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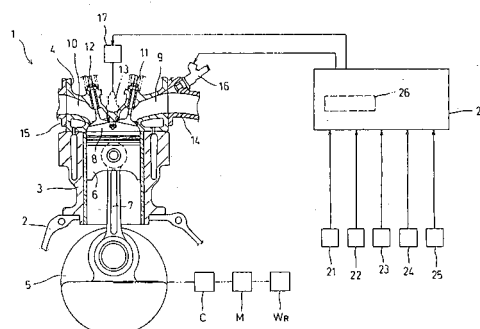
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(54) RECIPROCATING INTERNAL COMBUSTION ENGINE AND ITS OPERATING METHOD

(57) A method of operating a reciprocating internal combustion engine reduces friction power loss in an indicated power to increase net power and to reduce brake specific fuel consumption. A reciprocating internal combustion engine having an output characteristic to generate a maximum indicated power (PM) at a first set engine speed (N1) is provided with a power reducing unit (26). The power reducing unit (26) reduces the output power of the reciprocating internal combustion engine when the engine speed exceeds a second set engine speed lower than the first set engine speed (N2) lower than the first set engine speed (N1) so that the reciprocating internal combustion engine generates a necessary indicated power (PS) at the second set engine speed (N2). A high engine speed range in an engine speed range (R) in which the engine speed of the reciprocating internal combustion engine varies is lower than an engine speed (N3) at which an equivalent conventional reciprocating internal combustion engine generates a necessary indicated power (PS) (Maximum indicated power (Pm). Friction power loss (PL) in an indicated power of the reciprocating internal combustion engine is smaller than that in the same indicated power of the equivalent conventional reciprocating internal combustion engine.

bustion engine. So that net power is increased and brake specific fuel consumption is reduced.

Fig.1



Description

TECHNICAL FIELD

[0001] The present invention relates to a reciprocating internal combustion engine having an output characteristic to generate a maximum indicated power at a first set engine speed and designed to generate a necessary indicated power at an engine speed below the first set engine speed, and to a method of operating the engine.

BACKGROUND ART

[0002] The maximum indicated power on a performance curve indicating the performance of a conventional reciprocating internal combustion engine having a predetermined piston displacement in a full-load operation or a full-throttle operation is dependent on data on the components of the reciprocating internal combustion engine including dimensions of the passages of the intake and the exhaust system, the diameters and valve lifts of the intake and exhaust valves, and compression ratio. Internal combustion engines having a greater piston displacement generate a greater maximum indicated power. The engine speed of internal combustion engines having a greater piston displacement is lower than that of internal combustion engines having a smaller piston displacement for the same indicated power. An engine speed corresponding to the maximum indicated power of a reciprocating internal combustion engine is in a high engine speed range in an overall engine speed range in which the engine speed of the reciprocating internal combustion engine varies. This maximum indicated power is a necessary indicated power, i.e., a maximum indicated power required of the reciprocating internal combustion engine.

[0003] A reciprocating internal combustion engine for a vehicle is provided with a starting clutch which starts transmitting the torque of the crankshaft of the reciprocating internal combustion engine to a power transmission system including a transmission upon the increase of the engine speed to a predetermined engine speed. When the starting clutch is a centrifugal clutch provided with expanding weights supported for turning on support shafts and serving as friction shoes, the centrifugal clutch is of a leading-shoe type in which the center of gravity of each friction shoe lies ahead of the support shaft with respect to the rotating direction of the crankshaft. The torque transmitting ability, i.e., the clutch capacity, of the leading-shoe type centrifugal clutch is enhanced by the rotation of an outer clutch member because frictional force exerted by the outer clutch member on the friction shoes causes the friction shoes to turn diametrically outward. Such an action will be referred to as a self-locking action.

[0004] The rotor of an alternator connected to the crankshaft of the reciprocating internal combustion en-

gine is formed integrally with a flywheel for the maintenance of uniform rotation of the crankshaft.

[0005] Part of the indicated power of the reciprocating internal combustion engine is lost due to friction power loss including friction loss because of mechanical friction between the sliding components of the reciprocating internal combustion engine including the crankshaft, the crank pins, the pistons and the valve mechanism, and power dissipation by driving accessories including the oil pump and the power generator. Thus, the net power of the reciprocating internal combustion engine is equal to the remainder of subtraction of the friction power loss from the indicated power. Friction power loss increases, as well as indicated power, with the increase of engine speed. Therefore, it is difficult to reduce greatly brake specific fuel consumption, i.e., fuel consumption per unit net power per unit time, while the reciprocating internal combustion engine is operating at an engine speed in the high engine speed range including the engine speed corresponding to the maximum indicated power. Consequently, a reciprocating internal combustion engine which is frequently operated in the high engine speed range operates inevitably at a high fuel consumption rate.

[0006] In the leading-shoe type centrifugal clutch, the reduction of the rotating speed of the crankshaft resulting from the sharp increase of load on the crankshaft due to the exertion of pressure on the outer clutch member by the self-locking action of the friction shoes when the centrifugal clutch is engaged, and the increase of the rotating speed of the crankshaft resulting from the reduction of load on the crankshaft due to the reduction of the pressure exerted on the outer clutch member by the friction shoes when the rotating speed decreases are repeated alternately. Consequently, unpleasant vibrations called judders are liable to be generated and the vibrations are transmitted through the body of the vehicle to the passenger. In the operation of a reciprocating internal combustion engine which operates in a low engine speed range, a starting clutch connected to the reciprocating internal combustion engine is engaged at a low engine speed. Therefore, the starting clutch must be provided with large friction shoes or heavy friction shoes to secure a proper clutch capacity. However, such large or heavy friction shoes enhance vibratory force exerted on the body of the vehicle by judders, and the unpleasant high-energy vibrations are transmitted to the passenger.

[0007] The alternator having the rotor formed integrally with the flywheel must be changed when the flywheel is changed to adjust the degree of suppression of the variation of the rotating speed of the crankshaft by a flywheel and hence such adjustment of the degree of suppression of the variation of the rotating speed of the crankshaft cannot be easily achieved. Sometimes, a reciprocating internal combustion engine which operates mostly in a low engine speed range needs an additional flywheel to suppress the variation of the rotating speed

of the crankshaft. When an additional flywheel is necessary, a space for the additional flywheel must be secured and hence the size of the reciprocating internal combustion engine increases necessarily.

[0008] The present invention has been made in view of such problems and it is therefore an object of the present invention to provide a reciprocating internal combustion engine operating method capable of reducing friction loss at an indicated power to increase net power to reduce brake specific fuel consumption by setting a high engine speed range included in an engine speed range in which a reciprocating internal combustion engine operates below an engine speed corresponding to a maximum indicated power, and making the reciprocating internal combustion engine generate a necessary indicated power in the high engine speed range.

[0009] It is another object of the present invention to provide a reciprocating internal combustion engine capable of carrying out the reciprocating internal combustion engine operating method.

[0010] A further object of the present invention is to provide a flywheel structure for a reciprocating internal combustion engine, capable of reducing judders which are generated when a centrifugal clutch serving as a starting clutch is engaged and facilitating the adjustment of revolution inertia mass.

DISCLOSURE OF THE INVENTION

[0011] According to a first aspect of the present invention, a method of operating a reciprocating internal combustion engine having an output characteristic to generate a maximum indicated power at a first set engine speed comprises the steps of: providing the reciprocating internal combustion engine with an engine speed sensor for measuring engine speed, and a power reducing means for reducing the output power of the reciprocating internal combustion engine; and reducing the output power of the reciprocating internal combustion engine by the power reducing means when an engine speed measured by the engine speed sensor is higher than a second set engine speed lower than the first set engine speed so that the reciprocating internal combustion engine provides a necessary indicated power at the second set engine speed.

[0012] According to the present invention, the power reducing means that operates when an engine speed measured by the engine speed sensor exceeds the second set engine speed below the first set engine speed at which a maximum indicated power is achieved reduces the output power of the reciprocating internal combustion engine so that the reciprocating internal combustion engine provides a necessary indicated power, i. e., a maximum indicated power. In the conventional reciprocating internal combustion engine, the maximum indicated power is the necessary indicated power. The reciprocating internal combustion engine according to

the present invention has the output characteristic to generate a maximum indicated power greater than the necessary indicated power and has a piston displacement greater than that of the conventional internal combustion engine. Therefore, the second set engine speed at which the necessary indicated power is generated is lower than an engine speed at which the conventional internal combustion engine generates a maximum indicated power, and hence friction power loss included in the necessary indicated output in the reciprocating internal combustion engine of the present invention is lower than that in the conventional internal combustion engine. Accordingly, when the reciprocating internal combustion engine of the present invention operates in the high engine speed range, the friction power loss included in the indicated output is smaller than that in the conventional internal combustion engine and hence the net power of the reciprocating internal combustion engine of the present invention is high. Since the reciprocating internal combustion engine of the present invention operates at engine speeds in an engine speed range including the high engine speed range lower than the engine speed range including a high engine speed range in which the conventional internal combustion engine operates, the reciprocating internal combustion engine may comprise component parts having comparatively low rigidity and comparatively low strength as compared with the component parts of the conventional internal combustion engine which must have high rigidity and high strength to withstand severe operation in an engine speed range higher than that for the reciprocating internal combustion engine.

