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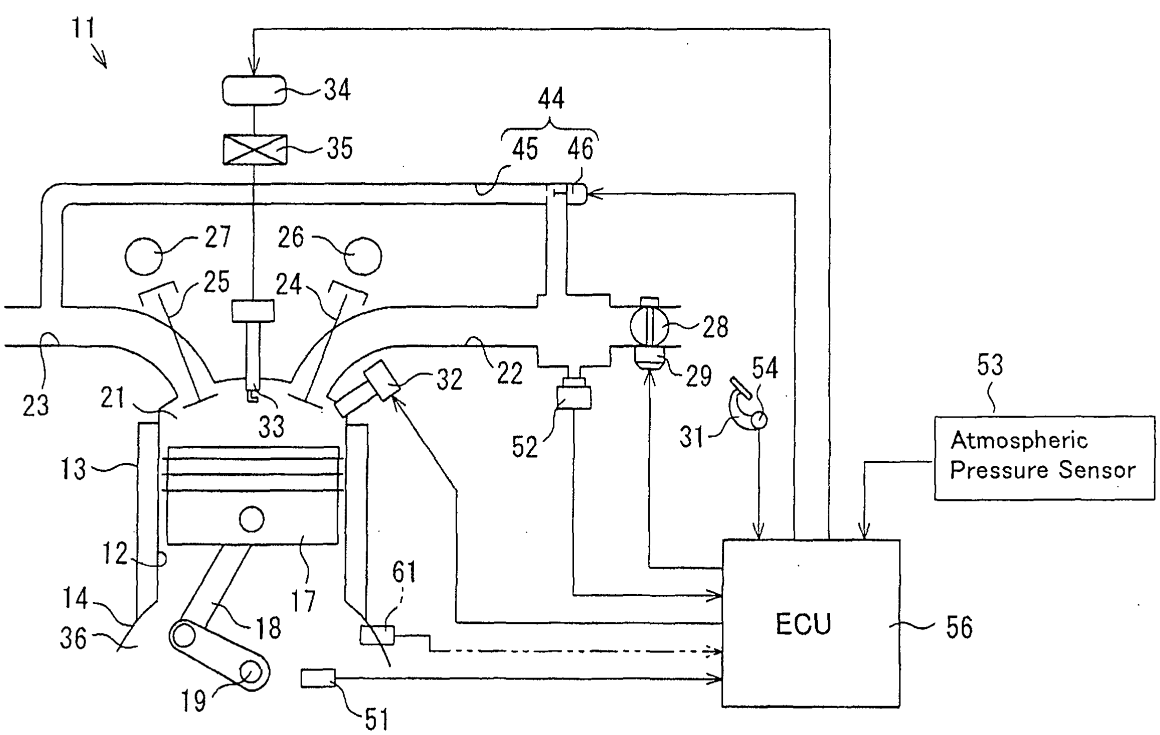
(54) **Controller for internal combustion engine**

(57) A controller for controlling an engine (11) to generate an output torque that is substantially equal to a target torque required by a driver. An ECU (56) calculates a pumping loss torque based on a pressure difference between a pressure in a crank chamber and an intake air pressure. The pumping loss torque is loss of an output torque of a crankshaft (19) caused by a pumping loss that occurs when a piston reciprocates. The ECU adds the pumping loss torque to the target torque required by the driver to calculate a final target torque. The ECU controls the engine based on the final target torque reflecting the pumping loss torque that is in accordance with the driving state of the engine.

Fig. 1



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Description

[0001] The present invention relates to a controller for controlling the fuel injection amount and the intake air amount of an internal combustion engine to obtain the required torque.

[0002] Torque demand control is known as one method for controlling the output torque of an internal combustion engine. The torque demand control calculates a target torque based on the engine speed and the amount an accelerator is depressed by a driver. Then, the fuel injection amount and an intake air amount are controlled so that the output torque of the internal combustion engine becomes equal to the target torque. To correct the target torque, a friction torque of the internal combustion engine is added to the target torque. Most of the friction torque is attributed to pressure loss (pumping loss), which is related to the resistance of intake air in a throttle valve, an EGR valve, etc. The friction torque is obtained from mechanical loss and auxiliary device loss. The mechanical loss includes friction loss of movable parts in the internal combustion engine. The auxiliary device loss occurs when driving auxiliary devices, such as an alternator and an air conditioner.

[0003] Japanese Laid-Open Patent Publication No. 11-62658 describes a conventional torque demand control. When the air-fuel ratio is lean, a large amount of air is drawn into combustion chambers under a high intake air pressure. This reduces the pumping loss. Focusing on this point, the conventional torque demand control calculates the pumping loss torque based on the intake air pressure or based on a parameter relating to the intake air pressure, such as the total amount of gas in the cylinders.

[0004] The pumping loss torque is calculated with higher accuracy when it is based on the intake air pressure or the total gas amount in the cylinders. However, the calculation accuracy of the pumping loss torque is still insufficient and requires improvements. For example, the engine actually has different pumping loss torques when driven on flatlands and highlands. However, the conventional control does not calculate the pumping loss torque considering such different driving states of the engine.

