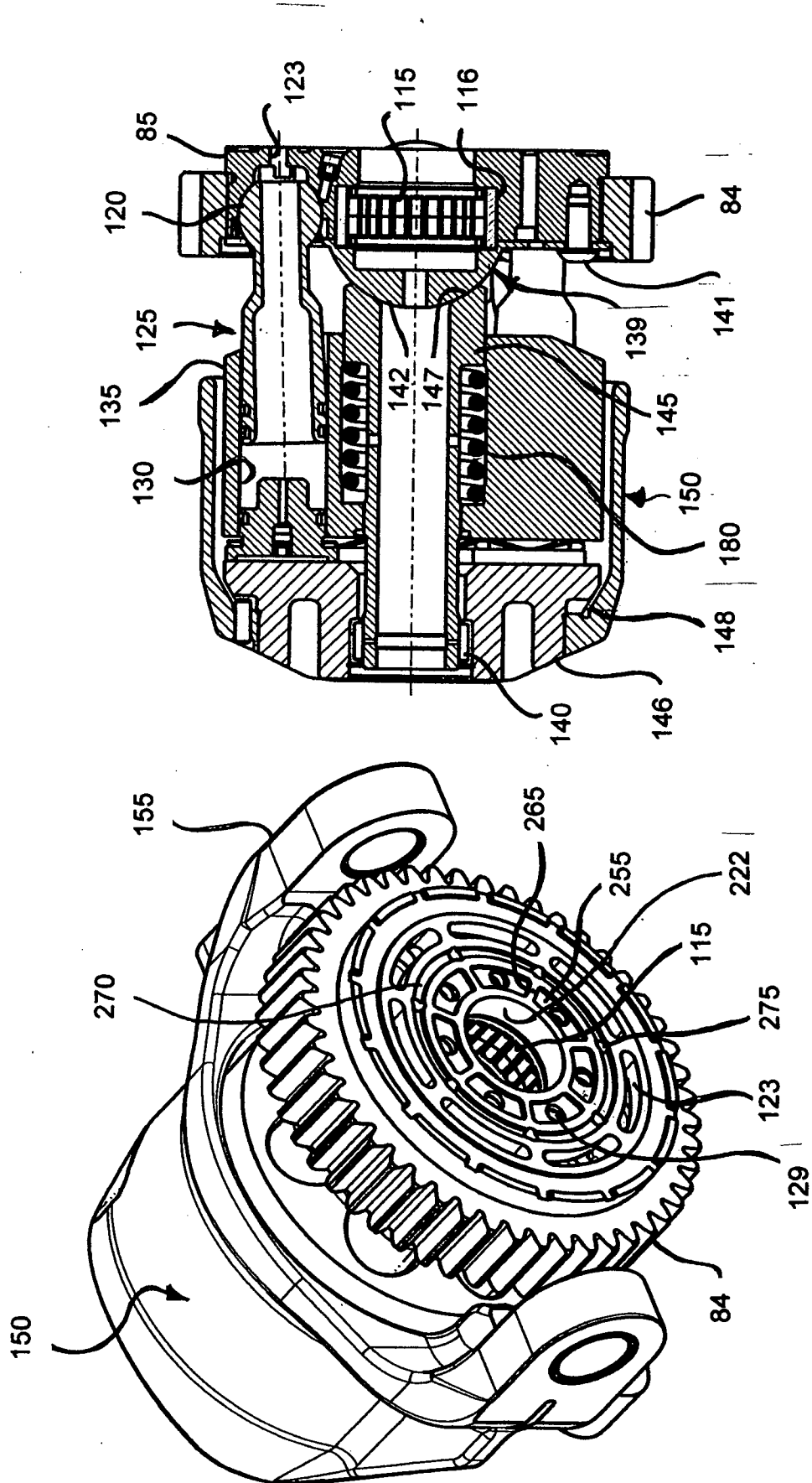


Fig. 18A

Fig. 18

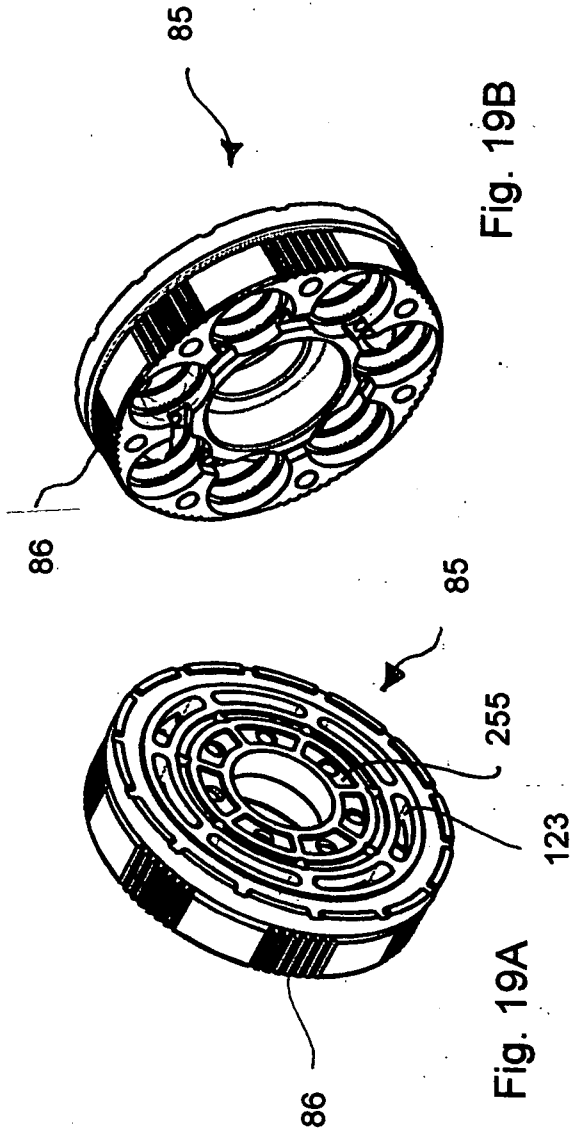