[0013] Since the reciprocating internal combustion engine of the present invention generates the necessary indicated power at a low engine speed as compared with that at which the conventional internal combustion engine generates a necessary indicated power. Therefore, the friction power loss included in the indicated power is reduced when the reciprocating internal combustion engine is operating in the high engine speed range in the engine speed range, the net power increases, the brake specific fuel consumption is reduced, and the fuel consumption rate of the reciprocating internal combustion engine can be reduced when the reciprocating internal combustion engine is operated frequently in the high engine speed range. Since the reciprocating internal combustion engine of the present invention operates in the engine speed range including a high engine speed range lower than that of the conventional internal combustion engine, the component parts of the reciprocating internal combustion engine of the present invention may have comparatively low rigidity and comparatively low strength, so that the reciprocating internal combustion engine of the present invention has lightweight construction, which contributes to the reduction of fuel consumption rate.

[0014] In the reciprocating internal combustion engine operating method of the present invention, an en-

engine speed range having an upper limit engine speed equal to the first set engine speed may be divided into a low engine speed range, a middle engine speed range and a high engine speed range, and the second set engine speed may be included in either the middle or the low engine speed range.

[0015] Thus, the engine speed corresponding to the maximum indicated power of the reciprocating internal combustion engine of the present invention is far lower than that corresponding to the maximum indicated power of the conventional internal combustion engine. Therefore, a friction power loss included in an indicated power generated by the reciprocating internal combustion engine operating at engine speeds in the high engine speed range including the second set engine speed becomes far less than that generated by the conventional internal combustion engine, and hence the net power increases accordingly. Since the reciprocating internal combustion engine operates in the engine speed range having the high engine speed range far lower than that in which the conventional internal combustion engine operates, the reciprocating internal combustion engine may comprise component parts having comparatively low rigidity and comparatively low strength.

[0016] The reciprocating internal combustion engine may be connected to a manual transmission having a plurality of speeds, and different second set engine speeds may be determined respectively for the plurality of speeds of the manual transmission. When the different second set engine speeds are thus determined respectively for the plurality of speeds, proper driving force can be generated for each speed, and driving force can be optionally changed during upshift by changing the second set engine speed for each speed. Consequently, the driving force can be smoothly changed during upshift and smooth acceleration can be achieved. Driving force can be secured and the variation of driving force during upshift can be easily adjusted for vehicles having transmissions respectively having different speeds.

[0017] The reduction of the power of the reciprocating internal combustion engine can be achieved by stopping the supply of fuel to the reciprocating internal combustion engine. This power reducing means reduces fuel consumption more effectively than reducing engine output by changing or controlling an ignition characteristic of the engine. For example, the change or control of the ignition characteristic can be made by delaying or advancing ignition timing relative to optimum ignition timing, stopping ignition, or omitting some ignition cycles.

[0018] According to a second aspect of the present invention, a reciprocating internal combustion engine comprises an engine speed sensor for measuring engine speed, a power reducing means for reducing the output power of the reciprocating internal combustion engine, and a control means; wherein the control means makes the power reducing means reduce the output power of the reciprocating internal combustion engine when an engine speed measured by the engine speed

sensor exceeds a second set engine speed below the first set engine speed.

[0019] The reciprocating internal combustion engine may be an automotive internal combustion engine comprising a crankshaft, and a starting clutch having an outer clutch member connected to the crankshaft, and the starting clutch may be a trailing-shoe type centrifugal clutch provided with swing centrifugal weights, i.e., friction shoes, supported on support shafts, respectively, for turning and capable of coming into contact with the outer clutch member when the engine speed of the reciprocating internal combustion engine exceeds a predetermined engine speed.

[0020] Each of the friction shoes of the trailing-shoe type centrifugal clutch has a center of gravity lying behind the support shaft with respect to the rotating direction of the crankshaft. Therefore, when the friction shoes come into contact with the outer clutch member to engage the centrifugal clutch, friction force tends to cause the friction shoes to turn diametrically inward, and hence the change of the pressure exerted by the friction shoes on the outer clutch member is small as compared with the change of the pressure exerted by the friction shoes on the outer clutch member in the leading-shoe type centrifugal clutch. Thus, it is possible to narrow the range of variation of engine speed attributable to the alternate repetition of the reduction of the rotating speed of the crankshaft resulting from the increase of the pressure on the outer clutch member, i.e., the increase of load on the crankshaft, and the increase of the rotating speed of the crankshaft resulting from the reduction of the pressure exerted on the outer clutch member by the friction shoes, i.e., the reduction of the load on the crankshaft.

[0021] Consequently, judders can be reduced, vibratory force exerted on the body of the vehicle by judders can be reduced and the transmission of the unpleasant vibrations to the passenger can be reduced. The starting clutch can be provided with large friction shoes or heavy friction shoes to secure a proper clutch capacity when the starting clutch is engaged while the reciprocating internal combustion engine is operating in a low engine speed range, and judders can be reduced.

[0022] The reciprocating internal combustion engine may include a driven member connected to the crankshaft and driven by a starter motor, and an overrunning clutch having an outer race interposed between the crankshaft and the driven member, and the outer race may be detachably connected to a connecting part of the rotor of an alternator that rotates together with the crankshaft, may be formed in a diameter greater than that of the connecting part and may serve as a flywheel having an adjustable revolution inertia mass.

[0023] Thus, the reciprocating internal combustion engine can be provided with an additional flywheel without being enlarged, so that the variation of the rotating speed of the crankshaft can be further effectively suppressed. Since the outer race serving as a flywheel can

be detachably connected to the rotor of the alternator, the revolution inertia mass of the flywheel can be readily adjusted simply by changing the outer race and hence the degree of suppression of the variation of the rotating speed of the crankshaft can be easily adjusted. The outer race may have a part of a maximum width outside a predetermined radius, and a recess for receiving one axial end part of the rotor of the alternator in a part inside the predetermined radius. Thus, the outer race having a large revolution inertial mass can be formed. Since the one axial end part of the rotor of the alternator is fitted in the recess of the outer race, the assembly of the rotor and the outer race can be formed in a small axial dimension.

[0024] Consequently, the outer race having a large revolution inertia mass can be incorporated into the reciprocating internal combustion engine without increasing the size of the reciprocating internal combustion engine to provide the reciprocating internal combustion engine with a flywheel effective in suppressing the variation of engine speed. Since the axial end part of the rotor of the alternator can be fitted in the recess formed in the part of the outer race inside the predetermined radius, the assembly of the outer race and the rotor has a comparatively small axial dimension, which is effective in preventing the enlargement of the reciprocating internal combustion engine.

[0025] In this specification, the term "axial direction" signifies a direction parallel to the axis of the crankshaft, and the term "radial direction" signifies a direction perpendicular to the axis of the crankshaft.

BRIEF DESCRIPTION OF THE DRAWINGS

[0026]

Fig. 1 is a sectional view of an essential part of a reciprocating internal combustion engine in a preferred embodiment of the present invention;
 Fig. 2 is sectional view in a plane including the axis of a crankshaft included in the reciprocating internal combustion engine shown in Fig. 1;
 Fig. 3 is an enlarged view of a part of Fig. 2;
 Fig. 4 is a view taken in the direction of the arrow IV in Fig. 3, in which a side cover covering an alternator and an overrunning clutch is removed;
 Fig. 5 is a view taken in the direction of the arrow V in Fig. 2, in which a drive plate is removed; and
 Fig. 6 is a graph comparatively showing engine performance curves showing the characteristic of the reciprocating internal combustion engine shown in Fig. 1 and those showing the characteristic of a conventional internal combustion engine.

BEST MODE FOR CARRYING OUT THE INVENTION

[0027] A reciprocating internal combustion engine in a preferred embodiment according to the present inven-

tion will be described with reference to Figs. 1 to 6.