[0005] It is an object of the present invention to provide a controller that improves the calculation accuracy of the pumping loss torque and drives an internal combustion engine at an output torque that is closer to the target torque.

[0006] One aspect of the present invention provides a controller for an internal combustion engine including a cylinder housing a combustion chamber and retaining a reciprocating piston, an output shaft of a crankshaft rotated in cooperation with the piston, and a crank chamber accommodating the crankshaft. The piston has one side receiving intake air pressure and another side receiving pressure of the crank chamber. The controller includes a calculation means for calculating pumping loss torque using loss of output torque of the output shaft caused by a pumping loss that occurs when the piston reciprocates. The controller generates a corrected target torque by adding the pumping loss torque calculated by the calculation means to a target torque of the internal combustion engine. The controller determines a control parameter that affects the output torque based on the corrected target torque. The calculation means calculates the pumping loss torque based on the difference between the pressure of the crank chamber and the intake air pressure.

[0007] Another aspect of the present invention is a method for controlling an internal combustion engine including a cylinder housing a combustion chamber and retaining a reciprocating piston, an output shaft of a crankshaft rotated in cooperation with the piston, and a crank chamber accommodating the crankshaft. The piston has one side receiving intake air pressure and another side receiving pressure of the crank chamber. The method includes the steps of calculating the difference between the pressure of the crank chamber and the intake air pressure, calculating pumping loss torque using loss of output torque of the output shaft caused by a pumping loss that occurs when the piston reciprocates, generating a corrected target torque by adding the pumping loss torque calculated by the calculation means to a target torque of the internal combustion engine, and determining a control parameter that affects the output torque based on the corrected target torque (T). The step for calculating the pumping loss includes calculating the pumping loss torque based on the difference between the pressure of the crank chamber and the intake air pressure.

[0008] Other aspects and advantages of the present invention will become apparent from the following description, taken in conjunction with the accompanying drawings, illustrating by way of example the principles of the invention.

[0009] The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a schematic diagram showing the structure of a controller for an engine according to a first embodiment of the present invention;
Fig. 2 is a schematic diagram showing the structure of a blow-by gas recirculation device;
Fig. 3 is a flowchart showing the procedures for calculating the friction torque; and
Fig. 4 is a flowchart showing the procedures for controlling the output torque of the engine.

[0010] A controller for an internal combustion engine according to a first embodiment of the present invention will now be described.

[0011] Figs. 1 and 2 show a gasoline engine 11, which is mounted on a vehicle as an internal combustion engine. The engine 11 includes a cylinder block 13 having a plurality of cylinders 12. A crankcase 14 and an oil pan 15 are attached under the cylinder block 13. A cylinder head 16 is mounted on the cylinder block 13. A piston 17 reciprocates in each cylinder 12. A connecting rod 18 connects each piston 17 to a crankshaft 19, which includes an output shaft of the engine 11. The connecting rod 18 converts the reciprocating motion of each piston 17 to rotational motion of the crankshaft 19.

[0012] An intake passage 22 and an exhaust passage 23 are connected to a combustion chamber 21 of each cylinder 12. Air outside the engine 11 is drawn into each combustion chamber 21 via the intake passage 22. Exhaust gas is discharged out of each combustion chamber 21 and into the exhaust passage 23. An intake valve 24 and an exhaust valve 25 are arranged in the cylinder head 16 for each combustion chamber 21. The intake valve 24 opens and closes between the corresponding combustion chamber 21 and the intake passage 22. The exhaust valve 25 opens and closes between the corresponding combustion chamber 21 and the exhaust passage 23. The intake valve 24 is driven by an intake camshaft 26. The exhaust valve 25 is driven by an exhaust camshaft 27. The intake camshaft 26 and the exhaust camshaft 27, which are connected to the crankshaft 19 by a pulley and a belt, rotate together with the crankshaft 19.

[0013] A throttle valve 28 is arranged in the intake passage 22. The throttle valve 28 is rotated by an actuator 29, such as a motor. The amount of air flowing through the intake passage 22 changes according to the angle of the throttle valve 28 (throttle opening degree). The throttle opening degree is adjusted by changing the driving amount of the actuator 29 according to the amount an accelerator pedal 31 is depressed by the driver.

[0014] The engine 11 includes fuel injection valves 32, each provided for one of the cylinders 12. High-pressure fuel discharged from a fuel pump (not shown) is supplied to each fuel injection valve 32. Each fuel injection valve 32 is controlled to open and close so that high-pressure fuel is directly injected into the corresponding combustion chamber 21. The injected fuel is mixed with air in the combustion chamber 21 to form an air-fuel mixture.

[0015] The engine 11, in which fuel is directly injected from the fuel injection valve 32 into the combustion chamber 21 to form an air-fuel mixture, is usually referred to as a "direct in-cylinder injection engine." Instead of this type of engine, the present invention may be applied to a port injection engine. In the port injection engine, fuel is injected from the fuel injection valve, which is arranged in the intake passage 22, in a direction downstream from the intake port. The injected fuel is mixed with air that flows through the intake passage 22 to form an air-fuel mixture.