Fig. 19B

Fig. 19A

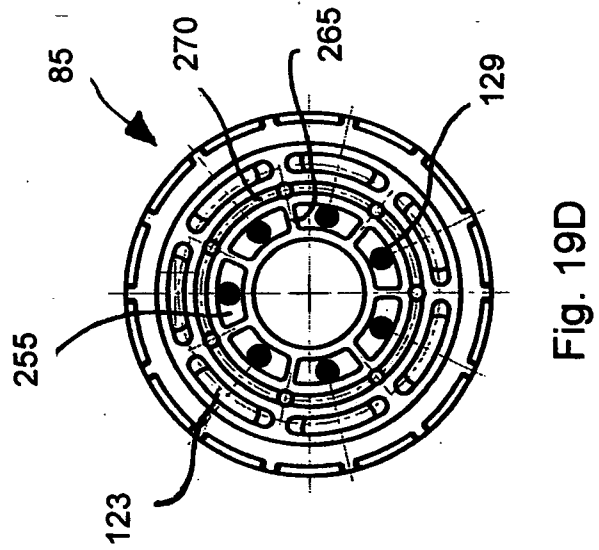


Fig. 19D

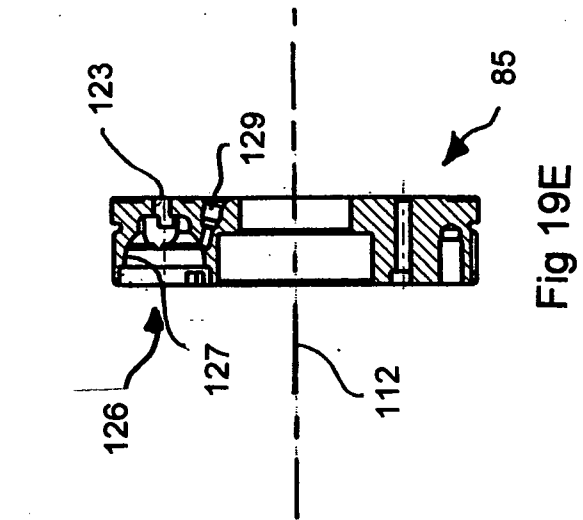