[0028] As shown in Fig. 1, a reciprocating internal combustion engine 1 embodying the present invention is a single-cylinder, four-stroke-cycle, spark-ignition, water-cooled reciprocating internal combustion engine intended to be mounted on a motorcycle. Referring to Figs. 1 and 2, the reciprocating internal combustion engine 1 is formed by assembling a crankcase 2, a cylinder 3, a cylinder head 4 and a head cover, not shown. A crankshaft 5 is supported for rotation in main bearings 18 and 19 on the crankcase 2. A piston 6 is fitted in a cylinder bore formed in the cylinder 3 and is connected to the crankshaft 5 by a connecting rod 7. The linear motion of the piston 6 is converted into the rotary motion of the crankshaft 5.

[0029] Referring to Fig. 2, a drive sprocket 30, a starter driven gear 62 and an alternator 31 are mounted in that order from right to left on a left end part extending to the left from the main bearing 18 of the crankshaft 5. The drive sprocket 30 is fixed to the crankshaft 5. A camshaft 32 is supported for rotation on the cylinder head 4 and a cam sprocket 33 is fixedly mounted on the camshaft 32. A timing chain 34 is extended between drive sprocket 30 and the cam sprocket 33 to drive the camshaft 32 for rotation at a rotating speed equal to half the rotating speed of the crankshaft 5. A water pump 36 is operatively connected to a left end part of the camshaft 5 by a magnetic coupling 35 provided with a permanent magnet.

[0030] Referring to Figs. 3 and 4, a pinion 60a is fixedly mounted on the output shaft of a starter motor 60 and is interlocked through a reduction gear train 61 to a starter driven gear 62. A boss 62a of the starter driven gear 62 is supported on the crankshaft 5 by needles 63 for rotation on the crankshaft 5. The starter driven gear 62 is interlocked with the crankshaft 5 by a known cam type overrunning clutch 64. The boss 62a of the starter driven gear 62 serves as an inner race 65 of the overrunning clutch 64. The overrunning clutch 64 has an outer race 66. Three bolts B are screwed through three holes formed in the outer race 66 in threaded holes formed in a rotor 31b included in the alternator 31 and fixedly mounted on the crankshaft 5 to fasten the outer race 66 to the rotor 31b.

[0031] The overrunning clutch 64 includes the boss 62a serving as the annular inner race 65, the annular outer race 66, a plurality of cam rolls 67 having a cam surface and interposed between the inner race 65 and the outer race 66, a retainer 68 (Fig. 4) for retaining the plurality of cam rolls 67 in place at predetermined intervals, side plates 69 and 70 restraining the cam rolls 67 from axial movement, and springs 71 connected to the cam rolls 67.

[0032] When the starter motor 60 is actuated to start the reciprocating internal combustion engine 1, the rotation of the output shaft of the starter motor 60 is transmitted through the pinion 60a and the reduction gear train 61 to the starter driven gear 62, the rotation of the

starter driven gear 62 is transmitted through the overrunning clutch 64 and the rotor 31b to the crankshaft 5 to drive the crankshaft 5 for rotation. Upon the increase of the rotating speed of the crankshaft 5 beyond the rotating speed of the starter driven gear 62, the transmission of rotation of the crankshaft 5 to the starter driven gear 62 is cut off by the overrunning clutch 64.

[0033] The alternator 31 includes a stator 31a fixed to a generator cover 37, and the cup-shaped rotor 31b fixedly mounted on the crankshaft 5 so as to surround the stator 31a. The rotor 31b has a boss 31c keyed to the crankshaft 5 and fastened to the same with a nut 38, a flange 31cl extending around the boss 31c, and a magnet holding rim 31d riveted to the flange 31c. The outer race 66 is detachably attached to the flange 31cl of the rotor 31b with the bolts B.

[0034] The outer race 66 has an outside diameter greater than that of the outer race of an equivalent, ordinary overrunning clutch and greater than that of the flange 31cl. Thus, the outer circumference 66a of the outer race 66 is close to the inner circumference 37a of the generator cover 37. A step 66b of a predetermined height and of a diameter substantially equal to the outside diameter of the flange 31cl is formed in an end surface on the side of the rotor 31b of the outer race 66. The outer race 66 has an annular outer part 66c of a maximum thickness extending between the step 66b and the outer circumference 66a. The step 66b and an annular inner part 66d having a thickness smaller than that of the annular outer part 66c by a value corresponding to the height of the step 66b define a recess 72. The flange 31cl is fitted in the recess 72.

[0035] The annular outer part 66c of the outer race 66 extends diametrically outside the flange 31cl and can be formed in a great thickness so as to lie between the overrunning clutch 64 and the rotor 31b. Therefore, the annular outer part 66c can be formed in a large mass. Since the outer race 66 is detachably attached to the rotor 31b of the alternator 31 fixed to the crankshaft 5, the mass of the assembly of the outer race 66 and the rotor 31b can be changed by replacing the outer race 66 with another one having a mass different from that of the outer race 66. The assembly of the outer race 66 and the rotor 31b serves as a flywheel for suppressing the undesirable variation of the rotating speed of the crankshaft 5.

[0036] A tubular member 40 is mounted for rotation about the axis L of the crankshaft 5 on a right end part of the crankshaft 5, as viewed in Fig. 2, extending on the right side of the main bearing 19. A drive gear 41 is formed integrally with an end part on the side of the main bearing 19 of the tubular member 40. A starting clutch 50 is mounted on a right end part of the tubular member 40.

[0037] Referring to Figs. 2 and 5, the starting clutch 50 is a trailing-shoe type centrifugal clutch including a drive plate 51 fixedly mounted on the crankshaft 5, a cup-shaped outer member 52 fixedly mounted on the

tubular member 40 so as to surround the drive plate 51, three support shafts 53 fixed to the drive plate 51, three friction shoes 54 serving as centrifugal weights, supported for turning on the support shafts 53, and having outer surfaces coated with clutch liners 55, respectively, and clutch springs 56 biasing the friction shoes 54 radially inward. The center of gravity of each friction shoe 54 lies behind the support shaft 53 with respect to the rotating direction A of the crankshaft 5. When the starting clutch 50 is engaged, the outer member 52 exerts a friction force to turn the friction shoes 54 radially inward on the support shafts 53.

[0038] The starting clutch 50 is engaged when the engine speed N exceeds a predetermined level. Upon the increase of the engine speed N beyond the predetermined level, the friction shoes 54 are turned radially outward on the support shafts 53 against the resilience of the clutch springs 56, the clutch liners 55 are pressed against the rim of the outer member 52, to engage the drive plate 51 with the outer member 52, and the outer member 52 starts rotating together with the drive plate 51. The starting clutch 50 is engaged completely before long.

[0039] The friction shoes 54 of the starting clutch 50 is greater and heavier than those of an ordinary, equivalent centrifugal clutch employed in the conventional internal combustion engine in order that the starting clutch 50 has a clutch capacity capable of surely transmitting torque even while the engine speed N of the reciprocating internal combustion engine 1 is low.

[0040] A driven gear 43 engaged with the drive gear 41 is mounted on a main shaft 44 of a manual transmission M, i.e., a constant-mesh gear type transmission. The driven gear 43 is operatively connected by a damping member to an outer member of a transmission clutch C mounted on a right end part of the main shaft 44 projecting to the right, as viewed in Fig. 2, from the crankcase 2. The transmission clutch C is a multiple disk friction clutch including a plurality of friction disks. The transmission clutch C is engaged and disengaged by manually operating a release mechanism. When transmission clutch C is engaged by frictionally engaging the plurality of friction disks by the resilience of clutch springs, the torque of the crankshaft 5 is transmitted through the outer member of the transmission clutch C to an inner member of the same fixedly mounted on the main shaft 44. When the transmission clutch C is disengaged by removing the pressure applied to the friction disks by the clutch springs from the friction disks, the outer member of the transmission clutch C is disconnected from the inner member of the same to stop transmitting the torque from the outer member to the inner member of the starting clutch C.

[0041] The manual transmission M is disposed behind the crankshaft 5 in the crankcase 2. The manual transmission M includes the main shaft 44, main gears 45 mounted on the main shaft 44, a countershaft 46, and counter gears mounted on the countershaft 46. A

shift drum 48 is turned by operating a transmission operating mechanism, not shown, to move a shift fork supported on a support shaft and engaged in a cam groove formed in the cir of the shift drum 48 in a lateral direction, as viewed in Fig. 2. Thus, the selected main gear 456 and the corresponding counter gear 47 are engaged to set the manual transmission M in a desired speed.

[0042] A cover 80 is attached to the right end surface of the drive plate 51 to form a centrifugal strainer 81. Foreign matters contained in a lubricating oil supplied through an oil passage 82 connected to a main gallery, not shown, to the centrifugal strainer 81 is separated from the lubricating oil by centrifugal force, and the thus cleaned lubricating oil is supplied through an oil passage 83 formed in the crankshaft 5 to parts to be lubricated including a crankpin 5a.