[0016] The engine 11 includes ignition plugs 33, each provided for one of the cylinders 12. Each ignition plug 33 operates in accordance with an ignition signal, which is generated by an igniter 34. An ignition coil 35 applies high voltage to the ignition plug 33. The ignition plug 33 generates an electric discharge to ignite and burn the mixture. This generates a high-temperature and high-pressure combustion gas. The combustion gas reciprocates the pistons 17 and rotates the crankshaft 19 to produce a driving force (output torque) of the engine 11. The exhaust valve 25 opens to discharge the combustion gas out of the combustion chamber 21 and into the exhaust passage 23.

[0017] During the compression stroke and the expansion stroke, gas leaks from gaps formed between the pistons 17 and the wall surfaces of the cylinders 12. The gas includes the mixture that leaks out during the compression stroke and the combustion gas leaking out during the expansion stroke. Such leakage gas is referred to as blow-by gas. The blow-by gas deteriorates the engine oil and rusts the engine 11. Therefore, the engine 11 includes a blow-by gas recirculation device 37. The blow-by gas recirculation device 37 recirculates the blow-by gas back to the intake system as indicated by the arrows drawn with solid lines in Fig. 2 so that the blow-by gas is burned again in the combustion chambers 21. A crank chamber 36, which is formed in the space encompassed by the cylinder block 13, the crankcase 14, and the oil pan 15. The crank chamber 36 accommodates the crankshaft 19.

[0018] The blow-by gas recirculation device 37 includes a blow-by gas passage 39. The blow-by gas passage 39 connects the crank chamber 36 to the intake passage 22 at a position downstream from the throttle valve 28. Negative pressure (pressure lower than atmospheric pressure), which is generated downstream from the throttle valve 28, is communicated to the crank chamber 36 via the blow-by gas passage 39. A positive crankcase ventilation (PCV) valve 41, which adjusts the recirculation amount of the blow-by gas, is arranged in the blow-by gas passage 39.

[0019] The blow-by gas recirculation device 37 includes an air intake passage 42 for drawing air (fresh air) into the crank chamber 36 from outside the engine 11 as indicated by the arrows drawn in broken line in Fig. 2. The fresh air drawn into the crank chamber 36 via the air intake passage 42 lowers the concentration of the blow-by gas in the crank chamber 36. The air intake passage 42 is connected to the intake passage 22 at a position upstream from the throttle valve 28. A head cover 43 is arranged on the cylinder head 16. The air intake passage 42 is connected to the crank chamber 36 via the head cover 43, the cylinder head 16, and the cylinder block 13.

[0020] As shown in Fig. 1, the engine 11 includes an exhaust gas recirculation (EGR) device 44. The EGR device 44 recirculates some of the exhaust gas flowing through the exhaust passage 23 back to the intake passage 22. The exhaust gas recirculated back to the intake passage 22 (EGR gas) is mixed with the intake air. As a result, the concentration of inert gas in the mixture increases and the maximum combustion temperature decreases. This suppresses generation of nitrogen oxide (NOx) and reduces exhaust emissions.

[0021] The EGR device 44 includes an EGR passage 45 and an EGR valve 46. The EGR passage 45 connects the exhaust passage 23 to the intake passage 22 at a position downstream from the throttle valve 28. The EGR valve 46 is arranged in the EGR passage 45. The negative pressure generated downstream from the throttle valve 28 in the intake passage 22 is communicated to the exhaust passage 23 via the EGR passage 45. Some of the exhaust gas discharged from the exhaust passage 23 is recirculated back to the intake passage 22 as EGR gas via the EGR passage 45. The flow amount of the EGR gas changes in accordance with the opening degree of the EGR valve 46 (EGR opening degree).

[0022] Various auxiliary devices (not shown) are attached to the engine 11. The auxiliary devices include, for example, an alternator, a power-steering pump, an air conditioner compressor, an engine oil pump, and an engine water pump. Each auxiliary device has an output shaft connected to the crankshaft 19 via a pulley and a belt in order to rotate together with the crankshaft 19.

[0023] The vehicle includes various sensors for detecting the state of various parts in the vehicle, including the driving state of the engine 11. For example, a crank angle sensor 51 is arranged near the crankshaft 19. The crank angle sensor 51 generates a pulse signal every time the crankshaft 19 is rotated by a fixed angle. The pulse signal generated by the crank angle sensor 51 is used to calculate the crank angle, which is the rotation angle of the crankshaft 19, and the rotation speed of the crankshaft 19 per unit time, or the engine speed.

[0024] An intake air pressure sensor 52 is arranged downstream from the throttle valve 28 in the intake passage 22. The intake air pressure sensor 52 detects the intake air pressure epim (absolute pressure). An atmospheric pressure sensor 53 is arranged in the passenger compartment. The atmospheric pressure sensor 53 detects the atmospheric pressure epa, which changes in accordance with the weather and the altitude (e.g., sea level or highland level). An accelerator sensor 54 is arranged on or near the accelerator pedal 31. The accelerator sensor 54 detects the amount the accelerator pedal 31 is depressed by the driver.