Fig. 19E

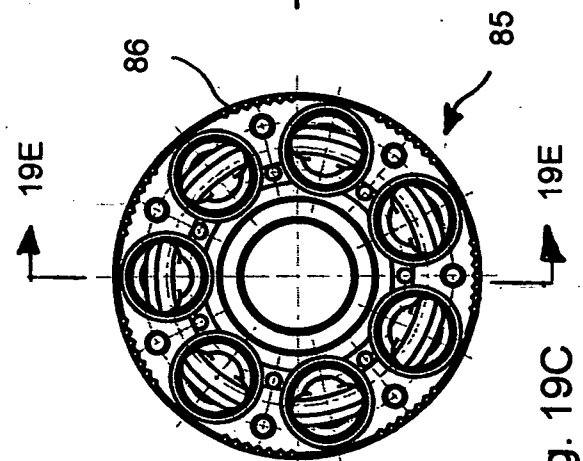


Fig. 19C

27/28

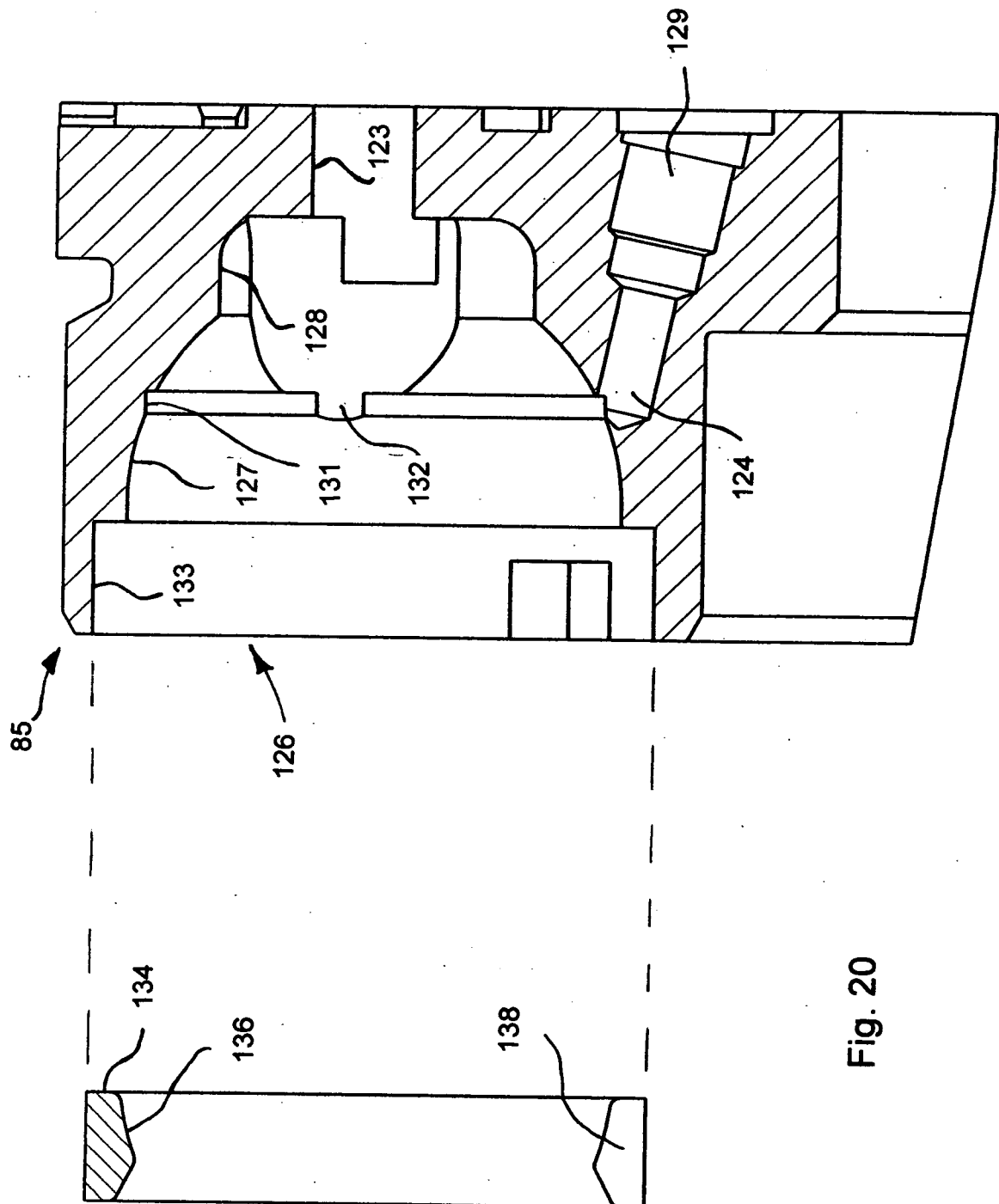


Fig. 20

28/28

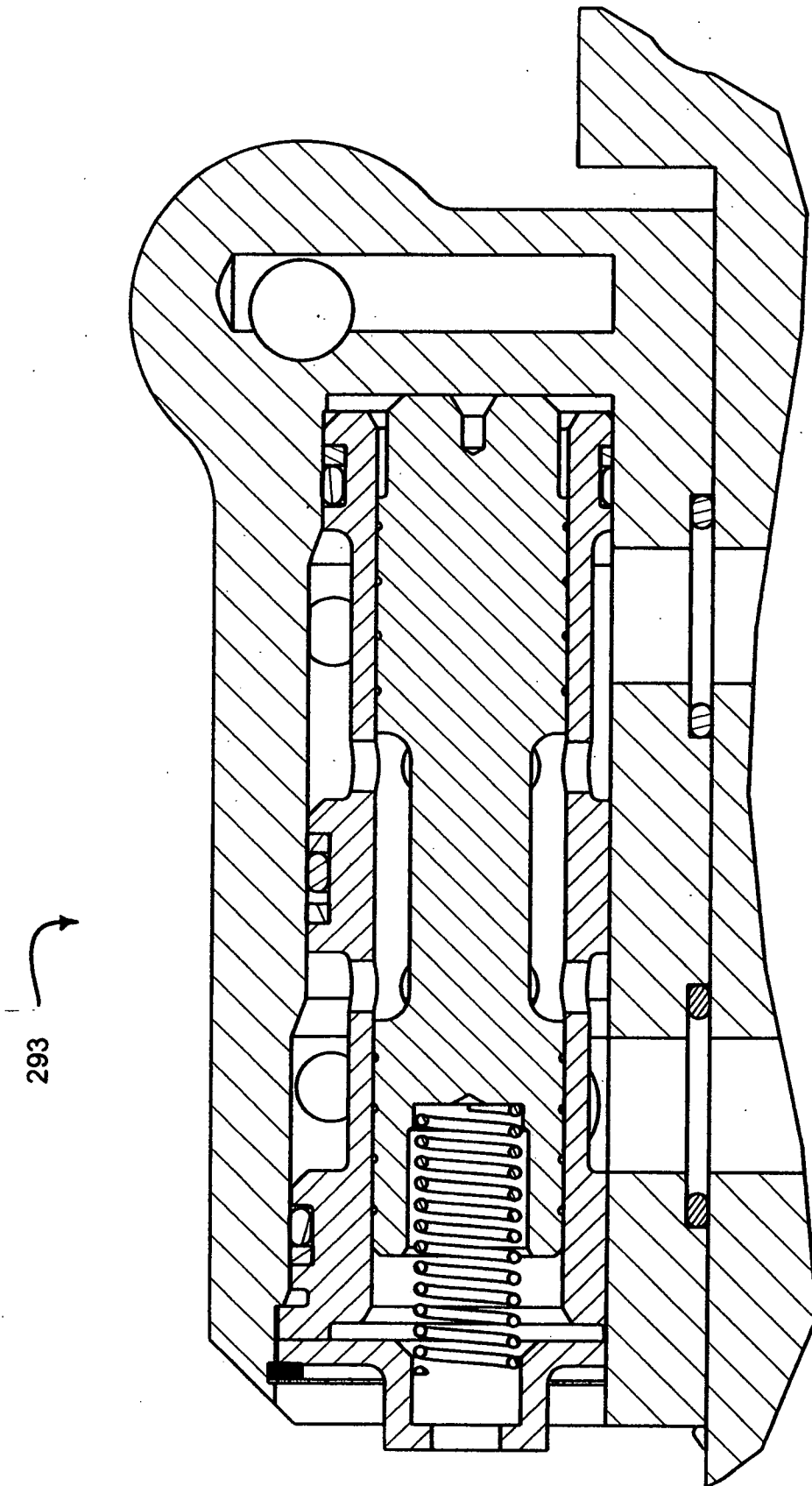


Fig. 21