OPIC
Office de la propriété intellectuelle du Canada

CIPO
Canadian Intellectual Property Office

(12) (19) (CA) **Demande-Application**

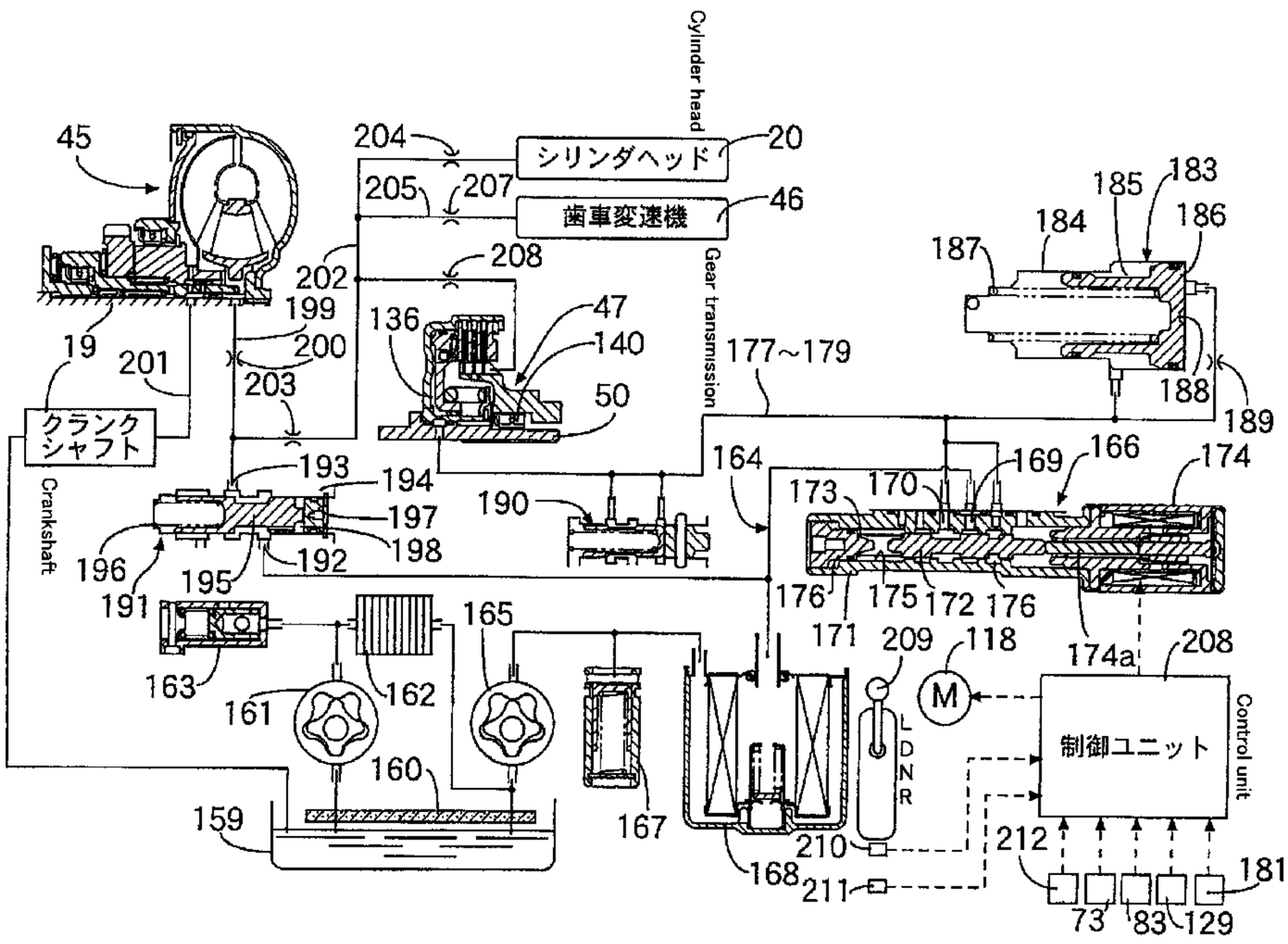
(21) (A1) **2,322,079**

(22) 2000/10/03
(43) 2001/04/05

(72) HORI, YOSHIAKI, JP
(72) TAKANO, NORIAKI, JP
(71) HONDA GIKEN KOGYO KABUSHIKI KAISHA, JP

(51) Int.Cl.[7] F16H 47/00
(30) 1999/10/05 (HEI-11-284209) JP

(54) **DISPOSITIF DE TRANSMISSION D'ENERGIE POUR VEHICULE**
(54) **POWER TRANSMISSION APPARATUS OF VEHICLE**



(57) In a power transmission apparatus of a vehicle having a hydraulic pressure type clutch capable of switching to transmit and cut power between a crankshaft of an engine and driving wheels and a clutch control valve for controlling hydraulic pressure operated to the clutch, the operational characteristic of the hydraulic pressure clutch is made variable and an adjustor mechanism for compensating for a wear amount of clutch plates is dispensed with. A clutch control valve includes a valve housing having an input port connected to a hydraulic pump and an output port connected to a clutch, a spool capable of switching to communicate and cut the input port and the output port and slidably fitted to the valve housing and a linear solenoid coaxially connected to the spool and is constituted to be capable of controlling output hydraulic pressure from the output port to a value in accordance with input current of the linear solenoid.

JJ-10 976CA

ABSTRACT OF THE DISCLOSURE

In a power transmission apparatus of a vehicle having a hydraulic pressure type clutch capable of switching to transmit and cut power between a crankshaft of an engine and driving wheels and a clutch control valve for controlling hydraulic pressure operated to the clutch, the operational characteristic of the hydraulic pressure clutch is made variable and an adjustor mechanism for compensating for a wear amount of clutch plates is dispensed with.

A clutch control valve includes a valve housing having an input port connected to a hydraulic pump and an output port connected to a clutch, a spool capable of switching to communicate and cut the input port and the output port and slidably fitted to the valve housing and a linear solenoid coaxially connected to the spool and is constituted to be capable of controlling output hydraulic pressure from the output port to a value in accordance with input current of the linear solenoid.

JJ-10 976CA

TITLE: POWER TRANSMISSION APPARATUS OF VEHICLE

FIELD OF THE INVENTION

The present invention relates to a power transmission apparatus of a vehicle having a hydraulic pressure type clutch capable of transmitting and cutting power between a crankshaft of an engine and driving wheels and a clutch control valve for controlling hydraulic pressure operated to the clutch.

Conventionally, such a power transmission apparatus has already been known in, for example, Japanese Patent Laid-Open No. S47-40417 or the like.

However, according to the above-described conventional constitution, a clutch control valve is an electromagnetic opening and closing valve for switching to communicate and cut a hydraulic pump and a hydraulic clutch and achieves a function of only switching to supply and cut hydraulic pressure to the hydraulic clutch and accordingly, operational characteristic of the hydraulic clutch is determined to be constant, further, an adjustor mechanism is needed for the hydraulic clutch to compensate for a wear amount of clutch plates.

SUMMARY OF THE INVENTION

The present invention has been carried out in view of such a situation and it is an object thereof to provide a power transmission apparatus of a vehicle for making variable operational characteristic of a hydraulic clutch and dispensing with an adjuster mechanism for compensating for a wear amount of clutch plates.

In order to achieve the above-described object, a first aspect of the invention is characterized in that in a power transmission apparatus of a vehicle having a hydraulic pressure type clutch capable of switching to transmit and cut power between a crankshaft of an engine and driving wheels and a clutch control valve for controlling hydraulic pressure operated to the clutch, the

clutch control valve comprises a valve housing having an input port connected to a hydraulic pump and an output port connected to the clutch, a spool capable of switching to communicate and cut between the input port and the output port and slidably fitted to the valve housing and a linear solenoid coaxially connected to the spool and is constituted such that output hydraulic pressure from the output port can be controlled to a value in correspondence with input current to the linear solenoid.

According to such a constitution, the output hydraulic pressure of the clutch control valve and accordingly, hydraulic pressure operated to the clutch can be controlled by controlling the input current to the linear solenoid and by controlling the hydraulic pressure operated to the clutch, operational characteristic of the clutch can be changed, by the change in the operational characteristic, an wear amount of clutch plates can be compensated for and accordingly, it is not necessary to provide an adjuster mechanism to the clutch.

Further, a second aspect of the invention is characterized, in addition to the constitution of the first aspect, in further including a gear transmission provided between the clutch and the driving wheels such that a plurality of shift stages of gear trains are selectively established, an actuator for driving to shift the gear transmission and a control unit for controlling the actuator and the linear solenoid such that the clutch is operated to cut and connect in cooperation with shift operation of the gear transmission and according to such a constitution, the clutch can be operated to cut and connect automatically in cooperation with driving to shift the gear transmission by the actuator, the operational characteristic of the clutch can be changed and accordingly, shift shock can be alleviated.

BRIEF DESCRIPTION OF THE DRAWINGS

Preferred embodiments of the invention are shown in the drawings, wherein:

Fig. 1 is a side view of an all terrain saddle type vehicle according to a first embodiment.

Fig. 2 is a simplified enlarged view of a power unit in view from an arrow mark 2 direction of Fig. 2.

Fig. 3 is a simplified enlarged view of the power unit in view from an arrow mark 3 direction of Fig. 1.

Fig. 4 is a sectional view showing a front half portion of a section taken along a line A-A of Fig. 2.

Fig. 5 is a sectional view showing a rear half portion of the section taken along the line A-A of Fig. 2.

Fig. 6 is a sectional view taken along a line 6-6 of Fig. 2.

Fig. 7 is an enlarged sectional view taken along a line 7-7 of Fig. 6.

Fig. 8 is a disassembled perspective view of a portion of a shift change mechanism.

Fig. 9 is a hydraulic circuit diagram.

Fig. 10 is a hydraulic circuit diagram of a second embodiment in correspondence with Fig. 9.

Fig. 11 is a sectional view of a third embodiment in correspondence with Fig. 4.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

An explanation will be given of embodiments according to the present invention based on embodiments of the present invention shown in attached drawings as follows.

Fig. 1 through Fig. 9 show a first embodiment in which Fig. 1 is a side view of an all terrain saddle type vehicle, Fig. 2 is simplified enlarged view of a power unit viewed from an arrow mark 2 direction of Fig. 1, Fig. 3 is a simplified enlarged view of the power unit viewed from an arrow mark 3 direction of Fig. 1, Fig. 4 is a sectional view showing a front half portion of a section taken along a line A-A of Fig. 2, Fig. 5 is a sectional view showing a rear half portion of the section taken along the line A-A of Fig. 2, Fig. 6 is a sectional view taken along a line 6-

6 of Fig. 2, Fig. 7 is an enlarged sectional view taken along a line 7-7 of Fig. 6, Fig. 8 is a disassembled perspective view of a portion of a shift change mechanism and Fig. 9 is a hydraulic circuit diagram.

First, in Fig. 1, the all terrain saddle type vehicle is provided with a vehicle body frame 15 constituted by welding steel pipes, a pair of left and right front wheels WF respectively mounted with balloon type low pressure tires are suspended at a front portion of the vehicle body frame 15 and a pair of left and right rear wheels WR respectively mounted with balloon type low pressure tires are suspended at a rear portion of the vehicle body frame 15. Further, at a front end of the vehicle body frame 15, a steering handle 16 for steering the pair of left and right front wheels WF is arranged, a fuel tank 17 is arranged at a middle portion of the vehicle body frame 15 in the forward and rearward direction and a saddle type seat 18 is arranged on a rear side of the fuel tank and at an upper portion of the vehicle body frame 15.

On lower sides of the fuel tank 17 and the seat 18, the vehicle body frame 15 is mounted with a power unit P including an engine E for driving the front wheels WF constituting the steering wheels as well as driving wheels and the rear wheels WR constituting driving wheels. The engine E is arranged in, for example, vertically by disposing a crankshaft 19 (refer to Fig. 2 through Fig. 6) in the forward and rearward direction of the saddle type vehicle and an exhaust pipe 21 communicating with an exhaust port (not illustrated) provided on the front side of a cylinder head 20 in the engine E, is bent by 180 degree to the right side of the vehicle body frame 15, extended rearwardly on the right side of the power unit P and is connected to an exhaust muffler 22 arranged on the rear side of the rear portion of the vehicle body.

An air cleaner 23 and a carburetor 24 connected to the air cleaner 23 are arranged on the rear side of the engine E and on the lower side of the seat 18 and the carburetor 24 is connected to an intake port (not

illustrated) provided on the rear side of the cylinder head 20.