[0025] An electronic control unit (ECU) 56, which is mainly formed by a microcomputer, controls various parts of the engine 11 based on the detection values of the sensors 51 to 54. The ECU 56 includes a central processing unit (CPU). The CPU performs calculations in accordance with control programs and initial data stored in a read only memory (ROM) and executes various controls based on the calculation results. The calculation result of the CPU is temporarily stored in a random access memory (RAM).

[0026] The ECU 56 determines values for control parameters that affect the output torque of the engine 11 so that the output torque of the engine 11 becomes equal to the torque required by the driver (target torque Tt). This control is referred to as "output torque control" or "torque demand control." Basically, the target torque Tt is calculated in accordance with the engine speed and the amount the accelerator pedal 31 is depressed by the driver. However, when the ECU 56 drives the engine 11 using the target torque as its command value, the actual output torque generated by the engine 11 is lower than the target torque Tt due to the friction produced when the engine 11 is driven (friction loss). Thus, the ECU 56 calculates the torque that is lowered (consumed) by such friction as a friction torque Tf. The ECU 56 adds the friction torque Tf to the target torque Tt to correct the target torque Tt. In this way, the ECU 56 generates a final target torque T. Based on the final target torque T, the ECU 56 determines values for the control parameters that affect the output torque of the engine 11.

[0027] The above friction loss includes pumping loss, mechanical loss, and auxiliary device loss. The pumping loss is the loss of pressure caused by the resistance of intake air in the throttle valve 28 and the EGR valve 46. The mechanical loss is caused by the friction produced in movable parts of the engine 11. The auxiliary loss is caused by the friction produced when driving the auxiliary devices. The mechanical loss and the auxiliary loss are directly determined by the viscosity of oil. The viscosity of oil does not change drastically but changes gradually when the engine is being driven. The viscosity of oil is estimated based on parameters such as the oil temperature and the coolant temperature. Thus, the mechanical loss and the auxiliary loss are estimated based on the oil temperature and the coolant temperature. Basically, the throttle valve 28 and the EGR valve 46 are constantly operated. Thus, the pumping loss constantly changes. In comparison with the mechanical loss and the auxiliary loss, the pumping loss is more difficult to calculate (estimate). The calculation (estimation) of the friction torque Tf with high accuracy is important for accurate calculation of the target torque Tt.

[0028] The calculation of the friction torque Tf and the control of the output torque of the engine 11 based on the friction torque Tf will now be described with reference to the flowcharts shown in Figs. 3 and 4.

[0029] Fig. 3 is a flowchart of a friction torque calculation routine. In steps S110 and S120, the ECU 56 calculates the pumping loss torque Tp. In step S110, the ECU 56 calculates the pressure difference ΔP between the intake air pressure epim and the crank chamber pressure epcr. The intake air pressure epim is, for example, the detection value of the intake air pressure sensor 52.

[0030] The reason for using the intake air pressure epim to calculate the pumping loss torque Tp will now be described.

[0031] The pumping loss is normally large when the throttle opening degree is small. The pumping loss decreases as the throttle opening degree increases. Thus, to calculate the pumping loss torque Tp, a map, which is generated in advance and which defines the relationship between the throttle opening degree and the pumping loss torque Tp, may referred to in order to determine the pumping loss torque Tp corresponding to any given throttle opening degree. However, the relationship between the throttle opening degree and the pumping loss torque Tp may deviate from the relationship defined by the map when, for example, the amount of deposits adhered to the throttle valve 28 increases thereby increasing the resistance of intake air. Such deviation of the relationship may also be caused by factors other than the deposits adhered to the throttle valve 28 such as wear of the throttle valve 28 or characteristic of each throttle valve 28.

[0032] The EGR opening degree has a similar relationship with the pumping loss torque Tp as the throttle opening degree has with the pumping loss torque Tp. The pumping loss decreases as the EGR opening degree increases. Thus, to calculate the pumping loss torque Tp, a map, which is generated in advance and which defines the relationship between the EGR opening degree and the pumping loss torque Tp, may also be referred to in order to determine the pumping loss torque Tp corresponding to any given EGR opening degree. However, like the problem caused by the throttle valve 28, the characteristic of each EGR valve 46 or wear of the EGR valve 46 may deviate the relationship between the EGR opening degree and the pumping loss torque Tp from that defined by the map.

[0033] By referring to each map as described above, the pumping loss torque Tp may be calculated based on the throttle opening degree and the EGR opening degree. However, the two parameters, the throttle opening degree and the EGR opening degree, may change at the same time. In such a case, the change in the throttle opening degree and the change in the EGR opening degree have a combined influence on the pumping loss torque Tp. This would change the pumping loss torque Tp in a complex manner. Thus, the pumping loss torque Tp would deviate from a value calculated by simply adding or subtracting the pumping loss torque Tp determined from the map in correspondence with the throttle opening degree or the EGR opening degree.