[0043] Thus, the torque of the crankshaft 5 is transmitted through the starting clutch 50 to the drive gear 41 formed integrally with the tubular member 40, from the drive gear 41 through a primary reduction gear mechanism including the driven gear 43, and the transmission clutch C to the manual transmission M. Torque of the counter shaft 46 rotating at a rotating speed determined by the manual transmission M is transmitted through a secondary reduction gear mechanism, not shown, to the rear wheels W_R to drive the rear wheels W_R for rotation as shown in Fig. 1.

[0044] Referring again to Figs. 1 and 2, the cylinder head 4 defining a combustion chamber 8 together with the piston 6 is provided with an inlet port 9 and an exhaust port 10 respectively opening into the combustion chamber 8. An inlet valve 11 and an exhaust valve 12 are inserted in the inlet port 9 and the exhaust port 10 to open and close the same, respectively. The inlet valve 11 and the exhaust valve 12 are operated in synchronism with the rotation of the crankshaft 5 by a valve mechanism including the cam shaft 32 and rocker arms, not shown, to open and close the inlet port 9 and the exhaust port 10 in predetermined timing and to move the inlet valve 11 and the exhaust valve 12 by predetermined lifts, respectively. An ignition plug 13 is attached to the cylinder head 4 with its spark gaps exposed to the combustion chamber 8 to ignite an air-fuel mixture supplied into the combustion chamber 8.

[0045] An intake pipe 14 is connected to a side surface of the cylinder head 4 so as to communicate with the upper open end of the inlet port 9. A fuel injection valve 16 is attached to the intake pipe 14 to inject a fuel into the inlet port 9. The fuel injected into the inlet port 9 is mixed with air to produce an air-fuel mixture.. An exhaust pipe 15 is connected to a side surface of the cylinder head 4 so as to communicate with a lower open end of the exhaust port 10.

[0046] An electronic control unit (ECU) 20, i.e., a control means, for controlling fuel injection rate and ignition timing receives signals from an engine speed sensor 21 for measuring the engine speed N, a throttle opening sensor 22 for measuring the opening of a throttle valve,

a pressure sensor 23 for measuring intake pipe pressure, a temperature sensor 24 for measuring the temperature of cooling water and a gear position switch 25. The electronic control unit 20 controls the fuel injection valve 16 to inject the fuel at a fuel injection rate meeting the operating condition of the reciprocating internal combustion engine 1 and determined on the basis of signals provided with the sensors 21 to 24 and the switch 25. The electronic control unit 20 controls an ignition coil 17 to control high-tension voltage generating timing so that the ignition plug 13 ignites the air-fuel mixture at ignition timing determined according to the operating condition of the reciprocating internal combustion engine 1.

[0047] Fig. 6 shows performance curves indicating the performance of the reciprocating internal combustion engine 1 when the reciprocating internal combustion engine 1 is in full-load operation (full-throttle operation). Suppose that the reciprocating internal combustion engine 1 has a piston displacement of 200 cm³. As indicated by the performance curve represented by a two-dot chain line, the reciprocating internal combustion engine 1 generates a maximum indicated power PM when the engine speed N is equal to a predetermined engine speed N1 of, for example, 6000 rpm. The maximum indicated power PM is dependent on data on the component parts of the reciprocating internal combustion engine 1, such as the sizes of the gas passages of the intake and the exhaust system, the diameters and the lifts of the inlet valve 11 and the exhaust valve 12, and compression ratio. In Fig. 6, a curve PL represented by continuous line is a friction power loss curve for the reciprocating internal combustion engine 1.

[0048] The reciprocating internal combustion engine 1 has an output characteristic to generate the maximum indicated power PM when internal combustion engine 1 operates at the first set engine speed N1 of, for example, 6000 rpm. The reciprocating internal combustion engine 1 is designed such that a maximum necessary indicated power PS is generated when the reciprocating internal combustion engine 1 is operating at a second set engine speed N2 of, for example, 3500 rpm lower than the first set engine speed N1. When an engine speed N measured by the engine speed sensor 21 (Fig. 1) exceeds the second set engine speed N2, the engine speed sensor 21 gives a signal to the electronic control unit 20, the electronic control unit 20 executes a series of operations on the basis of the signal received from the engine speed sensor 21, a power reducing unit 26 gives a signal to the fuel injection valve 16 to stop injecting the fuel, the fuel injection valve 16 stops injecting the fuel and, consequently, the output power of the reciprocating internal combustion engine 1 decreases.

[0049] The second set engine speed N2 makes the reciprocating internal combustion engine 1 generates the maximum necessary indicated power PS, and is determined taking into consideration the piston displacement and the maximum indicated power PM. Suppose

that the reciprocating internal combustion engine 1 operates at engine speeds in an engine speed range defined by a lower limit of 0 rpm (the engine speed $N = 0$ and the reciprocating internal combustion engine 1 is stopped) and an upper limit equal to the first set engine speed $N1$, and the engine speed range is divided into a low engine speed range, a middle engine speed range and a high engine speed range. In this embodiment, the second set engine speed $N2$ is included in the middle engine speed range. The middle engine speed range is included in a high engine speed range in an engine speed range R (Fig. 6) in which the engine speed of the reciprocating internal combustion engine 1 varies.

[0050] The operation and effects of the reciprocating internal combustion engine 1 will be described hereinafter. Fig. 6 shows an output characteristic curve represented by a broken line and indicating the output characteristic of a conventional internal combustion engine having a piston displacement equal to half the piston displacement of the reciprocating internal combustion engine 1 and capable of generating a maximum indicated power P_m equal to the necessary indicated output PS , and a friction power loss curve represented by a broken line and indicating maximum indicated power P_m including friction power loss PL_m . It is known from Fig. 6 that the engine speed at which the reciprocating internal combustion engine 1 generates the necessary indicated output PS , i.e., the second set engine speed $N2$ is not higher than half the engine speed $N3$ at which the conventional internal combustion engine generates the maximum indicated power P_m , and the friction power loss PL_m included in the maximum indicated power PS of the reciprocating internal combustion engine 1 is not greater than half the friction power loss PL_m included in the maximum indicated power P_m of the conventional internal combustion engine. It is known from the comparative examination of data on the reciprocating internal combustion engine 1 and that on the conventional internal combustion engine shown in Fig. 6 that the friction power loss PL of the reciprocating internal combustion engine 1 is smaller than that of the conventional internal combustion engine for the same indicated power. When measuring the values of the friction power loss PL shown in Fig. 6, the sliding parts that cause friction loss and accessories that cause auxiliary driving loss were the same for the reciprocating internal combustion engine 1 and the conventional internal combustion engine.

[0051] In the reciprocating internal combustion engine 1 having the output characteristic to generate the maximum indicated power PM at the first set engine speed $N1$, the power reducing unit 26, which operates when the engine speed N exceeds the second set engine speed $N2$ included in the middle engine speed range (one of the engine speed ranges determined by equally dividing an engine speed range having the upper limit equal to the first set engine speed $N1$, i.e., the high, the middle and the low engine speed range) and

below the first set engine speed $N1$, reduces the output power of the reciprocating internal combustion engine 1 such that the necessary indicated power PS , i.e., a maximum indicated power required of the reciprocating internal combustion engine 1, is generated at the second set engine speed $N2$ far less than the first set engine speed $N1$ for generating the maximum indicated power PM . Since the maximum indicated power P_m is equal to the necessary indicated power PS in the conventional internal combustion engine and the reciprocating internal combustion engine 1 has the output characteristic to generate the maximum indicated power PM higher than the necessary indicated power PS , the reciprocating internal combustion engine 1 has the piston displacement greater than that of the conventional internal combustion engine. The second set engine speed $N2$ at which the necessary indicated power PS is generated is far less than the engine speed $N3$ at which the conventional internal combustion engine generates the maximum indicated power P_m , and the friction power loss PL_m included in the necessary indicated power PS is far less than that of the friction power loss of the conventional internal combustion engine. Therefore, the friction power loss PL included in the indicated power is far less than that of the conventional internal combustion engine even while the reciprocating internal combustion engine 1 is operating at engine speeds in the high engine speed range of the engine speed range R and the net power increases accordingly. Consequently, the brake specific fuel consumption is reduced greatly and hence the fuel consumption rate of the reciprocating internal combustion engine 1 is low even when the reciprocating internal combustion engine 1 operates frequently at engine speeds in the high engine speed range. Since the reciprocating internal combustion engine 1 operates at engine speeds in the engine speed range R including the high engine speed range which is far lower than that for the conventional internal combustion engine, the reciprocating internal combustion engine 1 may comprise component parts having comparatively low rigidity and strength, whereas the conventional internal combustion engine must comprise component parts having high rigidity and strength capable of withstanding severe operation in the engine speed range higher than the engine speed range R . Therefore, the reciprocating internal combustion engine 1 can be formed in lightweight construction and fuel consumption rate can be reduced.