Also in reference to Fig. 2 through Fig. 5, a cylinder block 28 of the engine E is erected substantially vertically while being inclined slightly to the right side of the vehicle body and the cylinder head 20 is coupled to an upper portion of the cylinder block 28. Further, a piston 30 is slidably fitted to a cylinder liner 29 provided at the cylinder block 28.

A lower portion of the cylinder block 28 is coupled with a crankcase 33 constituted by coupling together a front case half 31 and a rear case half 32 such that a lower portion of the cylinder liner 29 is projected into the crankcase 33 and the crankshaft 19 having an axis line along the forward and rearward direction of the saddle type vehicle, is rotatably supported by the crankcase 33 and the piston 30 is connected to the crankshaft 19 via a connecting rod 34 and a crankpin 35.

The front case half 31 is coupled with a front casing 36 formed in a cylindrical shape and extended to the front side of the saddle type vehicle and a front cover 37 for closing a front end opening portion of the front casing 36 of the front casing 36 to thereby form a front operation chamber 40 between the parts and the front case half 31. Further, the rear case half 32 is coupled with a rear casing 38 forming a rear operation chamber 41 between the part and the rear case half 32 and the rear casing 38 is coupled with a rear cover 39 constituting a portion in correspondence with the crankshaft 19 and covering an opening portion 38a provided at the rear casing 38.

A ball bearing 42 is provided between the crankshaft 19 and the front case half 31 and a ball bearing 43 is provided between the crankshaft 19 and the rear case half 32. Further, a front end of the crankshaft 19 projected from the front case half 31 to a side of the front operation chambers 40 is rotatably supported by the front cover 37 via a ball bearing 44.

The power unit P is provided with the engine E, a torque converter 45 mounted to the crankshaft 19 of the engine E, a gear transmission 46 capable of selectively establishing a plurality of shift stages of gear trains G1, G2, G3 and GR and a clutch 47 in a hydraulic type capable of cutting power transmission between the gear transmission 46 and the torque converter 45 and power outputted from the gear transmission 46 is transmitted to the left and right front wheels WF via a front propeller shaft 48 extended from the power unit P in the forward direction and a front differential apparatus (not illustrated) and is transmitted to the left and right rear wheels WR via a rear propeller shaft 49 extended from the power unit P in the rearward direction and a rear differential apparatus (not illustrated).

Also in reference to Fig. 6, the gear transmission 46 is provided with an input shaft 50 and an output shaft 51 having axis lines in parallel with the crankshaft 19 and rotatably supported by the crankcase 33 and a middle shaft 52 having an axis line in parallel with the crankshaft 19 and fixed to the crankcase 33.

A ball bearing 54 is provided between the input shaft 50 and the front case half 31 and a ball bearing 55 is provided between the input shaft 50 and the rear case half 32 and a front end of the input shaft 50 is projected from the front case half 31 to a side of the front operation chamber 40. Further, a ball bearing 56 is provided between the output shaft 51 and the front case half 31, a ball bearing 57 is provided between the output shaft 51 and the rear case half 32 and a rear end of the output shaft 51 is projected from the rear case half 32 to a side of the rear operation chamber 41.

A plurality of shift stages of gear trains G1, G2, G3 and GR provided to the gear transmission 46 are contained in the crankcase 33. Further, the first speed gear train G1 is constituted of a drive gear 57 formed integrally with the input shaft 50 and a driven gear 58 supported rotatably relative to the output shaft 51 and in mesh with the

driving gear 57. The second speed gear train G2 is constituted of a driving gear 59 integrally formed with the input shaft 50 and a driven gear 60 supported rotatably relative to the output shaft 51 and in mesh with the driving gear 59. The third speed gear train G3 is constituted of a driving gear 61 fixed to the input shaft 50 and a driven gear 62 supported rotatably relative to the output shaft 51 and in mesh with the driving gear 61. Further, the rear gear train GR is constituted of the driving gear 59, a first middle gear 63 supported rotatably by the middle shaft 52 and in mesh with the driving gear 59, a second middle gear 64 integral with the first middle gear 63 and a driven gear 65 supported rotatably relative to the output shaft 51 and in mesh with the second middle gear 64.

A dog clutch member 66 for switching first speed and third speed, is spline-fitted slidably in the axial direction to the output shaft 51 between the driven gear 58 of the first speed gear train G1 and the driven gear 62 of the third speed gear train G3, when the dog clutch member 66 for switching first speed and third speed is engaged with the driven gear 58, the first speed gear train G1 is established and when the dog clutch member 66 for switching first speed and third speed is engaged with the driven gear 62, the third speed gear train G3 is established. Further, a dog clutch member 67 for switching second speed and rear, is spline-fitted slidably in the axial direction to the middle shaft 51 between the driven gear 60 of the second speed gear train G2 and the driven gear 65 in the rear gear train GR, and when the dog clutch member 67 for switching second speed and rear is engaged with the driven gear 60, the second speed gear train G2 is established and when the dog clutch member 67 for switching second speed and rear is engaged with the driven gear 65, the rear gear train GR is established.

A shift fork 68 incorporating the dog clutch member 66 for switching first speed and third speed and a shift fork 69 incorporating the dog clutch member 67 for

switching second speed and rear, are supported by a shift fork shaft 70 having an axis line in parallel with the output shaft 51 slidably in the axial direction of the shift fork shaft 70 and the shift forks 68 and 69 are respectively engaged with cam grooves 71a and 71b provided at an outer periphery of a shift drum 71.

A front end portion of the shift drum 71 is supported by the front case half 31 via a ball bearing 72 and a rear end portion of the shift drum is supported directly by the rear case half 32. Further, the shift drum 71 is pivoted by a shift change mechanism 72 connected to the front end portion of the shift drum 71 and by sliding the shift forks 68 an 69 by pivoting the shift drum 71, one of the respective gear trains G1 through G3 and GR is selectively established.

A pivotal position of the shift drum 71, that is, a shift position is detected by a shift position detector 73 comprising a potentiometer and the shift position detector 73 is fixed to the rear casing 38 and is connected to a rear end of the shift drum 71.

A driving gear 74 is fixed to a rear end of the output shaft 51 in the rear operation chamber 41. Meanwhile, a driving shaft 75 having an axis line in parallel with the output shaft 51 is rotatably supported by the crankcase 33 to interpose a ball bearing 76 between the driving shaft 75 and the front case half 31 and interpose a roller bearing 77 between the driving shaft 75 and the rear case half 32 and the driving gear 74 is in mesh with a driven gear 78 fixed to a rear portion of the drive shaft 75 in the rear operation chamber 41.

The front end of the drive shaft 75 is projected into the operation chamber 40 from the front case half 31, one end of a connecting shaft 79 is coaxially connected to the front end of the driving shaft 75 and a roller bearing 80 is interposed between the connecting shaft 79 and the front cover 37. Further, other end of the connecting shaft 79 is projected forward from the front cover 37 and the front propeller shaft 48 is connected to the other end of the

connecting shaft 79. Further, a rear end of the driving shaft 75 is projected rearward from the rear casing 38 while interposing a seal member 81 between the rear end and the casing 38 and the rear propeller shaft 49 is connected to the rear end of the driving shaft 75.

In the rear operation chamber 41, an outer periphery of the driving shaft 75 is projected with a plurality of projections 82 and the rear casing 38 is attached with a vehicle speed sensor 83 for detecting vehicle speed based on the rotational number of the driving shaft 75 by detecting the projections 82.

Also in reference to Fig. 7 and Fig. 8, the shift change mechanism 72 is provided with a shift spindle 87 having an axis line in parallel with the shift drum 71, an arm 88 fixed to the shift spindle 87, a change arm 90 fixedly attached to a collar 89 mounted rotatably relative to an outer periphery of the shift spindle 87, a regulating pin 91 fixedly attached to the front case half 31 and inserted to the change arm 90, a first torsional coil spring 92 for exerting spring force to return the change arm 90 to a neutral position, a shifter plate 93 capable of moving relative to the change arm 9 in a range restricted along the radius direction of the sheet spindle 87, a second torsional coil spring 94 exerting spring force urging the shifter plate 93 to an inner side of the radius direction, a pin plate 96 fixed to a front end of the shifter drum 71 opposed to the shifter plate 93 and implanted with six pieces of feed pins 95 and a roller 97 urged by a spring in a direction of being brought into contact with an outer periphery of the pin plate 96.

A front end of the shift spindle 87 a rear end of which is rotatably supported by the rear case half 32 and which is extended forwardly in the operation chamber 40 by rotatably penetrating the front case half 31, rotatably penetrates the front casing 37 and is supported by a gear case 98 fastened to the casing 37 via a ball bearing 99. Further, a roller bearing 100 and a seal member 101 are

interposed between the shift spindle 87 and the front casing 37.

The change arm 90 is provided with an opening portion 102 and a spring receiving portion 103 formed by folding an outer edge of the opening portion 102 and both end portions of the first torsional coil spring 92 supported by the collar 89 are brought into contact with both sides of the spring receiving portion 103 in a state in which the change arm 90 is disposed at the neutral position. Further, a front end portion 88a of the arm 88 and the regulating pin 91 are inserted into the opening portion 102 between both end portions of the first torsional coil spring 92.

When the shift spindle 87 is pivoted in either of the regular and reverse directions in a state in which the change arm 90 is disposed at the neutral position, the front end portion 88a of the arm 88 is brought into contact with either of the left and right side edges of the opening portion 102 while flexing the first torsional coil spring 92 to thereby pivot the change arm 90 in either of regular and reverse directions and the change arm 90 is pivoted until other of the left and right side edges of the opening portion 102 is brought into contact with the regulating pin 91. Further, under the state, when power operated to the shift spindle 87 becomes "null", by spring force exerted by the first torsional coil spring 92, the arm 88 and accordingly the shift spindle 87 and the change arm 90 return to the neutral position.

Guide pins 104 and 105 are respectively calked and fixed to the change arm 90 at two locations spaced apart from each other along a radius direction of the shift spindle 87. Meanwhile, the shifter plate 93 is formed with long holes 106 and 107 respectively inserted into and engaged with the guide pins 104 and 105 to prolong along the radius direction. The shifter plate 93 is supported by a face of the change arm 90 on the side of the shift drum 71 slidably along the radius direction of the shift spindle 87.

Feed claws 108 and 109 folded to the side of the shift drum 71 are provided on both sides of the shifter plate 93 and cams 110 and 111 are formed on outer sides of the feed claws 108 and 109.

The second torsional coil spring 94 is supported by the guide pin 104 which is an outer side one of the two guide pins 104 and 105 and by bringing both end portions of the second torsional coil spring 94 into contact with the two feed claws 108 and 109 of the shifter plate 93, the shifter plate 93 is urged to an inner side along the radius direction of the shift spindle 87 relative to the change arm 90.

An outer periphery of the pin plate 96 is formed in a star shape having six of recess portions 96a spaced apart from each other at equal intervals in the peripheral direction and the six feed pins 95 are implanted to the pin plate 96 such that two of the feed pins 95 are arranged between the two feed claws 108 and 109 of the shifter plate 93.

The roller 97 is axially supported by other end of an arm 113 one end of which is pivotably supported by the front case half 31 via a support shaft 112 and by spring force of a third torsional coil spring 114 provided between the front case half 31 and the arm 113, the arm 113 urges the roller 97 in a direction of being brought into contact with the outer periphery of the pin plate 96.

According to the shift change mechanism 72, by pivotal movement of the change arm 90 in either of regular and reverse directions in accordance with pivotal movement in either of regular and reverse directions of the shift spindle 87, one of the two feed claws 108 and 109 of the shifter plate 93 is engaged with one of the feed pins 95 of the pin plate 93 and the shift drum 71 is driven to pivot in either of regular and reverse directions. Further, when the change arm 90 returns to the neutral position by spring force exerted by the first torsional coil spring 92, by bringing one of the feed pins 95 of the pin plate 93 into contact with one of the cams 110 and 111 formed at the two

feed claws 108 and 109 of the shifter plate 93, the shifter plate 93 is moved to an outer side in the radius direction of the shift spindle 87 against spring force of the second torsional coil spring 94 and one of the two feed claws 108 and 109 rides over one of the feed pins 95. Further, the pivotal movement position of the pin plate 93 is held by engaging the roller 97 with one of the recess portions 86a of the outer periphery of the pin plate 96.

Paying attention particularly to Fig. 6, the shift spindle 87 of the shift change mechanism 72 is connected with an electric motor 118 rotatable in either of regular and reverse directions via a reduction gear train 119.