[0034] The intake air pressure epim is influenced by both of the throttle opening degree and the EGR opening degree. Thus, the intake air pressure epim reflects the influence of the throttle opening degree and the influence of the EGR opening degree. The intake air pressure epim further reflects a combined influence of a change in the throttle opening degree and a change in the EGR opening degree when such changes occur at the same time. The intake air pressure epim also reflects the influence of the characteristic of the throttle valve 28 or the EGR valve 46 and the influence of wear of the throttle valve 28 or the EGR valve 46. Thus, the intake air pressure epim is a parameter that reflects all of these influences. Use of the intake air pressure epim to calculate the pumping loss torque Tp eliminates the above problems that would occur when using values determined from maps for the throttle opening degree and the EGR opening degree. Accordingly, in the first embodiment, the pumping loss torque Tp is calculated using the intake air pressure epim instead of using maps.

[0035] The reason for using the crank chamber pressure epcr to calculate the pumping loss torque Tp will now be described. The pressure of the crank chamber 36 influences the descent of the piston 17. For example, when the pressure of the crank chamber 36 is high, the descent of the piston 17 is hindered. In this case, the pumping loss is small. Accordingly, to be accurate, in addition to the pressure in the combustion chamber 21 (intake air pressure epim), the crank chamber pressure epcr also influences the pumping loss torque Tp.

[0036] In the blow-by gas recirculation device 37, the negative pressure downstream from the throttle valve 28 is merely communicated to the crank chamber 36 to recirculate the blow-by gas back to the intake system. Thus, the value of the crank chamber pressure epcr is substantially the same as the value of the atmospheric pressure epa. For this reason, in the first embodiment, the atmospheric pressure epa detected by the atmospheric pressure sensor 53 is used as a value relating to the crank chamber pressure epcr.

[0037] In step S120, the ECU 56 calculates the pumping loss torque Tp using equation (i).

$$Tp = \Delta P * C * K \quad \text{(i)}$$

[0038] In equation (i), C is a coefficient for converting pressure (pressure difference ΔP) to torque. K is a correction coefficient for reflecting influences of parameters other than the pressure difference ΔP on the pumping loss torque Tp to calculate the pumping loss torque Tp. The correction coefficient K is, for example, a basic correction term k1 corresponding to an influence of the blow-by gas on the pumping loss torque Tp.

[0039] In step S130, the ECU 56 adds the mechanical loss torque Tm and the auxiliary device loss torque Ta to the pumping loss torque Tp to calculate the friction torque Tf. Values that are estimated with the oil temperature and the coolant temperature, which are parameters relating to the viscosity of the oil, are used as the mechanical loss torque Tm and the auxiliary device loss torque Ta. These values are calculated in another routine. The friction torque calculation routine ends after step S130.

[0040] The ECU 56 in executing steps S110 and S120 in the friction torque calculation routine functions as a calculation means.

[0041] The output torque control routine shown in Fig. 4 will now be described. In step S210, the ECU 56 reads the friction torque Tf, which is calculated in the friction torque calculation routine of Fig. 3, and the target torque

Tt, which is in accordance with the depression of the accelerator pedal 31 by the driver. The target torque Tt is calculated based on the depression amount of the accelerator pedal 31 and the engine speed, for example, in another routine.

[0042] In step S220, the ECU 56 adds the friction torque Tf to the target torque Tt to correct the target torque Tt. In this way, the ECU 56 generates the final target torque T.

[0043] The ECU 56 in executing step S220 in Fig. 4 functions as a correction means for correcting the target torque Tt.

[0044] In step 5230, the ECU 56 determines values for control parameters that influence the output torque of the engine 11. to achieve the final target torque T. The control parameters differ depending on the type of the engine 11. For the gasoline engine 11 of the first embodiment, the control parameters include the intake air amount, the fuel injection amount, and the ignition timing. The ECU 56 calculates the values of the throttle opening degree, the fuel injection amount, the ignition timing, etc. to achieve the final target torque T using a predetermine map and an operational expression. The ECU 56 controls the actuator 29, the fuel injection valve 32, and the igniter 34 based on the calculated values. Theoretically, the engine 11 generates an output torque substantially equal to the final target torque T. However, the engine 11 actually generates an output torque that is lower than the final target torque T by a value corresponding to the friction loss. Consequently, the engine 11 generates an output torque that is substantially equal to the target torque Tt required by the driver. The output torque control routine ends after step S230. The ECU 56 executing step S230 in Fig. 4 functions as a determination means for determining the control parameters.

[0045] The first embodiment has the advantages described below.

(1) The intake air pressure epim, which is a parameter reflecting the influence of the throttle opening degree and the influence of the EGR opening degree, is used to calculate the pumping loss torque Tp (step S110). This reflects the characteristics of the throttle valve 28 and the EGR valve 46 and changes in the flow amount of intake air in the calculation of the pumping loss torque Tp. Thus, the pumping loss torque Tp is calculated with high accuracy.