[0052] The power reducing unit 26 cuts off the fuel supplied to the reciprocating internal combustion engine 1 upon the increase of the engine speed beyond the second set engine speed $N2$. Therefore, fuel consumption is small as compared with fuel consumption when the power of the reciprocating internal combustion engine is reduced by controlling the ignition timing, and hence the fuel consumption rate is reduced still further.

[0053] Although the reciprocating internal combustion engine 1 operates at engine speeds in the engine speed

range R including the high engine speed range far lower than that for engine speeds at which the conventional internal combustion engine operates, the variation of the rotating speed of the crankshaft 5 when the crankshaft 5 rotates at a low rotating speed can be suppressed and hence the reciprocating internal combustion engine 1 operates smoothly because the starting clutch 50 and the alternator 31 mounted on the crankshaft 5 add a revolution inertia mass to the crankshaft 5. Since the engine speed range for the reciprocating internal combustion engine 1 is lower than that for the conventional internal combustion engine, the reciprocating internal combustion engine 1 is provided with the starting clutch 50 larger than that of the conventional internal combustion engine or is provided with the centrifugal weights of a mass greater than that of the centrifugal weights of the conventional internal combustion engine to ensure stable torque transmission during operation at low engine speeds. Thus, the revolution inertia mass of the reciprocating internal combustion engine 1 is large and hence the variation of engine speed can be still more effectively suppressed.

[0054] Since the starting clutch 50 is a trailing-shoe type centrifugal clutch, the center of gravity of each friction shoe 54 lies behind the support shaft 53 with respect to the rotating direction A of the crankshaft 5. The starting clutch 50 is engaged when the engine speed N exceeds the predetermined level. Upon the increase of the engine speed N beyond the predetermined level, the clutch liners 55 of the friction shoes 54 are pressed against the rim of the outer member 52. When the starting clutch 50 is thus engaged, the frictional force exerted by the rim of the outer member 52 on the clutch liners 55 tends to turn to turn the friction shoes 54 radially inward. Therefore, the variation of the pressure applied to the rim of the outer member 52 is more moderate than that of the pressure applied to the rim of the outer member of a leading-shoe type centrifugal clutch, and it is thus possible to narrow the range of variation of engine speed attributable to the alternate repetition of the reduction of the rotating speed of the crankshaft 5 resulting from the increase of the pressure on the outer clutch member, i.e., the increase of load on the crankshaft 5, and the increase of the rotating speed of the crankshaft resulting from the reduction of the pressure exerted on the outer clutch member by the friction shoes, i.e., the reduction of the load on the crankshaft. Consequently, judders can be reduced, vibratory force exerted on the body of the vehicle by judders can be reduced and the transmission of the unpleasant vibrations to the passenger can be reduced.

[0055] Since the second set engine speed N2 is included in the middle engine speed range and is not higher than half the engine speed N3 at which the conventional internal combustion engine generates the maximum indicated power Pm, judders can be reduced even when the reciprocating internal combustion engine 1 operates at engine speeds in the engine speed range R

lower than that for the conventional internal combustion engine because starting clutch 50 is provided with the heavy or large friction shoes 54 and has a proper clutch capacity.

[0056] The outer race 66 of the overrunning clutch 64 interlocking the starter driven gear 62 with the crankshaft 5 is attached to the flange 31cl of the rotor 31b of the alternator 31 in a space corresponding to that extending radially around the outer race of the conventional overrunning clutch so as to extend radially outside the flange 31cl. and the outer race 66 serves as a flywheel, the reciprocating internal combustion engine 1 can be provided with an additional flywheel without being enlarged, so that the variation of the rotation of the crankshaft 5 can be further effectively suppressed. Since the outer race 66 serving as a flywheel is detachably attached to the rotor 31b of the alternator 31, the revolution inertia mass of the additional flywheel can be readily changed simply by replacing the outer race 66 with another one and hence the degree of suppression of the variation of rotation can be easily adjusted.

[0057] The annular outer part 66c of the outer race 66 extending diametrically outside the circular step 66b of the predetermined radius in the space extending radially outside the flange 31cl of the rotor 31b can be formed in the great thickness. Therefore, the outer race 66 can be formed in a large revolution inertia mass.

[0058] Accordingly, the additional flywheel having a large revolution inertia mass can be incorporated into the reciprocating internal combustion engine 1 without enlarging the reciprocating internal combustion engine 1 for the effective suppression of the variation of the rotating speed of the crankshaft 5 even when the reciprocating internal combustion engine 1 operates at engine speeds in the engine speed range R lower than that for the conventional internal combustion engine because the second set engine speed N2 is included in the middle engine speed range and is not greater than half the engine speed N3 at which the conventional internal combustion engine generates the maximum indicated power Pm.

[0059] Since the flange 31cl of the rotor 31b of the alternator 31 is fitted in the recess 72 of the outer race 66, the axial dimension of the assembly of the outer race 66 and the rotor 31 is small, which contributes to avoiding the enlargement of the reciprocating internal combustion engine 1.

[0060] Possible changes and variations in the reciprocating internal combustion engine 1 in the foregoing embodiment will be described. Although the second set engine speed N2 is included in the middle engine speed range in the foregoing embodiment, the second set engine speed N2 may be any one of engine speeds below the first set engine speed N1 and may be determined on the basis of the piston displacement and the maximum indicated power PM of the reciprocating internal combustion engine 1. When the second set engine speed N2 is thus determined, the respective degrees of

the reduction of fuel consumption rate and the reduction of the weight of the reciprocating internal combustion engine 1 are less than those in the foregoing embodiment, the effect on the reduction of brake specific fuel consumption and on the reduction of the weight of the reciprocating internal combustion engine 1 is the same as that of the foregoing embodiment.

[0061] The reciprocating internal combustion engine 1 may be provided with the gear position switch 25 as shown in Fig. 1 to detect the gear position of the manual transmission M and second engine speeds may be determined for the first to the fourth speed of the manual transmission M, respectively. In such a case, higher second set engine speeds N are specified for the lower speeds so that the second set engine speed N for the first speed is the highest and that for the fourth speed is the lowest to secure necessary driving force when the manual transmission M is set in any one of the speeds.

[0062] Since the second set engine speeds are determined for the speeds of the manual transmission M, respectively, proper driving forces are generated respectively for the speeds, and driving force can be optionally changed during upshift by changing the second set engine speed for each speed. Consequently, the driving force can be smoothly changed during upshift and smooth acceleration can be achieved. Driving force can be secured and the variation of driving force during upshift can be easily adjusted for vehicles having transmissions respectively having different speeds.

[0063] Although the power reducing unit 26 of the foregoing embodiment controls the fuel injection valve 16 to cut fuel supply, the power reducing unit may be such as which reduces fuel injection, delays or advances ignition timing greatly relative to optimum ignition timing or stops ignition or omits some ignition cycles to reduce the output of the reciprocating internal combustion engine 1. When the power reducing unit 26 reduces fuel injection, advances or delays ignition timing, or omits some ignition cycles, the reduction of the output when the engine speeds exceeds the second set engine speed N2 is gentle as compared with that when the fuel supply is cut as shown by a chain line in Fig. 6, and the maximum engine speed can be controlled by cutting fuel supply or stopping ignition.

[0064] The second set engine speed N2 may be included in the low engine speed range defined by an upper limit equal to the first set engine speed N1 when the reciprocating internal combustion engine 1 has a large piston displacement and the necessary indicated power is low. The present invention is applicable to a multiple-cylinder internal combustion engine, and the vehicle may be other than motorcycle.

[0065] The outer race 66 is provided with the recess 72 in the foregoing embodiment, the outer race 66 does not need necessarily to be provided with the recess 72, and the predetermined radius may be greater than the radius of the flange 31c extending around the boss 31c of the rotor 31b.

Claims

1. A method of operating a reciprocating internal combustion engine having an output characteristic to generate a maximum indicated power at a first set engine speed, said method comprising the steps of:

providing the reciprocating internal combustion engine with an engine speed sensor for measuring engine speed, and a power reducing means for reducing output power of the reciprocating internal combustion engine; and reducing the output power of the reciprocating internal combustion engine by the power reducing means when an engine speed measured by the engine speed sensor is higher than a second set engine speed lower than the first set engine speed so that the reciprocating internal combustion engine provides a necessary indicated power at the second set engine speed.