The electric motor 118 is provided with an axis line in parallel with the shift spindle 87 and attached to the gear case 98 and the reduction gear train 119 is contained in a gear chamber 120 formed between the front cover 37 and the gear case 98.

The reduction gear train 119 is constituted of a first reduction gear 121 provided to an output shaft of the electric motor 118, a second reduction gear 122 in mesh with the first reduction gear 121, a third reduction gear 123 rotated integrally with the second reduction gear 122, a fourth reduction gear 124 in mesh with the third reduction gear 123, a fifth reduction gear 125 rotated integrally with the fourth reduction gear 124 and a sixth reduction gear 126 in mesh with the fifth reduction gear 125 and pivotally moved along with the shift spindle 87.

The second and the third reduction gears 122 and 123 are provided integrally with a first speed reduction shaft 127 rotatably supported by the front cover 37 and the gear case 98 and the fourth and the fifth reduction gears 124 and 125 are provided integrally with a second speed reduction gear 128 rotatably supported by the front cover 37 and the gear case 98. Further, the sixth reduction gear 126 is a sector gear and is fixed to the shift spindle 87.

The gear case 98 is attached with rotational phase detecting means 129 which is a potentiometer and the

rotational phase detecting means 129 is connected to a front end of the shift spindle 87.

Paying attention particularly to Fig. 4 and Fig. 6, the clutch 47 is mounted to one end portion, that is, a front end portion of the input shaft 50 projected from the crankcase 33 into the front operation chamber 40, and is provided with a clutch center 130 rotated around the axis line of the input shaft 50, a plurality of sheets of first clutch plates 131 in mesh with an outer periphery of the clutch center 130 relatively unrotatably, a plurality of sheets of second clutch plates 132 arranged to overlap the first clutch plates 132, a clutch drum 133 containing the first and the second clutch plates 131 and 132 and rotated integrally with the input shaft 50 by being brought in mesh with the outer peripheries of the second clutch plates 132 relatively unrotatably and a piston 134 slidably fitted to the clutch drum 133 by enabling to be engaged frictionally with the first and the second clutch plates 131 and 132.

An outer periphery of a front end portion of the input shaft 50 is fixed with a sleeve 135 coaxially surrounding the input shaft 50, the clutch center 130 is supported relatively rotatably to the sleeve 135 and the clutch drum 133 is fixed to the sleeve 135. The piston 134 is slidably fitted to the clutch drum 133 by being brought into contact with one of the second clutch plates 132 and a hydraulic chamber 136 is formed between the clutch drum 133 and the piston 134. Further, a return spring 137 is provided between the clutch drum 133 and the piston 134 and the piston 134 is urged by spring to a side of reducing the volume of the hydraulic chamber 136 by the return spring 137.

In the clutch center 130, a projected end portion of the clutch drum 133 is connected with a driven gear 139 to which power from the torque converter 45 is transmitted via a buffer spring 138 and a first one way clutch 140 capable of transmitting power from the input shaft 50 to the clutch center 130 is provided between the sleeve 135 fixed to the input shaft 50 and the clutch center 50 at a position in

correspondence with a position between the clutch drum 133 and the driven gear 139 along the axis line direction of the input shaft 50.

The torque converter 45 is mounted to one end portion, that is, a front end portion of the crankshaft 19 projected from the crankcase 33 to the side of the front operation chamber 40 and the torque converter 45 is mounted to the crankshaft 19 at a position in correspondence with a position between the front cover 37 and the clutch 47 along the axis line direction of the crankshaft 19. Further, a distance between the input shaft 50 of the gear transmission 46 and the crankshaft 19 is set such that a portion of the clutch 47 overlaps the torque converter 45 in view from the axis line direction of the crankshaft 19.

The torque converter 45 is constructed by a conventionally well-known constitution having a pump 141, a stator 142 and a turbine 143, the pump 141 is fixed to the crankshaft 19, a stator shaft 144 in a cylindrical shape coupled to the stator 142 and coaxially surrounding the crankshaft 19, is supported rotatably relative to the crankshaft 19 and a turbine shaft 145 in a cylindrical shape coupled to the turbine 143 and coaxially surrounding the stator shaft 144 is supported rotatably relative to the stator shaft 144.

The turbine shaft 145 is fixedly attached with a driving gear 146 and the driving gear 146 is brought in mesh with the driven gear 139 connected to the clutch center 130 of the clutch 47.

A second one way clutch 148 permitting the stator shaft 144 and the stator 142 to rotate in one direction, is provided between a support member 147 fixed to the front case half 31 of the crankcase 33 and the stator shaft 144 and the driving gear 146 is arranged between the second one way clutch 148 and the torque converter 45 along the axial direction of the crankshaft 19. Further, the clutch 47 is mounted to the front end portion of the input shaft 50 in an arrangement in which at least a portion of the clutch drum 133 provided to the clutch 47 is contiguous to a side

of the second one way clutch 148 along the axis line direction of the crankshaft 19.

Further, a third one way clutch 150 capable of transmitting power from the turbine shaft 145 to the pump 141 is provided between the pump 141 of the torque converter 45 and the turbine shaft 145 fixed with the driving gear 146.

Paying attention particularly to Fig. 5, a rotor 152 of a generator 151 is attached to a rear end of the crankshaft 19 projected from the rear case half 32 of the crankcase 33 into the rear operation chamber 41 and a stator 153 of the generator 151 is fixed to the rear cover 39.

A starter gear 154 connected to a starter motor, not illustrated, is supported relatively rotatably by the crankshaft 19 at a position contiguous to the rotor 152 of the generator 151 and the starter gear 154 is connected to the rotor 152 via a fourth one way clutch 155. Thereby, when the starter gear 154 is driven by operating the starter motor, the crankshaft 19 can be cranked via the fourth one way clutch 155 and the rotor 152 and when the engine E is started by the cranking operation, the fourth one way clutch 155 is brought into a free state and transmission of power from the rotor 152 to the starter gear 154 is cut.

A terminal end of the crankshaft 19 is fixedly attached with a starter wheel 156 and a recoil starter 157 claws of which can be engaged with the starter wheel 156 is attached to the rear cover 39. Therefore, the crankshaft 19 can be cranked also by operating to pull a rope 158 of the recoil starter 157.

A plurality of projections 180 are projected from the outer periphery of the rotor 152 and an engine rotational number sensor 181 for detecting a rotational number of the crankshaft 19 by detecting the projections 180 is attached to the rear cover 39 to be opposed to the projections 180.

In Fig. 9, oil stored to an oil storage 159 formed at a bottom portion in the crankcase 33 is sucked by a

scavenging pump 161 driven by the crankshaft 19 via strainer 160 and oil delivered from the scavenging pump 161 is returned to the oil storage 159 via an oil cooler 162 and a first relief valve 163 is connected between the scavenging pump 161 and the oil cooler 162.

The hydraulic type clutch 47 is driven by a second actuator 164 which is independent from the electric motor 118 for driving to shift the gear transmission 46 and the second actuator 164 is provided with a hydraulic pump 165 driven by the crankshaft 19 and a clutch control valve 166 for controlling delivery pressure of the hydraulic pump 165 to operate to the hydraulic chamber 136 of the clutch 47.

Oil stored in the oil storage 159 is sucked by the hydraulic pump 165 via the strainer 160 and oil delivered from the hydraulic pump 165 is supplied to the clutch control valve 166 via a second relief valve 167 and an oil filter 168.

The clutch control valve 166 is provided with a valve housing 171 having an input port 169 and an output port 170, a spool 172 slidably fitted to the valve housing 171 to be able to switch to communicate and cut the input port 169 and the output port 170, a spring 173 provided between the spool 172 and the valve housing 171 and a linear solenoid 174 coaxially connected to the spool 172, the input port 169 is connected to the oil filter 168 and the output port is connected to the hydraulic chamber 136 of the clutch 47.

The valve housing 171 is provided with a slide hole 175 both ends of which are opened, the spring 173 is provided between a cap 176 closing one end of the slide hole 175 and one end of the spool 172 slidably fitted to the slide hole 175 and a rod 174a of the linear solenoid 174 fastened to the valve housing 171 to close other end of the slide hole 175 is brought into contact coaxially with other end of the spool 172.

The spring 173 urges the spool 172 to a side of communicating the input port 169 and the output port 170 and thrust force from the linear solenoid exerting thrust

force in accordance with input current is exerted from the rod 174a to the spool 172 to move the spool 172 to a side of cutting the input port 169 from the output port 170.

Further, ring-like pressure chambers 176 for applying pressure of the output port 170 are formed between the valve housing 171 and the spool 172 and a pressure receiving area of the spool 172 facing the pressure chamber 176 from the side of the linear solenoid 174 is set to be smaller than a pressure receiving area of the spool 172 facing the pressure chamber 172 from the side of the spring 174. Therefore, hydraulic pressure of the pressure chamber 176 is operated to the spool 172 to move the spool 172 to the side of communicating the input port 169 and the output port 170.

According to the clutch control valve 166, hydraulic pressure of the hydraulic pump 165 is controlled and outputted from the output port 170 such that the larger the input current of the linear solenoid 174, the lower the pressure.

As shown by Fig. 4, the valve housing 171 of the clutch control valve 166 is attached to an outer face side of the front cover 37 at a position substantially in correspondence with the clutch 47, there is provided an oil supply tube 179 for connecting an oil path 177 communicated with the hydraulic chamber 136 of the clutch 47 and coaxially provided to the front end portion of the input shaft 50 and an oil path 178 provided to the front cover 37 to communicate with the output port 170 of the clutch control valve 166 between the front end portion of the input shaft 50 and the front cover 37 and the oil supply tube 179 is arranged to extend coaxially with the input shaft 50 at the side of the torque converter 45.

The output port 170 of the clutch control valve 166 is connected with an accumulator 183 and the accumulator 183 is provided with a fixed housing 184, an accumulator piston 186 forming an accumulating chamber 185 in a ring-like shape communicating with the output port 170 between the housing 184 and the piston and slidably fitted to the

housing 184 and a spring 187 provided between the housing 184 and the accumulator piston 186 for exerting spring force to a side of increasing the volume of the accumulating chamber 185. Further, a pilot chamber 188 capable of exerting hydraulic force urging the accumulator piston 186 to a side of reducing the volume of the accumulating chamber 185, is formed between the accumulator piston 186 and the housing 184 and the pilot chamber 188 is connected to the output port 170 of the clutch control valve 166 via a first orifice 189.

Such accumulator 183 is effectively used in a vehicle having large generated torque, a change in control hydraulic pressure by the clutch control valve 166 can be alleviated by the accumulator 183 and shift shock of the clutch 47 can be alleviated.

Further, the output port 170 of the clutch control valve 166 is connected with an opening and closing valve 190 which is opened in accordance with lowering of hydraulic pressure of the output port 170 to thereby discharge oil in the hydraulic chamber 136 at the clutch 47 to the oil storage 159 and by operating the opening and closing valve 190, the clutch 47 can be swiftly operated to the side of cutting power transmission.

An outlet of the oil filter 168 is connected to the input port of the clutch control valve 166 and also connected to an input port 192 of a torque converter control valve 191.

The torque converter control valve 191 is provided with a valve housing 194 having the input port 192 and an output port 193, a spool 195 slidably fitted to the valve housing 194 to be capable of switching to communicate and cut the input port 192 and the output port 193 and a spring 196 provided between the valve housing 194 and one end of the spool 195 for exerting spring force urging the spool 195 in a direction of communicating the input port 192 and the output port 193. Further, a pilot chamber 198 for exerting hydraulic pressure force for urging the spool 195 in a direction of cutting the input port 192 from the

output port 193, is formed between a cap 197 mounted to the valve housing 194 and other end of the spool 195 and the pilot chamber 198 is communicated with the input port 192.

According to the torque converter control valve 191, outputted hydraulic pressure of the hydraulic pump 167 is controlled constant and is outputted from the output port 193.

As shown by Fig. 4, the valve housing 194 of the torque converter control valve 191 is attached to the front cover 37 at a position substantially in correspondence with the crankshaft 19 and the output port 193 is connected to an oil path 199 provided in the crankshaft 19 to communicate with the inside of the torque converter 45 via a second orifice 200 provided in the cover 37. Further, the crankshaft 19 is provided with an oil path 201 for guiding oil discharged from the torque converter 45 to the side of the crankpin 35.