(2) The crank chamber pressure epcr and the intake air pressure epim are applied to both sides of the reciprocating piston 17 (upper and lower sides of the piston 17 in Fig. 1). The crank chamber pressure epcr and the intake air pressure epim are parameters influencing the pumping loss torque Tp. The pumping loss torque Tp is calculated based on the pressure difference ΔP between the crank chamber pressure epcr and the intake air pressure epim. Thus, in comparison with when using only the intake air pressure or the total gas amount in the cylinders (Japanese Laid-Open Patent Publication No. 11-62658), the calculation accuracy of the pumping loss torque Tp is improved and the calculation accuracy of the corrected target torque is improved. The engine 11 generates an output torque that is substantially equal to the target torque Tt even when, for example, the altitude of the vehicle changes (from sea level to highland level) and the atmospheric pressure alters when the vehicle is being driven.

(3) The correction coefficient K is set so that influences of parameters other than the pressure difference ΔP on the pumping loss torque Tp are reflected in calculation of the pumping loss torque Tp. The pressure difference ΔP is multiplied by the correction coefficient K. This multiplication further improves the calculation accuracy of the pumping loss torque Tp.

A controller for an internal combustion engine according to a second embodiment of the present invention will now be described. The second embodiment differs from the first embodiment in step S120 in the friction torque calculation routine shown in Fig. 3. More specifically, the correction coefficient K is calculated using equation (ii) in step S120.

$$K = k1 + k2 \qquad \text{(ii)}$$

In equation (ii), k1 is the basic correction term, which is described above, and k2 is an auxiliary correction term for calculating the pumping loss torque per cylinder. The auxiliary correction term is determined by a function of the area S of the top surface of the piston 17 to which the pressure in the combustion chamber 21 is applied. Normally, the area to which the pressure difference ΔP is applied increases as the area S increases. Under the same pressure difference ΔP, the force necessary to descend the piston 17 increases and the pumping loss Tp increases as the area S increases. Thus, the auxiliary correction term k2 is set to increase as the top surface area S increases. Except for this point, the processing in the second embodiment is the same as the processing in the first embodiment.

The second embodiment has the advantages described below in addition to advantages (1) to (3), which are described above.

(4) The correction coefficient K reflects the correction term (auxiliary correction term k2) corresponding to the top surface area S of the piston 17 that affects the pumping loss torque Tp. Thus, in comparison with when the auxiliary correction term k2 is not used (the first embodiment), the calculation

accuracy of the pumping loss torque Tp is further improved.

(5) The auxiliary correction term k2 corresponding to the area S of the top surface of the piston is used to calculate the pumping loss torque Tp. Thus, even when calculating the pumping loss torque Tp of a plurality of engines, which differ from one another in the area S, the correction coefficient K is easily determined. Thus, the present invention may easily be applied to various types of engines.

A controller for an internal combustion engine according to a third embodiment of the present invention will now be described. The third embodiment differs from the second embodiment in step S120 in the friction torque calculation routine shown in Fig. 3. More specifically, the correction coefficient K is calculated using equation (iii) in step S120.

$$K = k1 + (k2 * n) \qquad \text{(iii)}$$

In equation (iii), n is the number of cylinders included in the engine 11. The auxiliary correction term k2 is multiplied by the number of cylinders n for the following reason. In the engine 11 that has a plurality of cylinders, pumping loss occurs in each of the cylinders. Thus, the total pumping loss of the entire engine 11 is obtained by multiplying the pumping loss per cylinder by the number of cylinders n. Except for this point, the processing in the third embodiment is the same as the processing in the first embodiment.

The third embodiment has the advantages described below in addition to advantages (1) to (5), which are described above.

(6) The auxiliary correction term k2 is multiplied by the number of cylinders n, which affects the pumping loss torque Tp. Thus, in comparison with the calculation of the second embodiment, the calculation accuracy of the pumping loss torque Tp is further improved.

(7) The auxiliary correction term k2 is multiplied by the number of cylinders n. Thus, even when calculating the pumping loss torque Tp of various types of engines, which differ from one another in the top surface area S and in the number of cylinders included, the correction coefficient K is easily determined. Thus, the present invention is easily applied to various types of engines.

A controller for an internal combustion engine according to a fourth embodiment of the present invention will now be described. In the fourth embodiment, an estimated value for the crank chamber pressure epcr is used in step S110 of the friction torque calculation routine shown in Fig. 3.

The estimation of the crank chamber pressure epcr considers the following points. Normally, a change in the driving state of the engine, such as the engine speed and the engine load, accordingly changes the negative pressure generated downstream from the throttle valve 28 in the intake passage 22. The negative pressure is communicated to the crank chamber 36 via the blow-by gas passage 39. Thus, the degree of influence the negative pressure has on the crank chamber pressure epcr varies. Accordingly, in the fourth embodiment, the ECU 56 determines the degree of influence the engine speed and the engine load has on the crank chamber pressure epcr (influence coefficient L). For example, a map defining the relationship between the engine speed and the influence coefficient L and the relationship between the engine load and the influence coefficient L is generated in advance and used to determine the influence coefficient L. The map associates various combinations of the engine speed and the engine load with values of the influence coefficient L, which represents the percentage of change in the crank chamber pressure epcr with respect to the atmospheric pressure epa.