2. The reciprocating internal combustion engine operating method according to claim 1, wherein an engine speed range having an upper limit engine speed equal to the first set engine speed is divided into a low engine speed range, a middle engine speed range and a high engine speed range, and the second set engine speed is included in either the middle or the low engine speed range.
3. The reciprocating internal combustion engine operating method according to claim 1, wherein the reciprocating internal combustion engine is connected to a manual transmission having a plurality of speeds, and different second set engine speeds are determined respectively for the plurality of speeds of the manual transmission.
4. The reciprocating internal combustion engine operating method according to claim 1, wherein the power reducing means stops supply of fuel to the reciprocating internal combustion engine.
5. The reciprocating internal combustion engine operating method according to claim 1, wherein the power reducing means changes an ignition characteristic of the reciprocating internal combustion engine.
6. A reciprocating internal combustion engine having an output characteristic to generate a maximum indicated power at a first predetermined engine speed, said reciprocating internal combustion engine comprising:

an engine speed sensor for measuring engine speed;
a power reducing means for reducing output

power of the reciprocating internal combustion engine; and
a control means;

wherein the control means makes the power reducing means reduce the output power of the reciprocating internal combustion engine when an engine speed measured by the engine speed sensor exceeds a second set engine speed below the first set engine speed.

7. The reciprocating internal combustion engine according to claim 6, wherein the reciprocating internal combustion engine includes a crankshaft interlocked with an automotive trailing-shoe type centrifugal starting clutch, and
the trailing-shoe type centrifugal starting clutch includes:

an outer clutch member connected to the crankshaft; and
centrifugal friction shoes supported for turning on support shafts, respectively, and capable of coming into frictional contact with the outer clutch member when the engine speed of the reciprocating internal combustion engine exceeds a predetermined engine speed.

8. The reciprocating internal combustion engine according to claim 7, further comprising an overrunning clutch including:

a driven member connected to the crankshaft and driven by a starter motor;
an outer race interposed between the crankshaft and the driven member, detachably connected to a connecting part of a rotor included in an alternator that rotates together with the crankshaft, formed in a diameter greater than that of the connecting part, having a revolution inertia mass, and capable of serving as a fly-wheel and of being replaced with another one having another revolution inertia mass.

9. The reciprocating internal combustion engine according to claim 8, wherein the outer race has an annular outer part of a maximum thickness extending between a circular recess of a predetermined radius formed in one end surface thereof and the outer circumference thereof, and the connecting part of the rotor of the alternator is fitted in the circular recess.