An oil path 202 communicating with the output port 193 of the torque converter control valve 191 is provided at the front cover 137 and the front casing 36 and a third orifice 203 interposed between the front cover 37 and the front casing 36 is interposed in the oil path 202. Further, the oil path 202 is connected to an oil path (not illustrated) in the cylinder head 20 via a fourth orifice 204 and is connected to an oil path 205 (refer to Fig. 5) provided at the rear casing 38 and the oil path 205 is connected to an oil path 206 provided at the output shaft 51 of the gear transmission 46 via a fifth orifice 207. Further, oil in the oil path 202 is supplied into the clutch 47 via a sixth orifice 208.

The electric motor 118 for driving to shift the gear transmission 46 and the linear solenoid 174 of the clutch control valve 166 are controlled by a control unit 208 to operate to cut and connect the clutch 47 in cooperation with shift operation of the gear transmission 46, and the control unit 208 is inputted with a detection signal of a select position detector 210 for detecting a position of operating a select lever 209 operable to switch the shift

position to four of "L", "D", "N" and "R", an ON/OFF signal of a power button 211 operable to switch the shift stage of the gear transmission 46 when the select lever 209 is disposed to "D", a detection signal of a throttle sensor 212, and detection signals of a shift position detector 73, a vehicle speed sensor 83, the rotational phase detecting means 129 and the engine rotational number sensor 181.

Further, according to the control unit 208, the electric motor 118 is controlled to establish the first speed shift stage G1 of the gear transmission 46 when "L" position is selected by the select lever 209, the electric motor 118 is controlled to establish the second speed shift stage G2 when "D" position is selected by the select lever 209 and the power button 211 is brought into an OFF state, the electric motor 118 is controlled to establish the third speed shift stage G3 of the gear transmission 46 when "D" is selected by the select lever 209 and the power button 211 is brought into an ON state, the electric motor 118 is controlled not to establish any of the shift stages G1 through G3, and GR when "N" position is selected by the select lever 209 and the electric motor 118 is controlled to establish the rear gear train GR by the gear transmission 46 when "R" position is selected by the select lever 209.

Further, although the control unit 208 controls the linear solenoid 174 such that the clutch 47 is brought into a power transmitting state when "L", "D" and "R" positions are selected by the select lever 209, when "N" position is selected by the select lever 209, the control unit 208 controls the linear solenoid 174 such that the clutch 47 is brought into a power cut state.

Further, in the case in which the clutch 47 is switched from the power cut state to the power transmitting state when the select lever 209 is operated to switch selection of respective positions of "L", "D", "N" and "R", input current to the linear solenoid 174 and accordingly, hydraulic pressure of the clutch 47 is controlled in consideration of detection signals of the throttle sensor

212, the shift position detector 73, the vehicle speed sensor 83 and the engine rotational number sensor 181.

Next, explaining operation of the first embodiment, whereas the gear transmission 46 is driven to shift by the electric motor 118, the clutch 47 is driven by the actuator 164 which is independent from the electric motor 118 and accordingly, in comparison with a constitution in which the gear transmission 46 and the clutch 47 are mechanically connected by a link mechanism or the like, even when load of the clutch 47 is increased with an increase in the displacement of the engine E, it is not necessary to increase rigidity of a link mechanism or the like and an increase in the weight can be avoided. Further, the load of the clutch 47 does not effect influence on shift load or shift operation stroke and therefore, the shift operability can be promoted.

Further, according to the gear transmission 46, there is provided the input shaft 50 supported rotatably by the crankcase 33, the clutch 47 operated by hydraulic pressure is mounted to the front end portion of the input shaft 50 projected from the crankcase 33, the clutch control valve 166 constituting a portion of the actuator 164 is provided at the front cover 37 covering the clutch 47 and the torque converter 45 and coupled to the crank case 33 and accordingly, the clutch 47 and the clutch control valve 166 can be arranged at positions proximate to each other and the operational response of the clutch 47 by operation of the clutch control valve 166 can be promoted.

Further, the torque converter 45 is provided at the front end portion of the crankshaft 19, the torque converter control valve 191 for controlling hydraulic pressure operated to the torque converter 45 is provided at the front cover 37 and therefore, both of the clutch control valve 166 and the torque converter control valve 191 are provided at the front cover 37, thereby, pressure leakage inspection at the two control valves 166 and 191 can be carried out concentratedly on the side of the front

cover 37, which can contribute to promotion of productivity.

Further, the clutch control valve 166 is constituted to be capable of controlling output hydraulic pressure from the output port 170 communicating with the hydraulic chamber 136 of the clutch 47 at a value in correspondence with input current to the linear solenoid 174 and accordingly, hydraulic pressure operated to the clutch 47 can be controlled by controlling the input current to the linear solenoid 47. Therefore, by controlling hydraulic pressure operated to the clutch 47, the operational characteristic of the clutch 47 can be changed, by changing the operational characteristic, wear amounts of the first and the second clutch plates 131 and 132 can be compensated for and therefore, it is not necessary to provide an adjustor mechanism to the clutch 47.

Further, operation of the linear solenoid 174 provided to the clutch control valve 166 and operation of the electric motor 118 are controlled by the control unit 208, by electrically controlling both of the clutch control valve 166 and the electric motor 118, operation of the clutch 47 in cooperation with shift operation of the gear transmission 46 is facilitated, the clutch 47 can automatically be operated to cut and connect in cooperation with driving to shift the gear transmission 46 by the electric motor 118 and the operational characteristic of the clutch 47 can be changed and accordingly, the shift shock at the clutch 47 can be alleviated.

Further, the clutch 47 arranged at a position shifted from the torque converter 45 along the axis line direction of the crankshaft 19 is mounted at the front end portion of the input shaft 50 to overlap a portion thereof to the torque converter 45 in view from the axis line direction of the crankshaft 19 and therefore, the power transmission apparatus constituted to contain the gear transmission 46 in the crankcase 33 can be made compact in a direction along the axis line of the crankshaft 19. Further, the clutch 47 can be arranged to be proximate to the side of

the crankshaft 19 and the power transmission apparatus can also be restrained from being large-sized in a direction orthogonal to the axis line of the crankshaft 19.

The oil supply tube 179 extended coaxially with the input shaft 50 at the side of the torque converter 45 is provided between the front cover 37 coupled to the crankcase 33 by covering the torque converter 45 and the clutch 47 and the one end portion of the input shaft 50 and a system of supplying operating oil to the clutch 47 can be constituted by effectively utilizing a space produced at the side of the torque converter 45.

Further, the clutch 47 is provided with the clutch center 130 connected with the driven gear 139, the first clutch plates 131 in mesh with the outer periphery of the clutch center 130 relatively unrotatably, the second clutch plates 132 arranged to overlap the first clutch plates 131, the clutch drum containing the first and the second clutch plates 131 and 132, in mesh with the outer peripheries of the second clutch plates 132 relatively unrotatably and rotated integrally with the input shaft 50 and the piston 134 capable of exerting the hydraulic force for frictionally engaging the first and the second clutch plates 131 and 132 and slidably fitted to the clutch drum 133, and the clutch 47 having the clutch drum 133 having a comparatively large diameter can effectively be arranged such that the driven gear 139 and the clutch drum 133 do not interfere with the crankshaft 19 at the position contiguous to the torque converter 45 along the axis line of the crankshaft 19.

Further, the first one way clutch 140 capable of transmitting power from the input shaft 50 to the clutch center 130 is provided between the clutch center 130 and the input shaft 50 and accordingly, power from the side of the gear transmission 46 can be transmitted to the side of the torque converter in braking the engine and power from the side of the gear transmission 46 in braking the engine can be transmitted to the crankshaft 18 by detouring the torque converter 45 by the third one way clutch 150.

Further, the second one way clutch 148 for permitting to rotate the stator 142 of the torque converter 45 in one direction is provided between the stator 142 and the crankcase 33, the driving gear 146 rotated integrally with the turbine 143 of the torque converter 45 is arranged between the second one way clutch 148 and the torque converter 45 along the axial direction of the crankshaft 19 and the clutch 47 is mounted to the front end portion of the input shaft 50 in the arrangement in which at least a portion of the clutch drum 133 is contiguous to the side of the second one way clutch 148 along the axis line direction of the crankshaft 19. Therefore, the efficiency of the torque converter 45 can be promoted by permitting idling of the stator 142 by the second one way clutch 148 and the second one way clutch 148 having a comparatively small diameter can effectively be arranged at the space produced between the clutch drum 133 and the crankshaft 19 in correspondence with at least a portion of the clutch drum 133.

Fig. 10 shows a second embodiment of the present invention, in applying the present invention to a vehicle having a comparatively small generated torque, the accumulator 183 and the opening and closing valve 150 can be omitted and in place of the torque converter control valve 191, a sixth orifice 215 can be used.

Fig. 11 shows a third embodiment of the present invention in which in place of the torque converter 45 according to the first and the second embodiments, a centrifugal type starting clutch 220 is mounted to the crankshaft 19.

The power unit P' is provided with the engine E, the starting clutch 220 mounted to the crankshaft 19 of the engine E, the gear transmission 46 and the hydraulic pressure type clutch 47 capable of cutting power transmission between the gear transmission 46 and the centrifugal clutch 220.

A front cover 221 is coupled to the front case half 31 of the crankcase 33 to thereby form a front operation

chamber 40' between the front case half 31 and the front cover 221 and the front end of the crankshaft 19 projected from the front case half 31 to a side of the front operational chamber 40' is rotatably supported by the front cover 221 via a ball bearing 222.

The input shaft 50 of the gear transmission 46 is projected from the crankcase 33 to a side of the front operation chamber 40' and the clutch 47 is mounted to one end portion, that is, a front end portion of the input shaft 50 in the front operation chamber 40'. Further, in the clutch center 130 of the clutch 47, the projected end portion from the clutch drum 133 is connected with a driven gear 139' to which power from the centrifugal clutch 220 is transmitted via the buffer spring 138 and the one way clutch 140 enabling to transmit power from the input shaft 50 to the clutch center 130, is provided between the sleeve 135 fixed to the input shaft 50 and the clutch center 130.

The starting clutch 220 is constructed by a conventionally well-known constitution having a clutch inner 223 fixed to the crankshaft 19, a clutch outer 224 capable of rotating relative to the crankshaft 19 and a plurality of centrifugal weights 225 capable of being frictionally engaged with an inner peripheral face of the clutch outer 224 and supported by the clutch inner 223 and a driving gear 146' in mesh with the driven gear 139' connected to the clutch 47, is fixed to the clutch outer 224.

Further, a fifth one way clutch 226 for transmitting power from the clutch 47 to the crankshaft 18 in braking the engine, is provided between the clutch inner 223 and the clutch outer 224.

Further, the valve housing 171 of the clutch control valve 166 for controlling hydraulic pressure of the clutch 47 is attached to an outer face side of the front cover 221 at a position substantially in correspondence with the clutch 47 and a power supply tube 179' extended coaxially with the input shaft 50 for introducing hydraulic pressure controlled by the clutch control valve 166 to the clutch

47, is provided over an interval between the front end portion of the input shaft 50 and the front cover 221.

Also according to the third embodiment, the clutch control valve 166 for controlling hydraulic pressure of the clutch 47 is constituted to be capable of controlling the output hydraulic pressure in accordance with the input current to the linear solenoid 174 (refer to Fig. 9, Fig. 10) and therefore, it is not necessary to provide an adjuster mechanism to the clutch 47.

Although a detailed description has been given of the embodiments of the present invention as mentioned above, the present invention is not limited to the above-described embodiments but various design change can be carried out without deviating from the present invention described in the scope of claims.

For example, the present invention is applicable not only to the all terrain saddle type vehicle but also widely in relation to a power transmission apparatus of a vehicle having a hydraulic pressure type clutch capable of switching to transmit and cut power between a crankshaft and driving wheels and a clutch control valve for controlling hydraulic pressure operated to the clutch.

Further, although according to the above-described embodiments the linear solenoid 174 of the clutch control valve 166 is controlled by the control unit 208 in accordance with operation of the electric motor 118 for driving to shift the gear transmission 46, the present invention is applicable also to a power transmission apparatus in which the gear transmission 46 is operated to shift by a manual operation and in that case, a detector for detecting displacement of a manual shift operation member may be provided and the linear solenoid 174 of the clutch control valve 166 may be controlled based on a detection signal of the detector.

As described above, according to the first aspect, by controlling hydraulic pressure operated to the clutch, the operational characteristic of the clutch can be changed and by the change in the position characteristic, the wear

amount of the clutch plates can be compensated for and therefore, it is not necessary to provide an adjustor mechanism to the clutch.