The ECU 56 uses equation (iv) to estimate the crank chamber pressure epcr based on the influence coefficient L calculated from the map as described above and the atmospheric pressure epa detected by the atmospheric pressure sensor 53.

$$epcr = epa * L \qquad \text{(iv)}$$

The ECU 56 functions as an estimation means for estimating the crank chamber pressure epcr.

The crank chamber pressure epcr calculated (estimated) from equation (iv) is used to calculate the pressure difference ΔP in step S110. Except for this point, the processing in the fourth embodiment is the same as the processing in each of the first to third embodiments.

The fourth embodiment has the advantages described below in addition to advantages (1) to (7), which are described above.

(8) The crank chamber pressure epcr is estimated by multiplying the atmospheric pressure epa by the influence coefficient L. The estimated value is used to calculate the pressure difference ΔP. Thus, in comparison with when simply using the atmospheric pressure epa as the crank chamber pressure epcr, the calculation accuracy of the pumping loss torque Tp is further improved.

[0046] The first to fourth embodiments may be modified in the following forms.

[0047] The auxiliary correction term k2 in the second embodiment may be determined based on a value representing the top surface area S, such as the inner diameter (bore diameter) of the cylinder 12, diameter of the piston 17, a function of the internal diameter of the cylinder 12, or a function of the diameter of the piston 17.

[0048] The crank chamber pressure epcr may be estimated through a process that differs from that of the fourth embodiment. For example, since the capacity of the crank chamber 36 is fixed, the crank chamber pressure epcr increases by an amount corresponding to an increase in the generation amount of the blow-by gas. Further, when the engine load changes, the negative pressure drawing in the blow-by gas changes and alters the crank chamber pressure epcr. Normally, the negative pressure decreases as the engine load increases. Thus, the generated amount of the blow-by gas and the engine load are parameters that influence the crank chamber pressure epcr. Focusing on this point, the crank chamber pressure epcr may be estimated using a predetermined operational expression, based on the capacity of the crank chamber 36, the generation amount of the blow-by gas, and the engine load.

[0049] As indicated by the broken line in Fig. 1, a pressure sensor 61 for directly detecting the pressure of the crank chamber 36 may be used as a pressure detection means. The detection value of the pressure sensor 61 is used as the crank chamber pressure epcr in the first to third embodiments. In this case, the calculation accuracy of the pumping loss torque Tp is further improved.

[0050] In step S130 in Fig. 3, the pumping loss torque Tp is necessary for calculation of the friction torque Tf. However, the terms other than the pumping loss torque Tp (the mechanical loss torque Tm and the auxiliary device loss torque Ta) may be freely changed. For example, a term representing a further loss torque may additionally be used in the calculation.

[0051] The first to fourth embodiments are applicable to engines other than a gasoline engine, such as a diesel engine. Further, the first to fourth embodiments are also applicable to engines that do not include the blow-by gas recirculation device 37.

Claims

1. A controller for an internal combustion engine (11) including a cylinder (12) housing a combustion chamber (21) and retaining a reciprocating piston (17), an output shaft of a crankshaft (19) rotated in cooperation with the piston, and a crank chamber (36) accommodating the crankshaft, wherein the piston has one side receiving intake air pressure (epim) and another side receiving pressure (epcr) of the crank chamber, the controller including a calculation means (S120) for calculating pumping loss torque using loss of output torque of the output shaft caused by a pumping loss that occurs when the piston reciprocates, wherein the controller generates a corrected target torque (T) by adding the pumping loss torque calculated by the calculation means to a target torque (Tt) of the internal combustion engine, and determines a control parameter that affects the output torque based on the corrected target torque (T), the controller being **characterized in that** the calculation means calculates the pumping loss torque based on the difference (ΔP) between the pressure of the crank chamber and the intake air pressure.

2. The controller according to claim 1 **characterized in that** the pressure of the crank chamber is atmospheric pressure.

3. The controller according to claim 1 or 2 further being **characterized by**:

   an estimation means (56) for estimating the pressure of the crank chamber using atmospheric pressure that is corrected in accordance with a driving state of the internal combustion engine;

   wherein the calculation means calculates the pumping loss torque using a value of the pressure of the crank chamber estimated by the estimation means.

4. The controller according to claim 1 further being **characterized by**:

   a pressure detection means (61) for detecting the pressure of the crank chamber;

   wherein the calculation means calculates the pumping loss torque using the pressure of the crank chamber detected by the pressure detecting means.

5. The controller according to any one of claims 1 to 4 **characterized in that**:

   the one side of the piston is a top surface of the piston; and
   the calculation means corrects the pressure difference in accordance with area of the top surface of the piston or a value representing the area to calculate the pumping loss torque using the corrected pressure difference.