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Fig.1

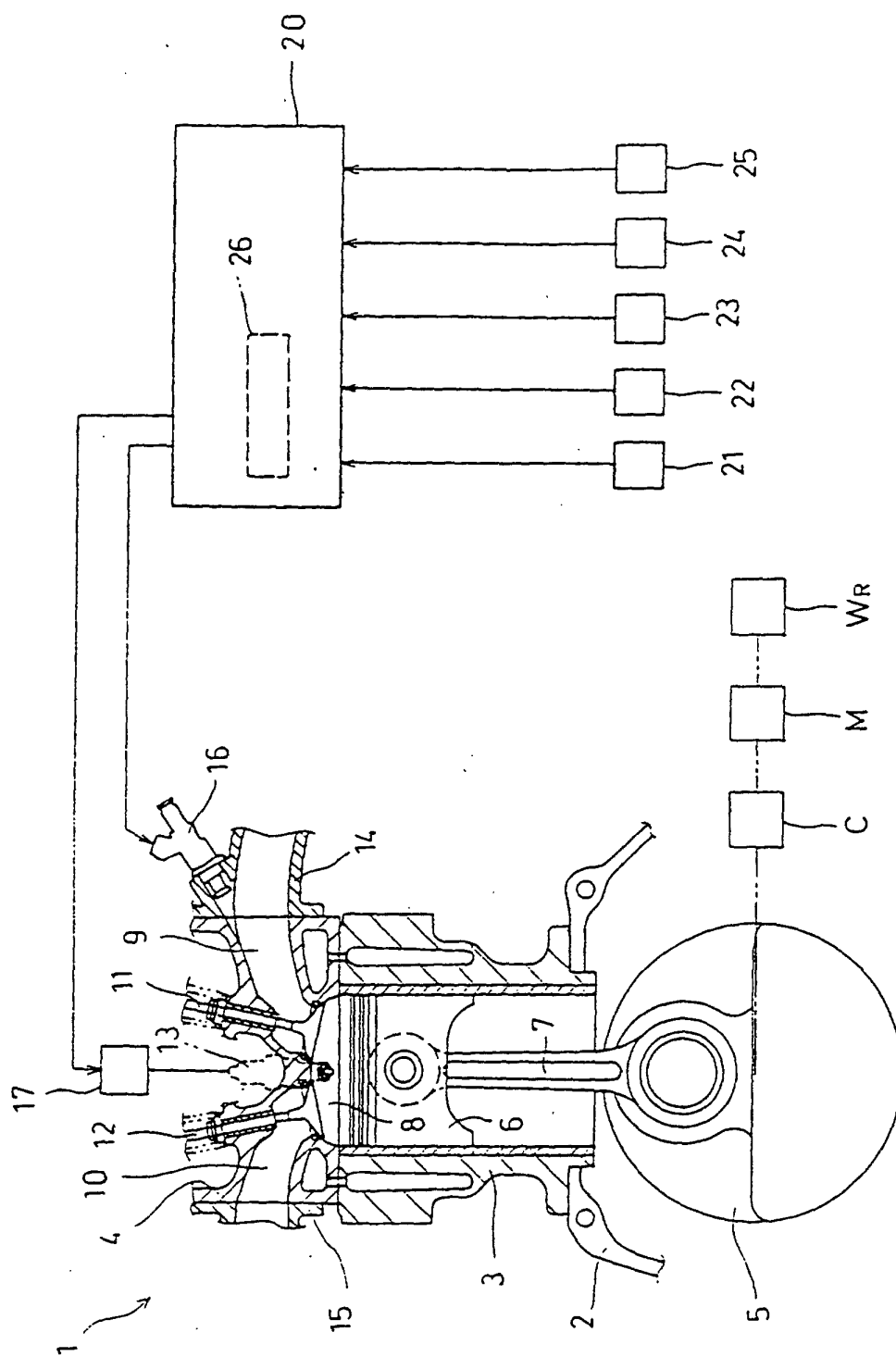


Fig.2

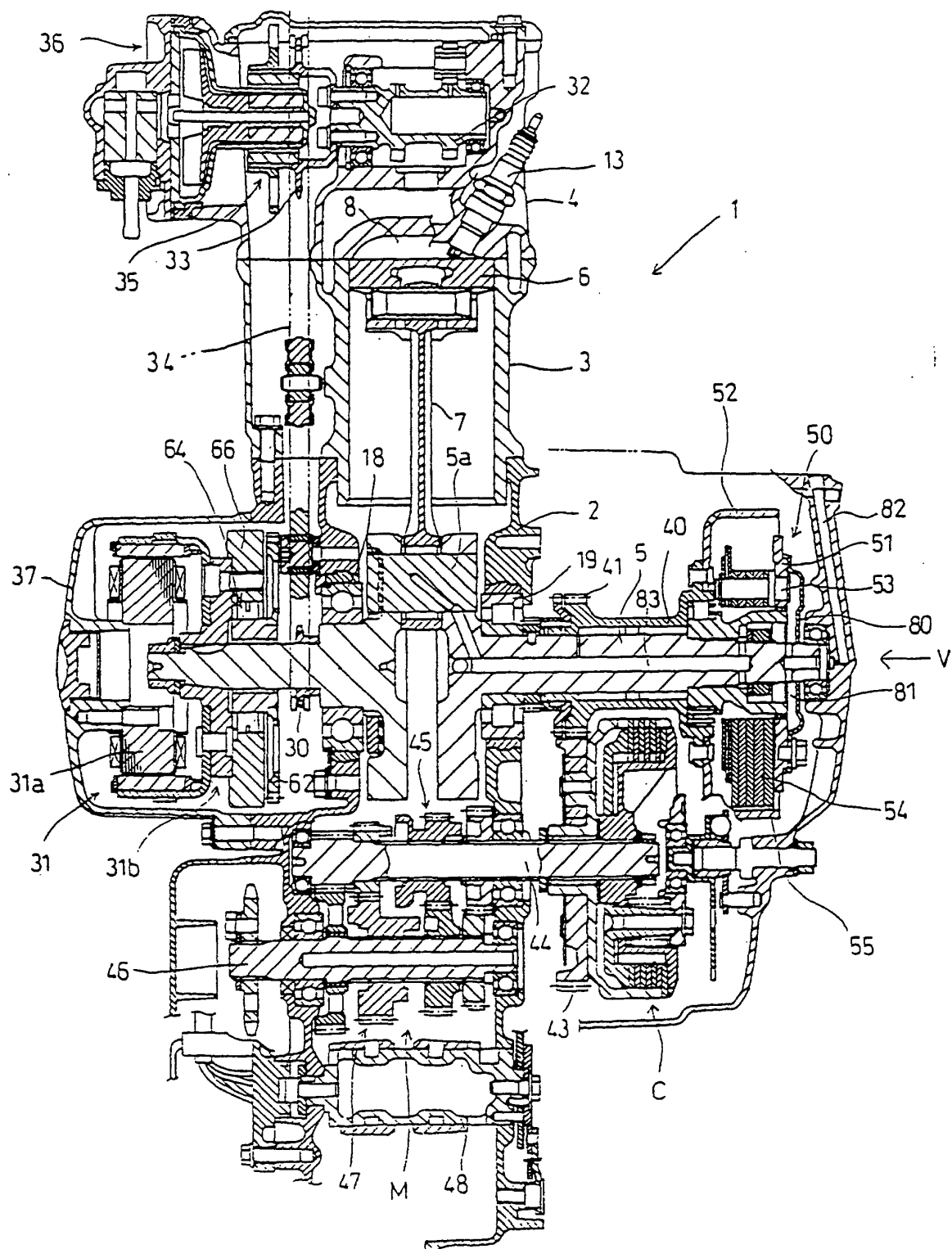


Fig.3

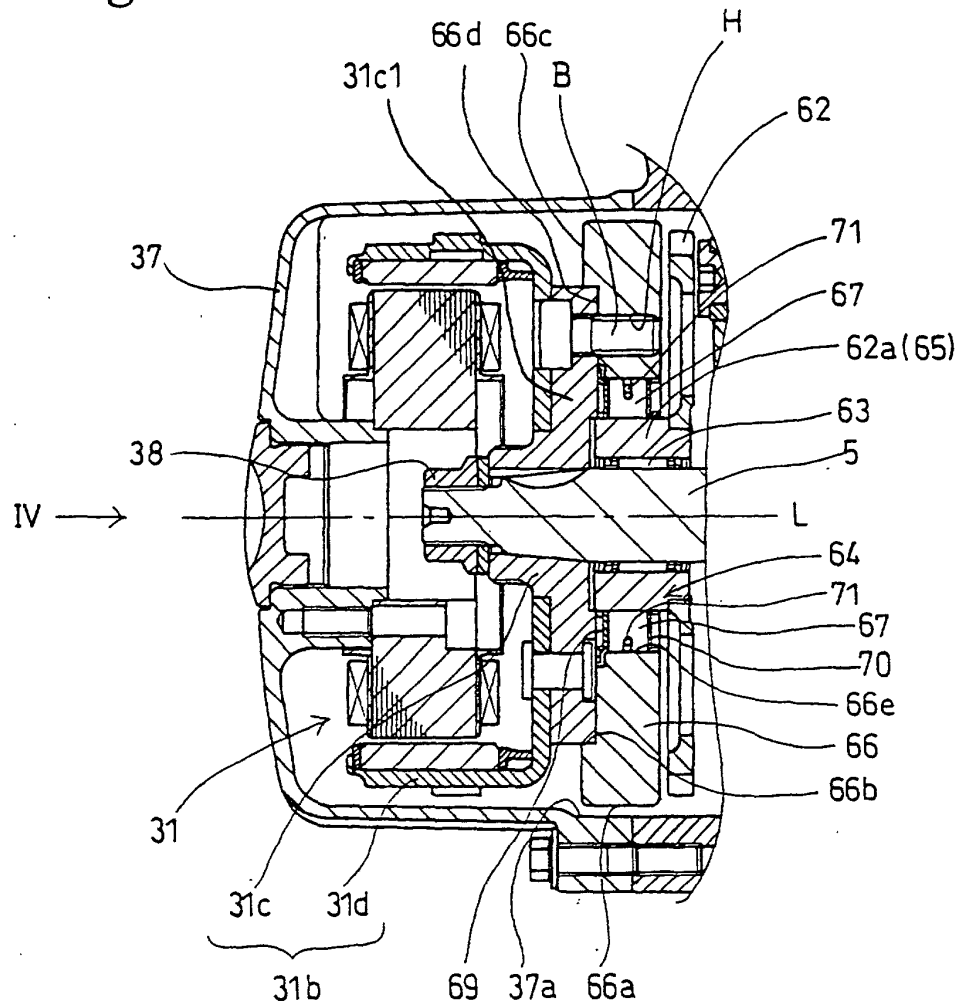


Fig.4

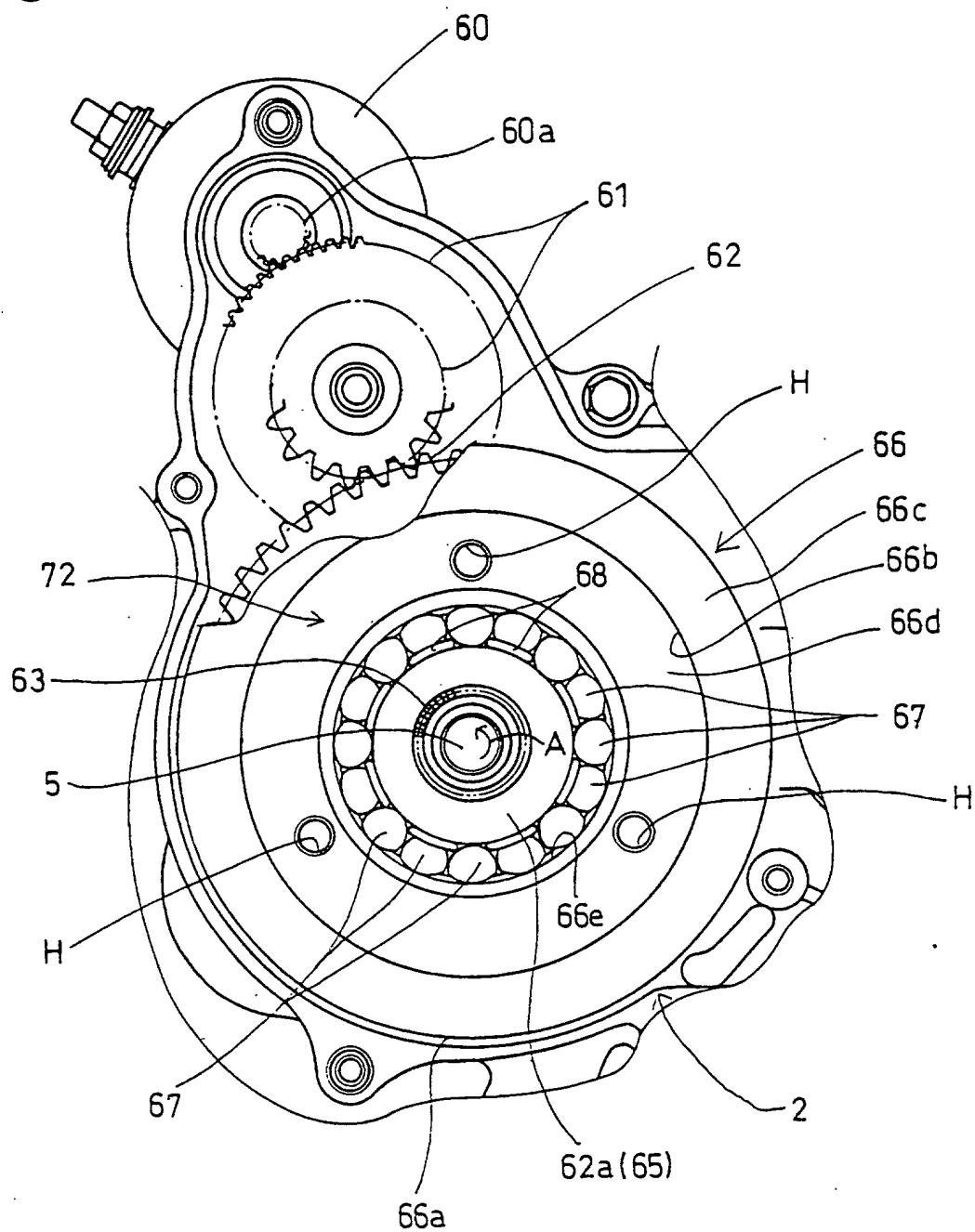


Fig.5

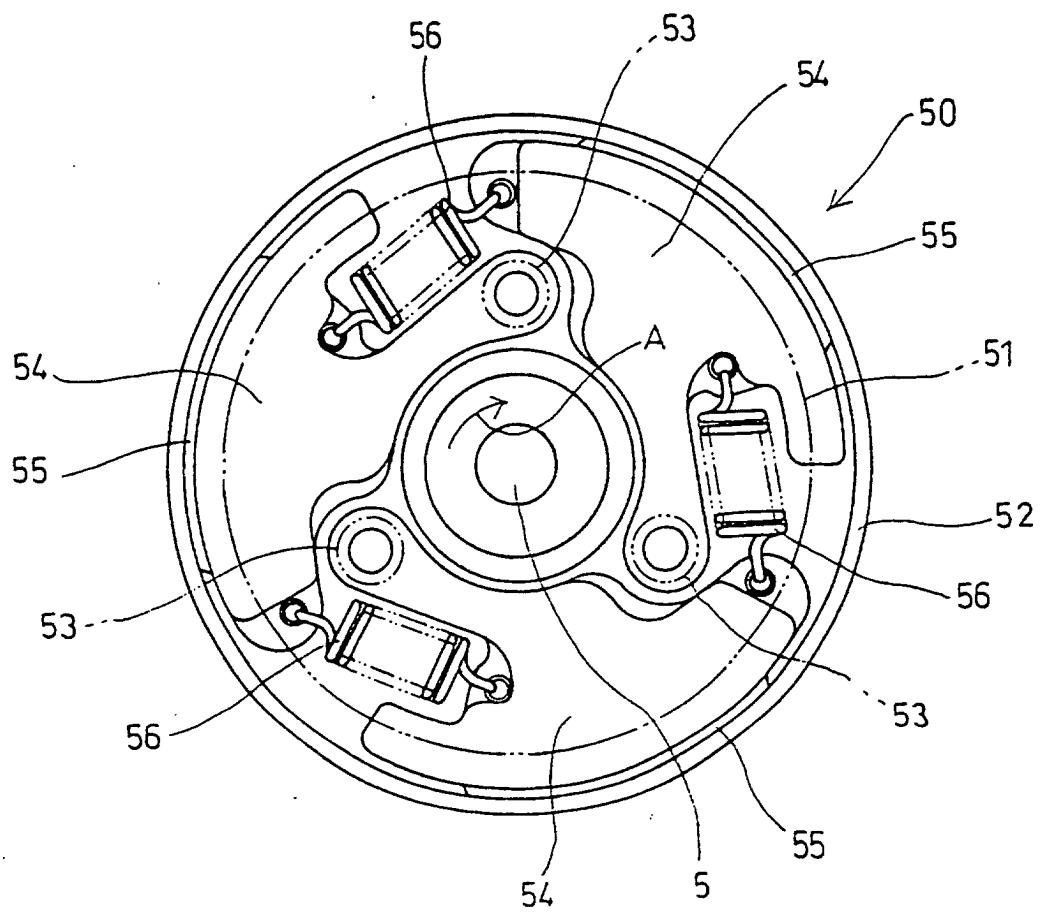
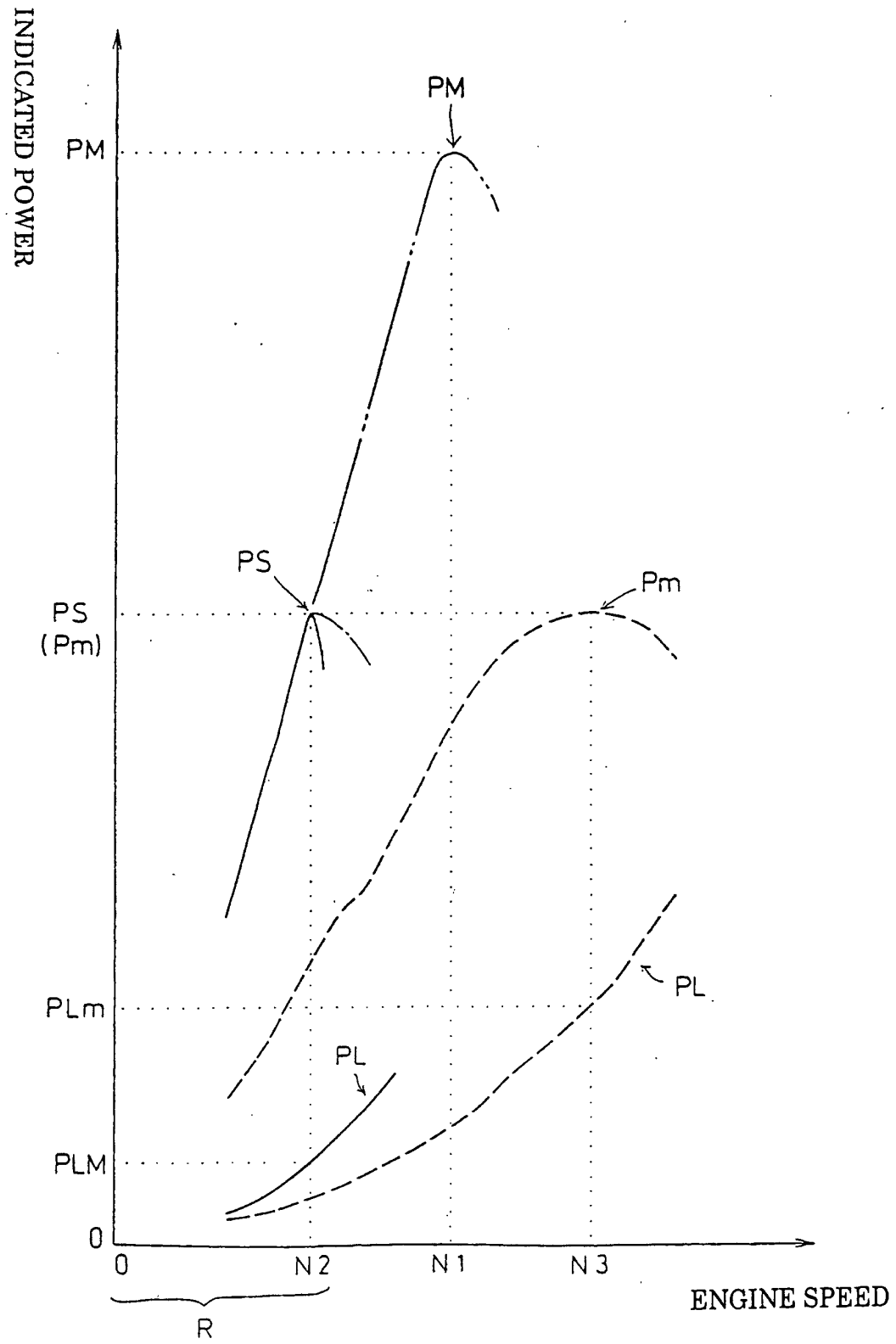


Fig.6



INTERNATIONAL SEARCH REPORT

International application No.

PCT/JP01/10113

A. CLASSIFICATION OF SUBJECT MATTER Int.Cl. ⁷ F02D45/00, F02P5/15, F02D17/02		
According to International Patent Classification (IPC) or to both national classification and IPC		
B. FIELDS SEARCHED Minimum documentation searched (classification system followed by classification symbols) Int.Cl. ⁷ F02D45/00, F02P5/15, F02D17/02		
Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched Jitsuyo Shinan Koho 1926-1996 Jitsuyo Shinan Toroku Koho 1996-2001 Kokai Jitsuyo Shinan Koho 1971-2001 Toroku Jitsuyo Shinan Koho 1994-2001		
Electronic data base consulted during the international search (name of data base and, where practicable, search terms used) ECLA (English; text retrieval in the following fields: [F16H63/18+F02D29/00+F02D41/04])		
C. DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	EP 1010880 A2 (Honda Motor Co., Ltd.), 21 June, 2000 (21.06.00), Claims & JP 2000-179439 A Claims	1-6
Y	JP 9-032602 A (Nissan Motor Co., Ltd.), 04 February, 1997 (04.02.97), Claims (Family: none)	1-6
A	JP 10-159973 A (Honda Motor Co., Ltd.), 16 June, 1998 (16.06.98), Claims	7-9
A	EP 1048845 A (Honda Motor Co., Ltd.), 02 November, 2000 (02.11.00), Claims & JP 2000-310134 A Claims	7-9
☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.		
* Special categories of cited documents: "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier document but published on or after the international filing date "I" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified) "O" document referring to an oral disclosure, use, exhibition or other means "P" document published prior to the international filing date but later than the priority date claimed		"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art "&" document member of the same patent family
Date of the actual completion of the international search 07 December, 2001 (07.12.01)		Date of mailing of the international search report 18 December, 2001 (18.12.01)
Name and mailing address of the ISA/ Japanese Patent Office		Authorized officer
Facsimile No.		Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)