Further, according to the second aspect, the clutch can automatically be operated to cut and connect in cooperation with driving to shift the gear transmission by the actuator, the operational characteristic of the clutch can be changed and therefore, shift shock can be alleviated.

JJ-10 976CA

THE EMBODIMENTS OF THE INVENTION IN WHICH AN EXCLUSIVE PROPERTY OR PRIVILEGE IS CLAIMED ARE DEFINED AS FOLLOWS:

1. A power transmission apparatus of a vehicle characterized in that in a power transmission apparatus of a vehicle having a hydraulic pressure type clutch (47) capable of switching to transmit and cut power between a crankshaft (19) of an engine (E) and driving wheels (WF, WR) and a clutch control valve (166) for controlling hydraulic pressure operated to the clutch (47), the clutch control valve (166) comprises a valve housing (171) having an input port (169) connected to a hydraulic pump (165) and an output port (170) connected to the clutch (47), a spool (172) capable of switching to communicate and cut between the input port (169) and the output port (170) and slidably fitted to the valve housing (171) and a linear solenoid (174) coaxially connected to the spool (172) and is constituted such that output hydraulic pressure from the output port (170) can be controlled to a value in correspondence with input current to the linear solenoid (174).

2. A power transmission apparatus of a vehicle according to Claim 1, characterized in further including a gear transmission (46) provided between the clutch (47) and the driving wheels (WF, WR) such that a plurality of shift stages of gear trains (G1, G2, G3, GR) are selectively established, an actuator (118) for driving to shift the gear transmission and a control unit (208) for controlling the actuator (118) and the linear solenoid (174) such that the clutch (47) is operated to cut and connect in cooperation with shift operation of the gear transmission (46).

図 1

Fig. 1

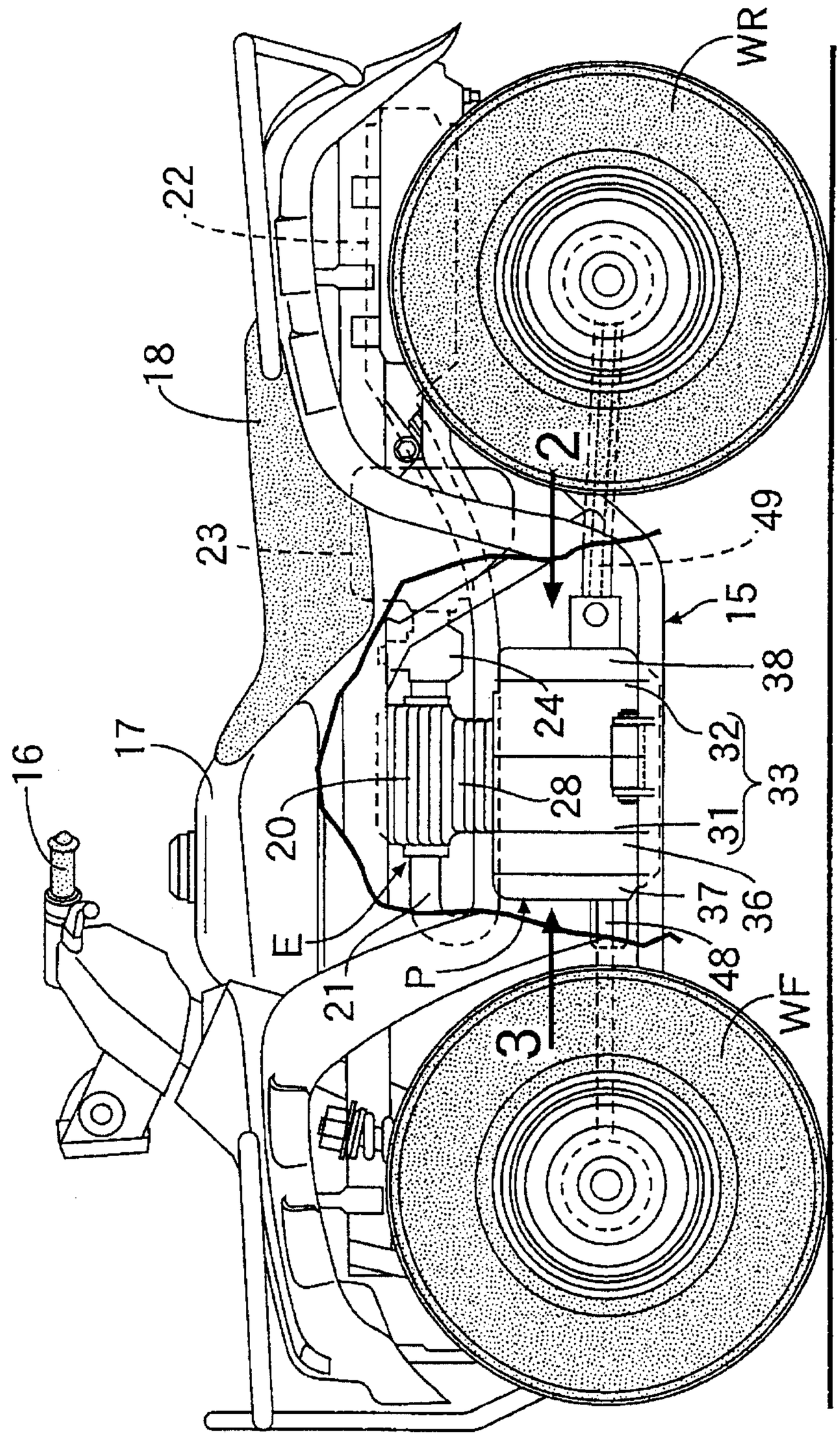


図 2
Fig. 2

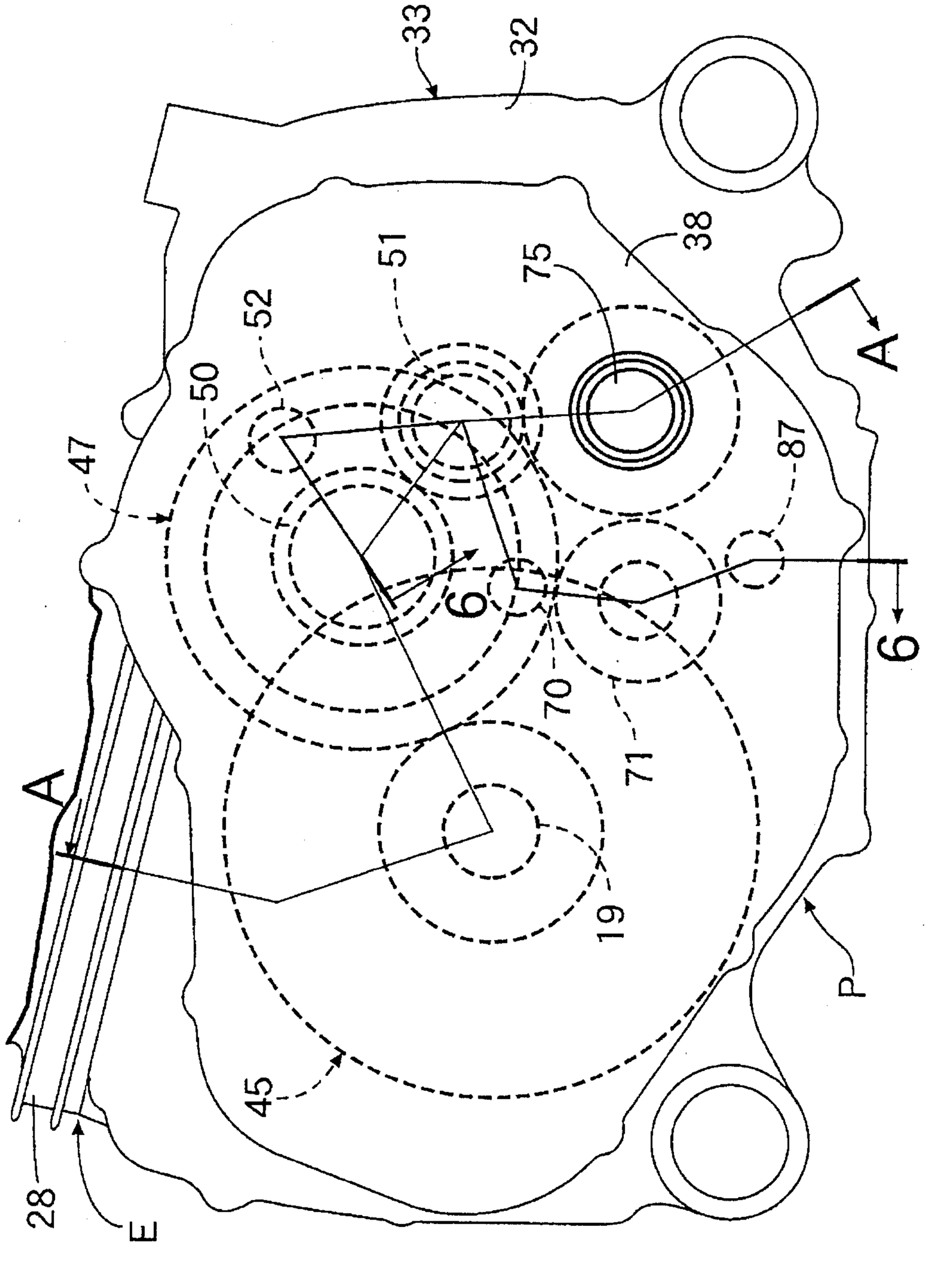


図 3
Fig. 3

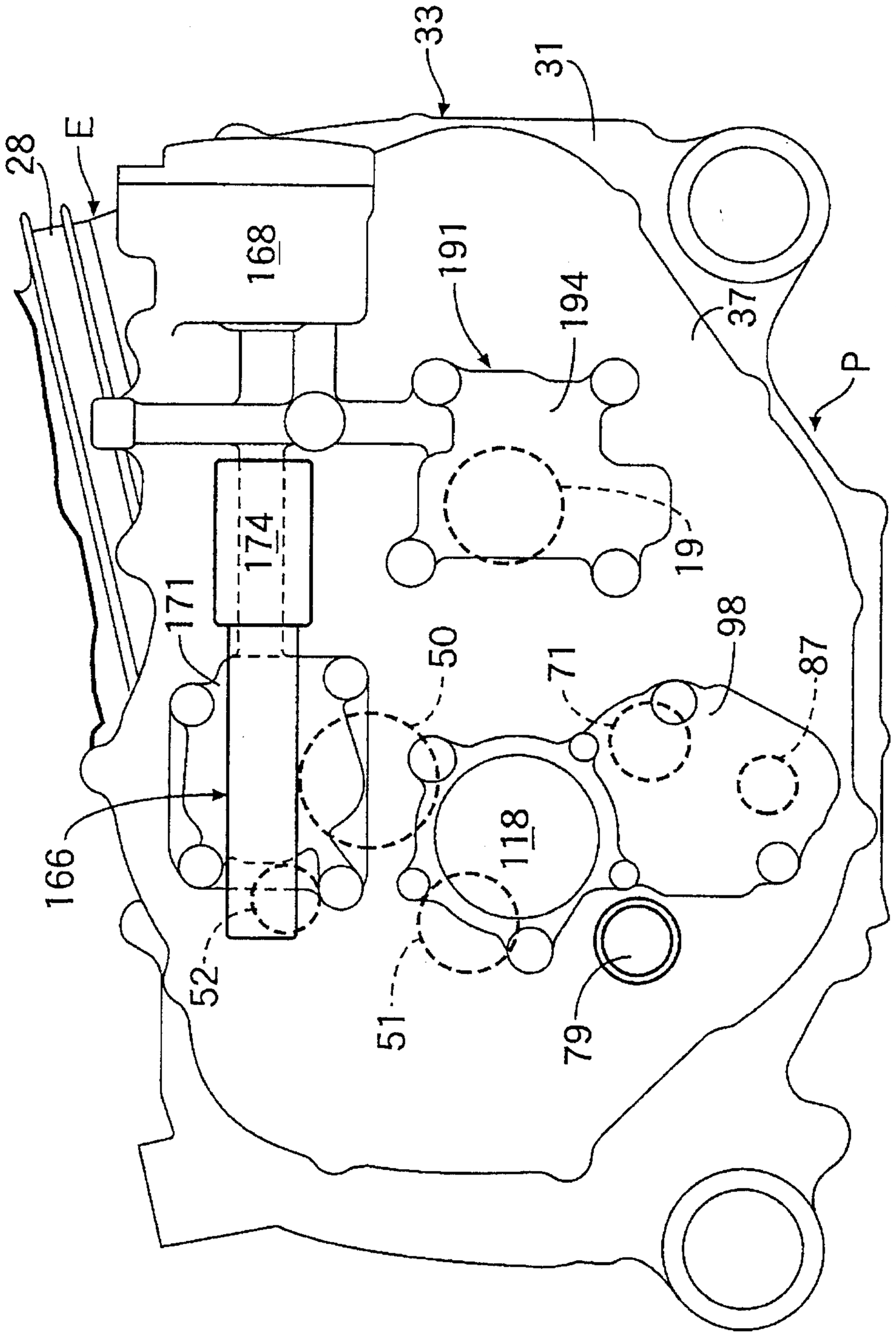


図 4

Fig. 4

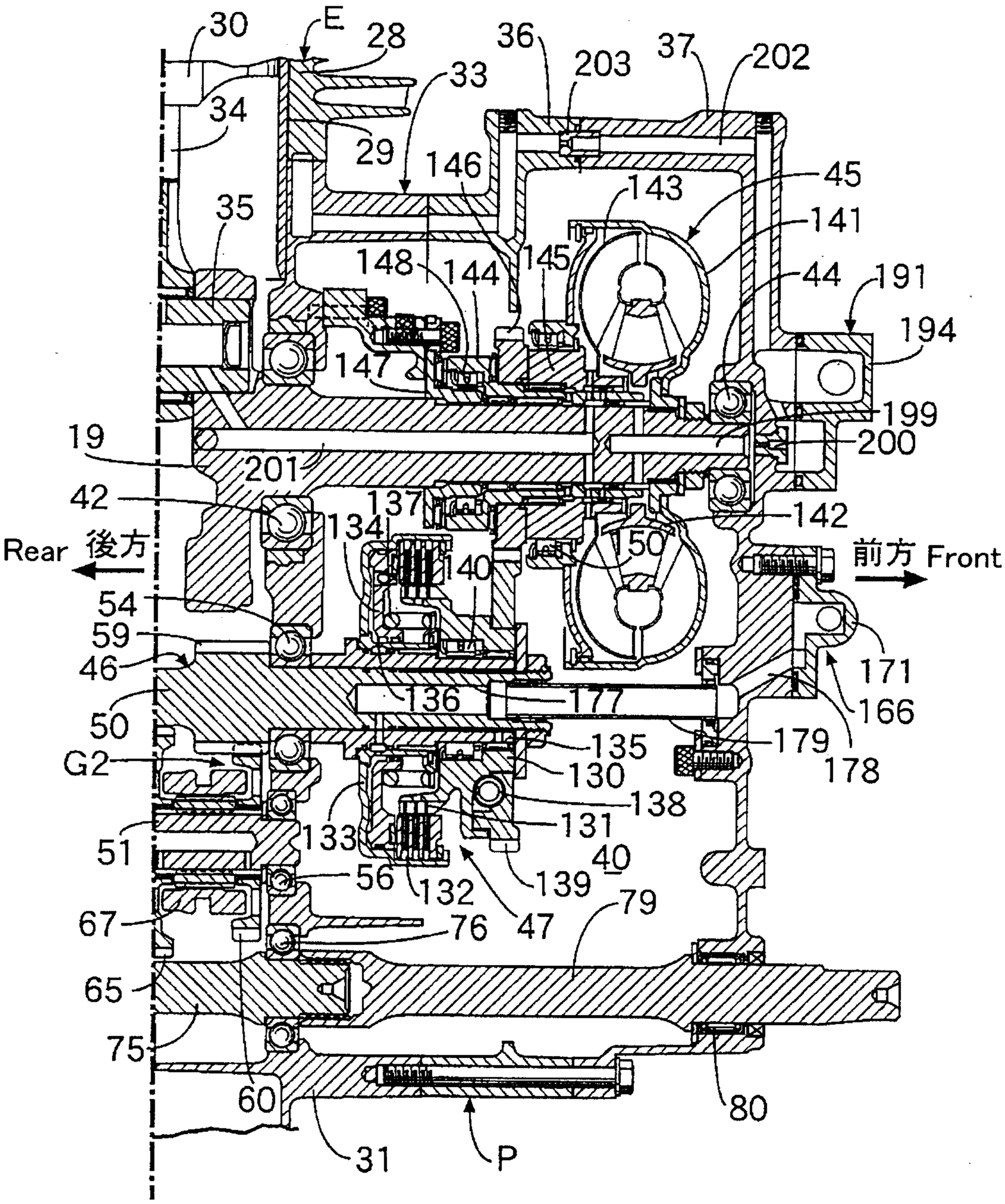

30
E
28
36
203
37
202
33
34
29
146
45
143
35
141
148
144
145
44
191
194
147
199
19
200
201
42
137
142
134
150
Rear 後方
前方 Front
140
54
59
46
171
136
177
166
50
135
179
G2
130
178
138
131
133
40
51
56
132
139
79
67
76
47
65
75
60
80
31
P

図 5

Fig. 5

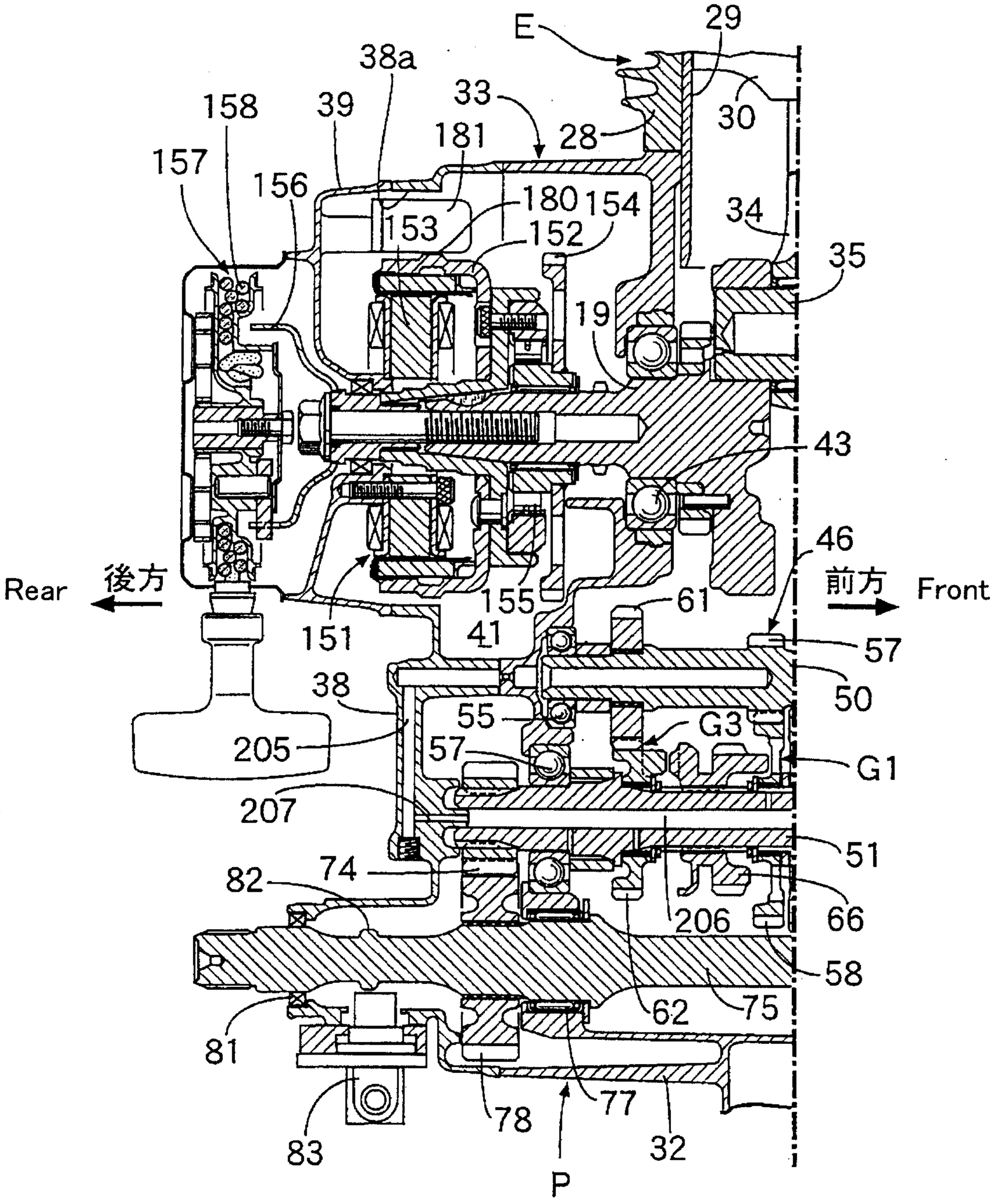

E
29
38a
33
158
39
181
30
28
157
156
180
154
153
152
34
35
19
43
46
Rear
後方
前方
Front
155
61
151
41
57
38
50
55
G3
205
57
G1
207
51
74
82
206
66
58
62
75
81
78
77
32
83
P

図 6

Fig. 6

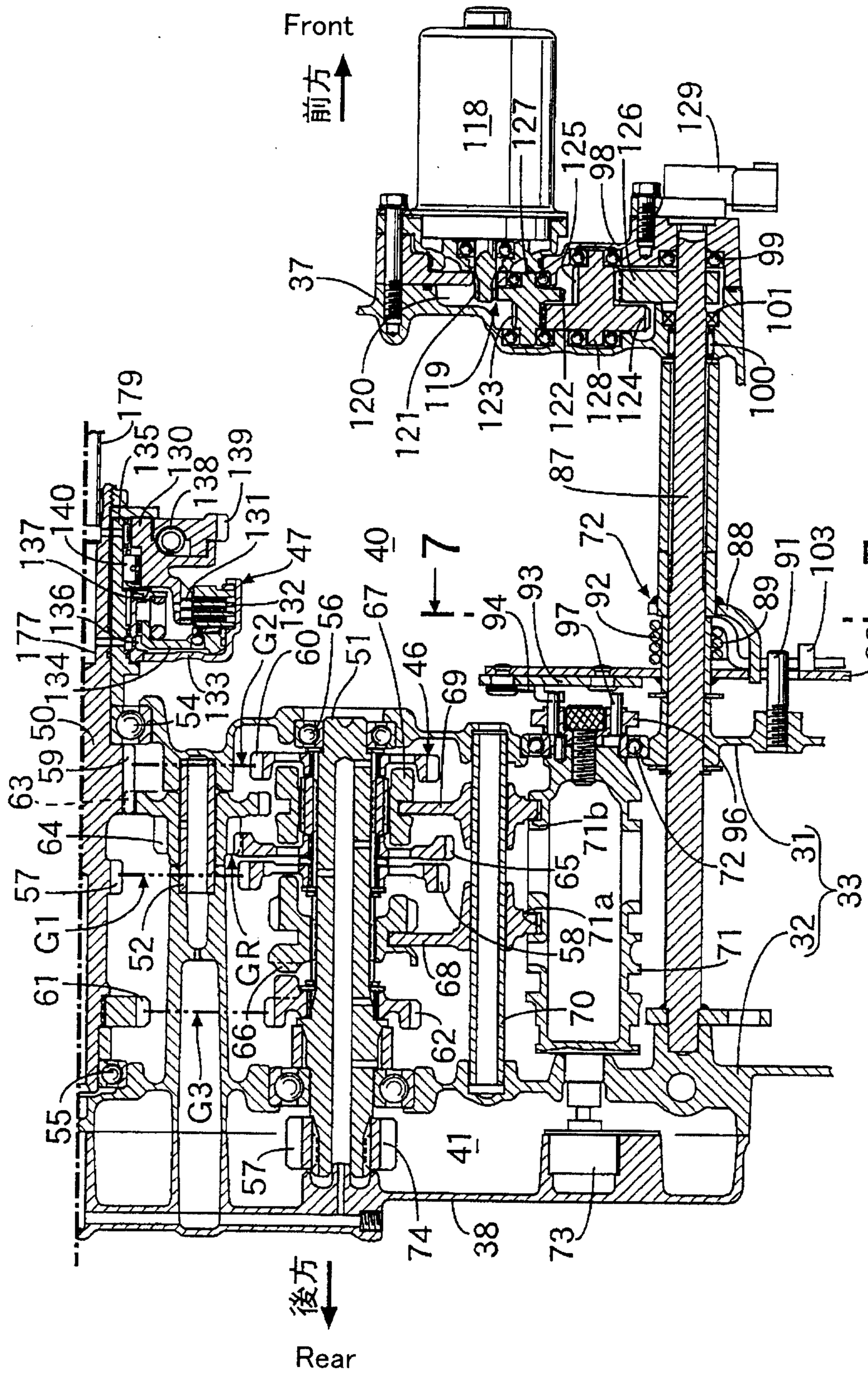

Front
前方
後方
Rear
7
7
118
127
125
98
126
129
37
120
121
119
123
122
128
124
99
101
100
87
179
135
130
138
139
131
140
137
177
136
134
47
132
G2
60
56
51
67
40
46
69
94
93
97
92
72
88
89
91
103
90
50
54
133
63
59
64
57
G1
61
52
55
G3
GR
66
68
62
70
58
65
71b
71a
71
72
96
31
32
33
41
57
74
38
73

図 7

Fig. 7

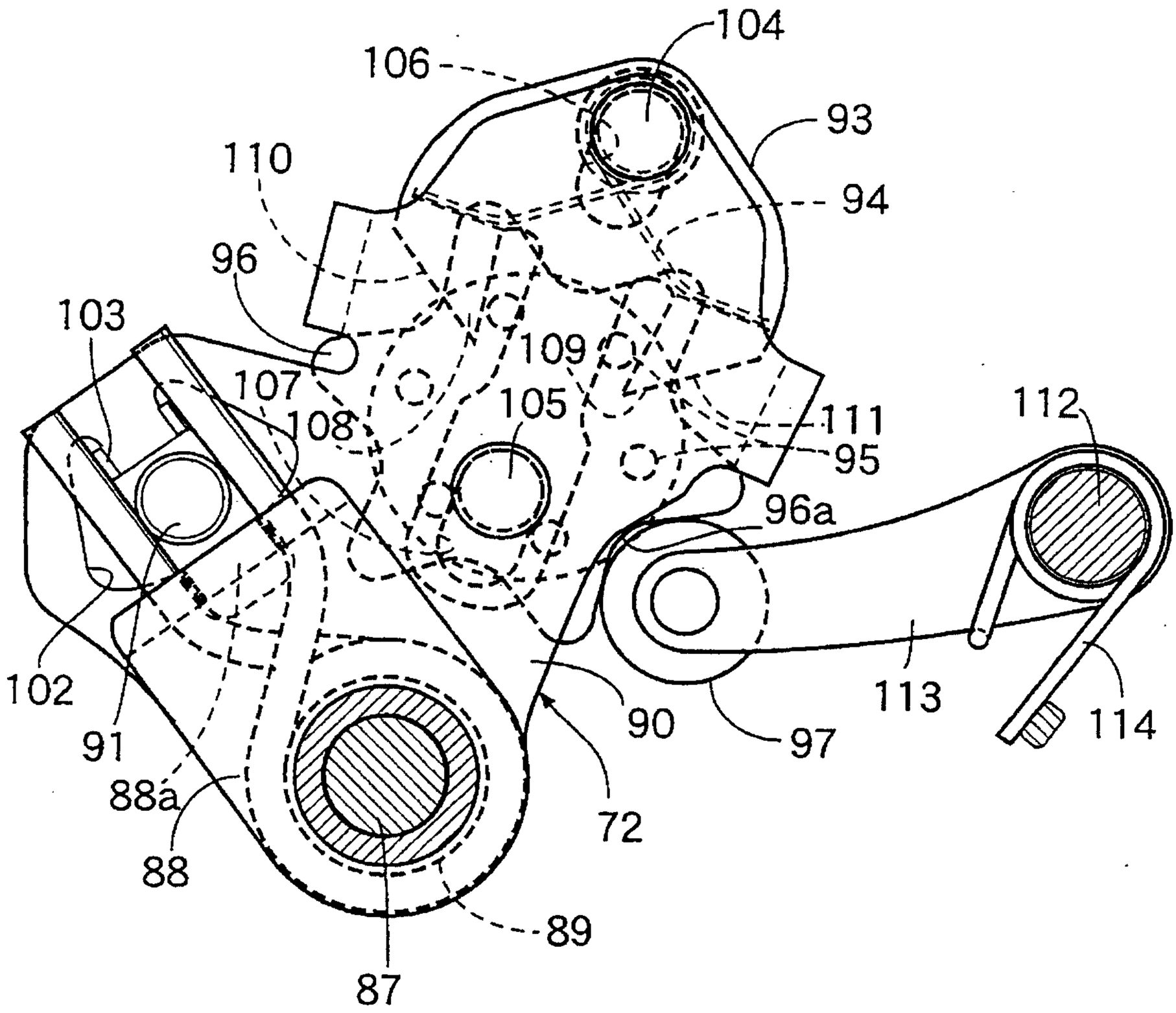

104
106
93
110
94
96
103
109
107
105
111
112
108
95
96a
102
90
113
91
97
114
88a
72
88
89
87

図 8

Fig. 8

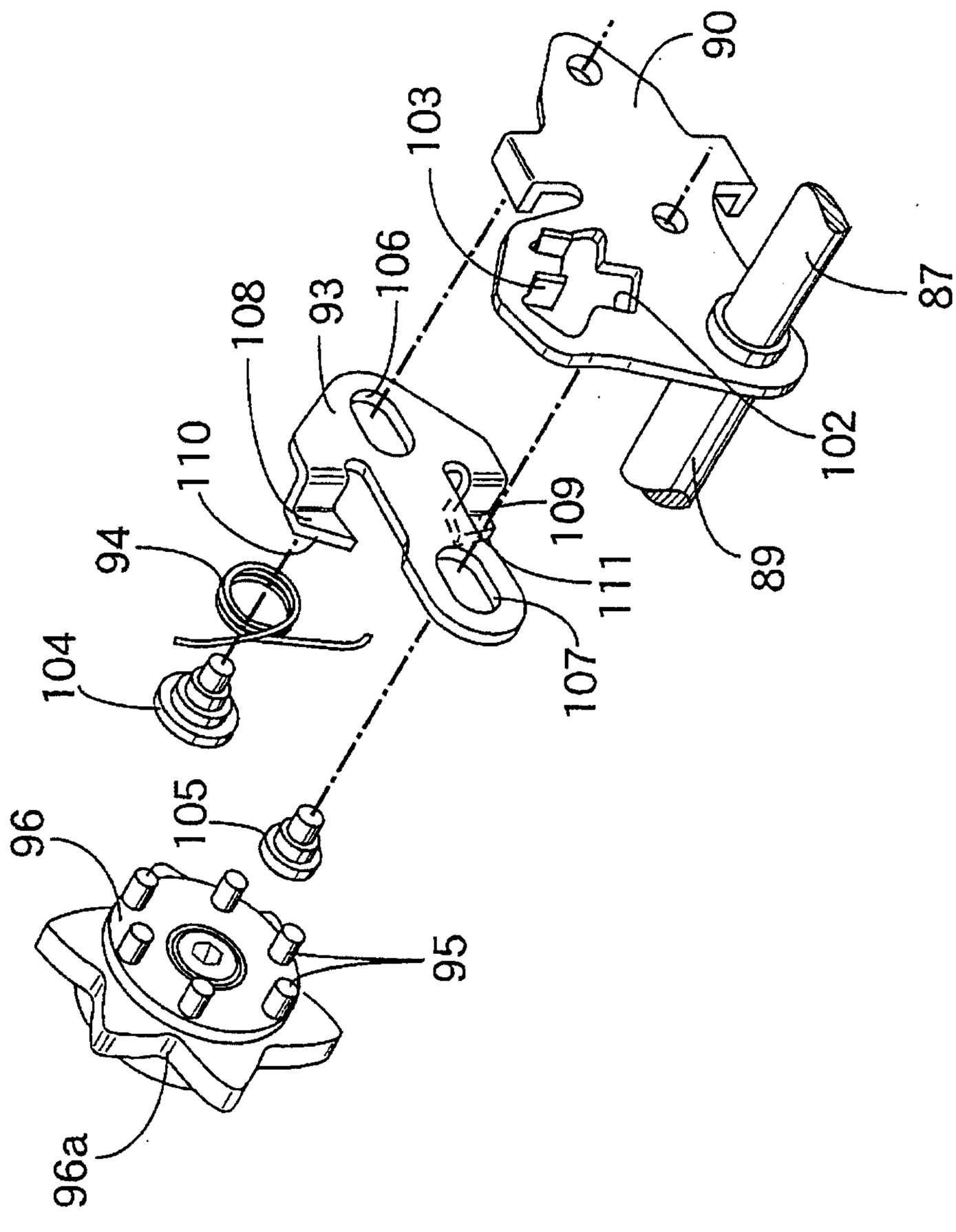


図 9

Fig. 9

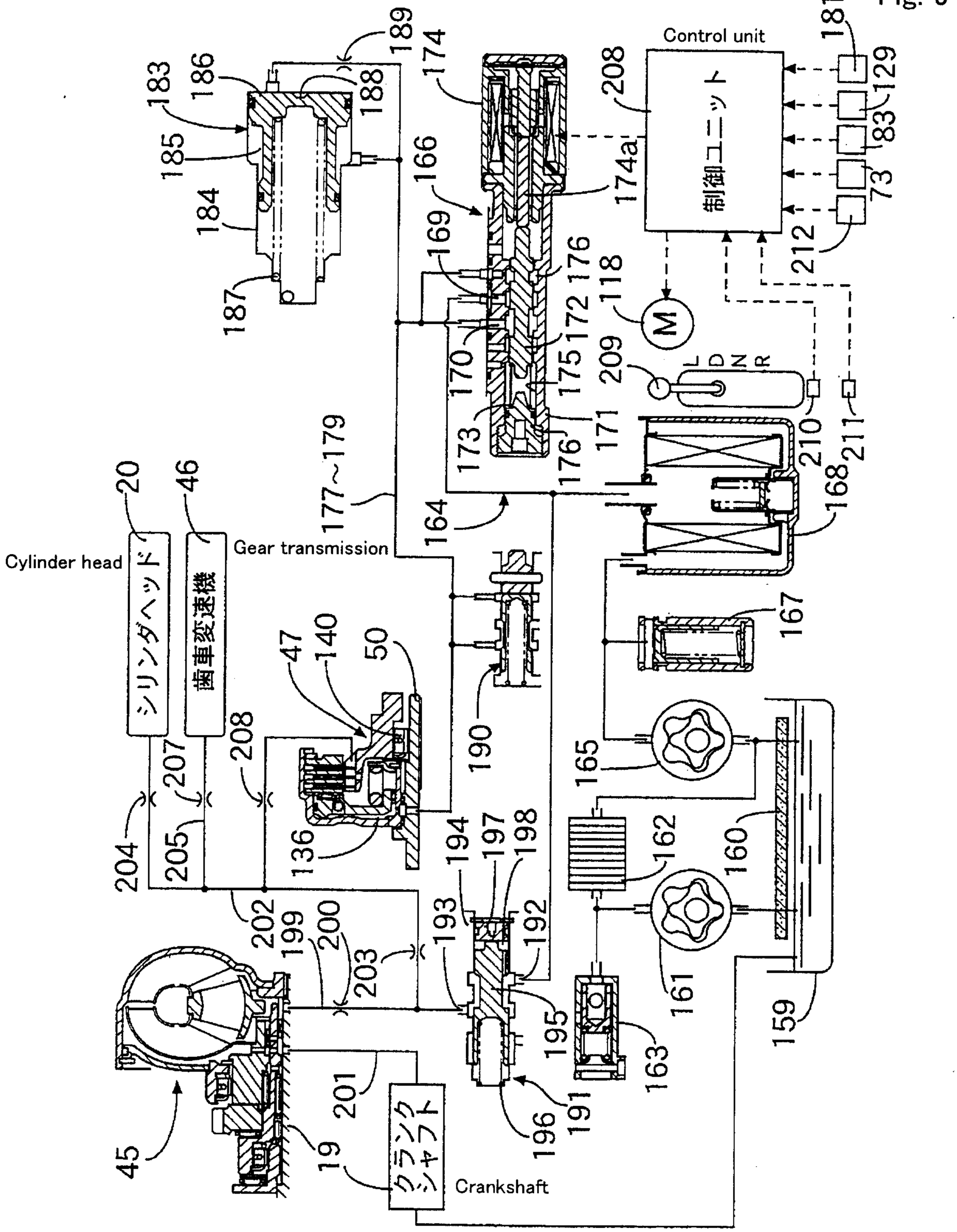

Control unit
制御ユニット
Cylinder head
シリンダヘッド
Gear transmission
歯車変速機
Crankshaft
クランクシャフト
L
D
N
R
M

図 10

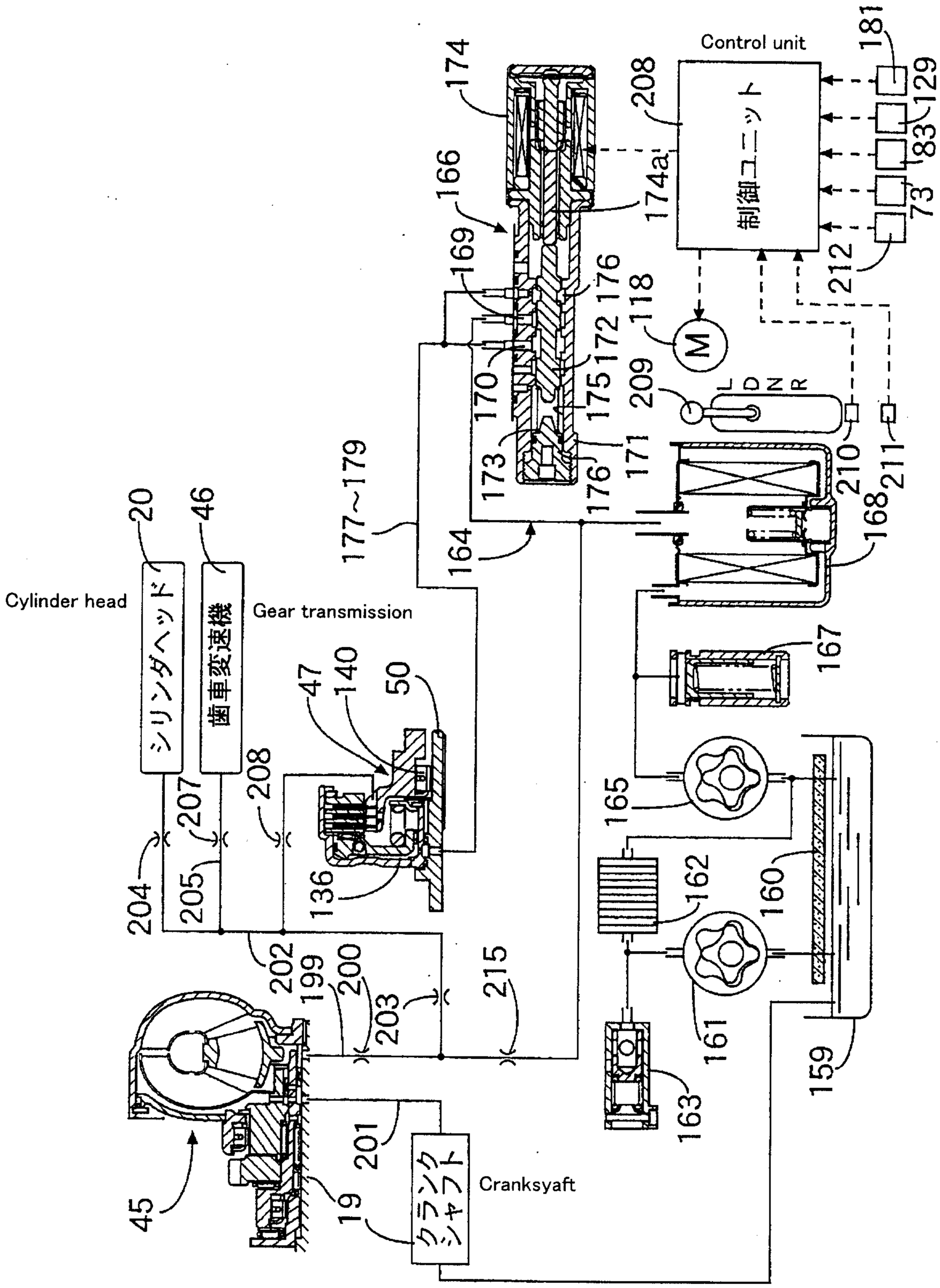
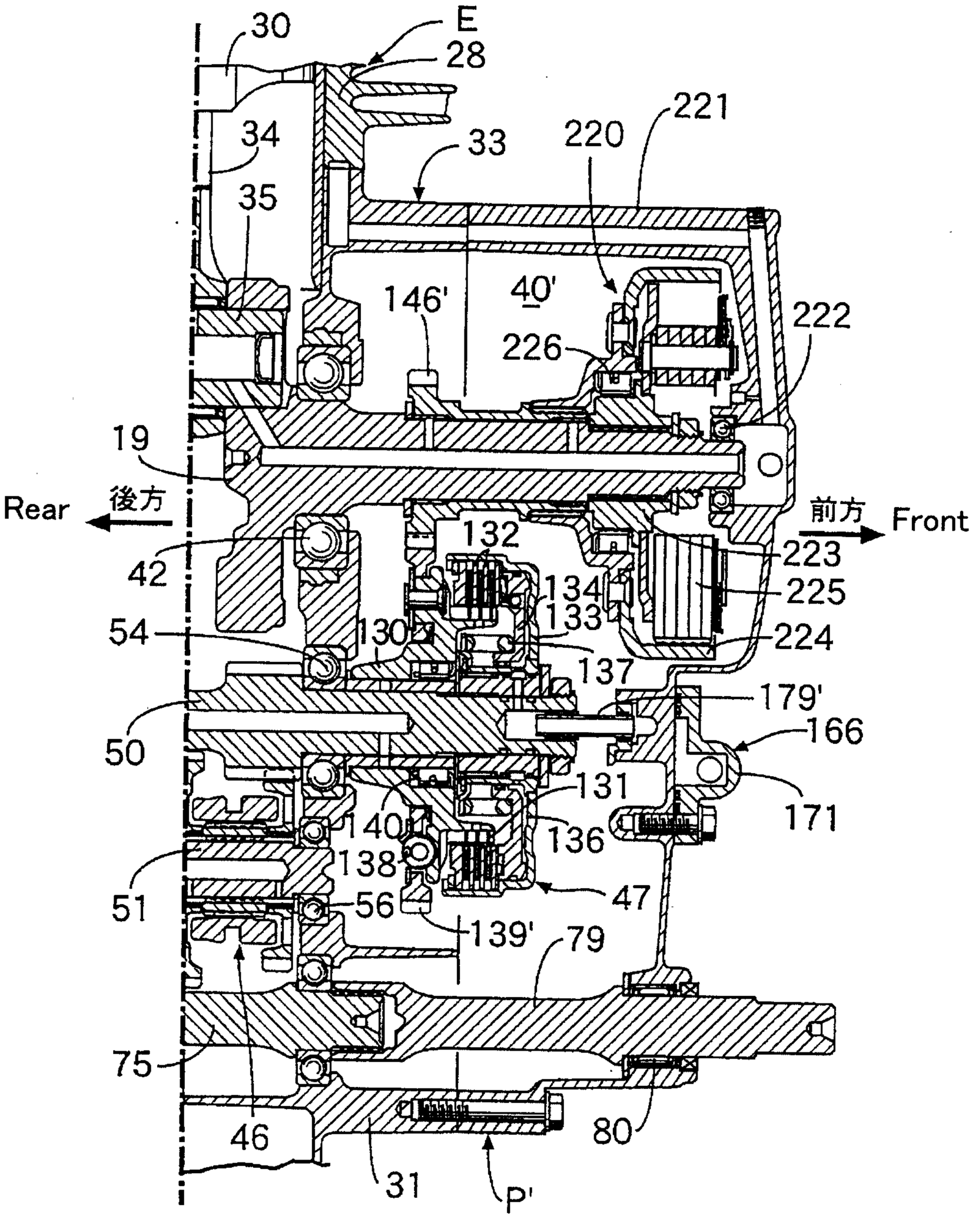
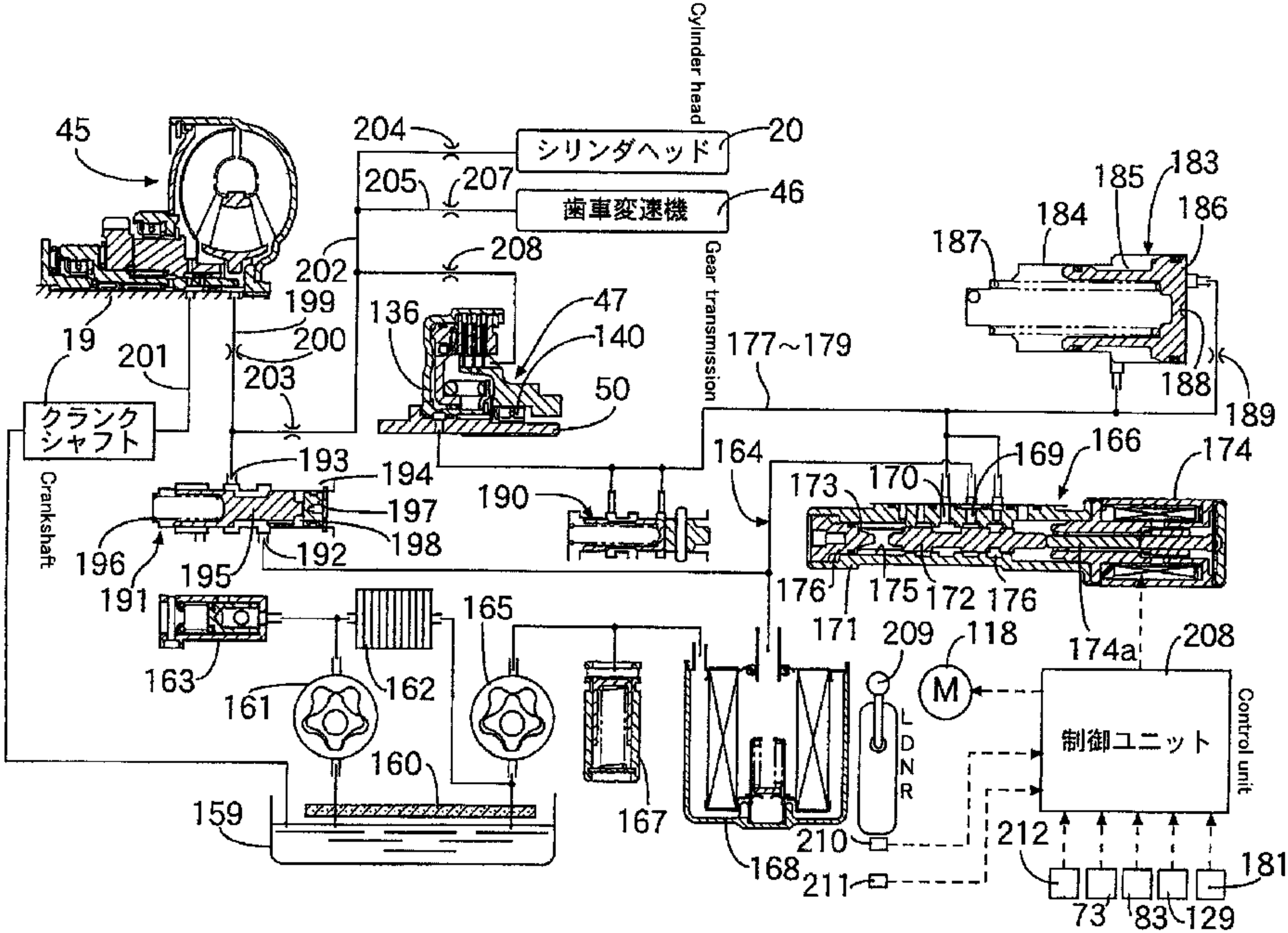

Control unit
制御ユニット
Cylinder head
シリンダヘッド
Gear transmission
歯車変速機
Cranksyaft
クランクシャフト
L
D
N
R
M

図 11

Fig. 11


30
E
28
34
33
35
221
220
146'
40'
222
226
19
Rear
後方
前方
Front
132
42
223
134
225
133
54
130
224
137
179'
50
166
131
171
140
136
138
47
51
56
139'
79
75
46
80
31
P'


Cylinder head
シリンダヘッド
20
歯車変速機
46
Gear transmission
45
204
205
207
208
202
199
200
203
201
19
クランクシャフト
Crankshaft
136
47
140
50
177~179
183
185
184
186
187
188
189
193
194
197
198
192
196
195
191
190
164
166
169
174
173
170
176
175
172
171
174a
163
162
161
165
160
159
167
168
209
118
M
208
制御ユニット
Control unit
L
D
N
R
210
211
212
181
73
83
129