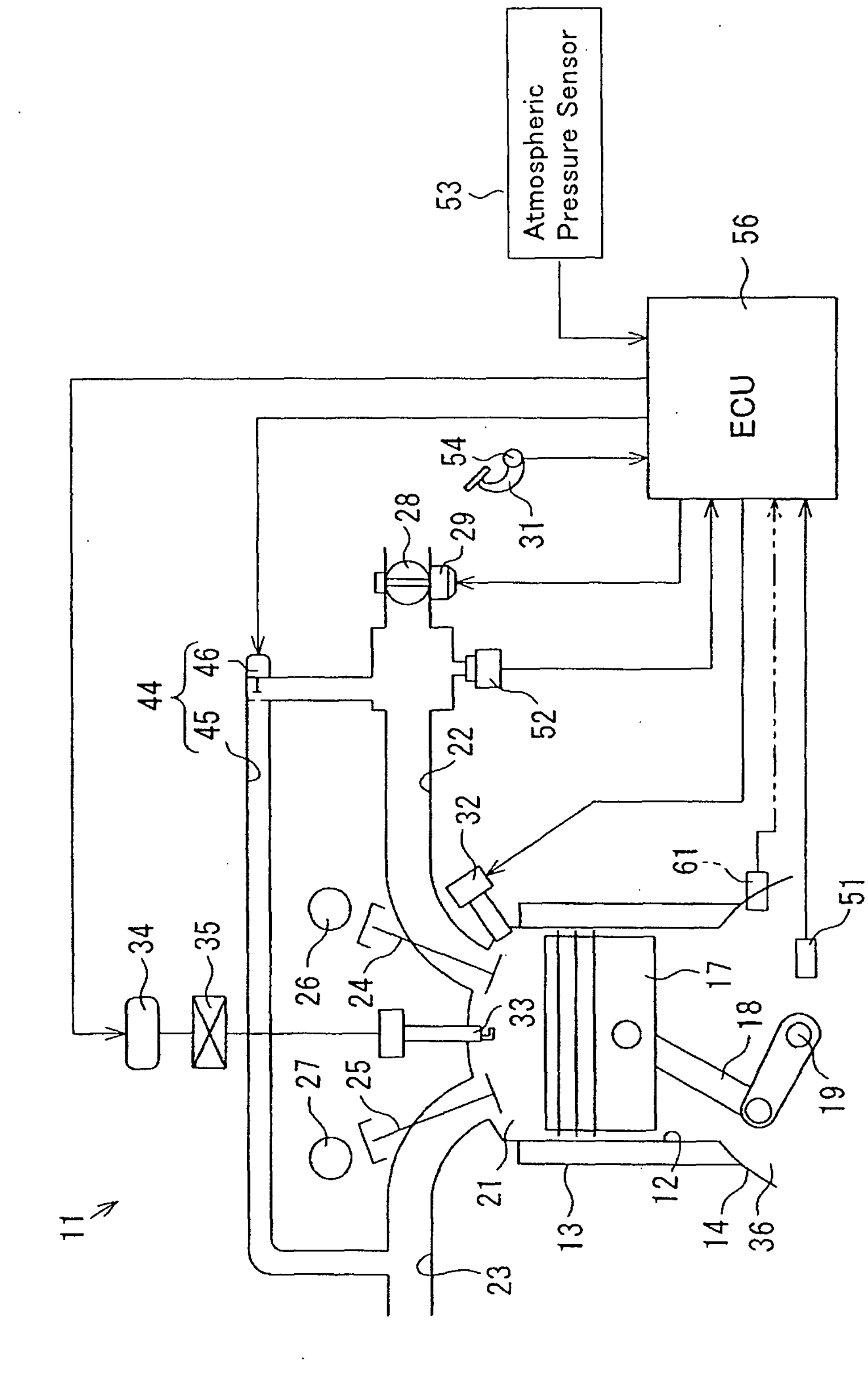
6. The controller according to any one of claims 1 to 5, **characterized in that** the calculation means corrects the pressure difference in accordance with the number of cylinders included in the internal combustion engine to calculate the pumping loss torque using the corrected pressure difference.

7. A method for controlling an internal combustion en-

gine (11) including a cylinder (12) housing a combustion chamber (21) and retaining a reciprocating piston (17), an output shaft of a crankshaft (19) rotated in cooperation with the piston, and a crank chamber (36) accommodating the crankshaft, wherein the piston has one side receiving intake air pressure (epim) and another side receiving pressure (epcr) of the crank chamber, the method including:

calculating the difference between the pressure of the crank chamber and the intake air pressure (S110);
calculating pumping loss torque using loss of output torque of the output shaft caused by a pumping loss that occurs when the piston reciprocates (S120);
generating a corrected target torque (T) by adding the pumping loss torque calculated by the calculation means to a target torque (Tt) of the internal combustion engine (S220); and
determining a control parameter that affects the output torque based on the corrected target torque (T) (S239), the method being **characterized in that** the step for calculating the pumping loss includes calculating the pumping loss torque based on the difference (ΔP) between the pressure of the crank chamber and the intake air pressure.

Fig. 1


Atmospheric Pressure Sensor
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ECU
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Fig. 2

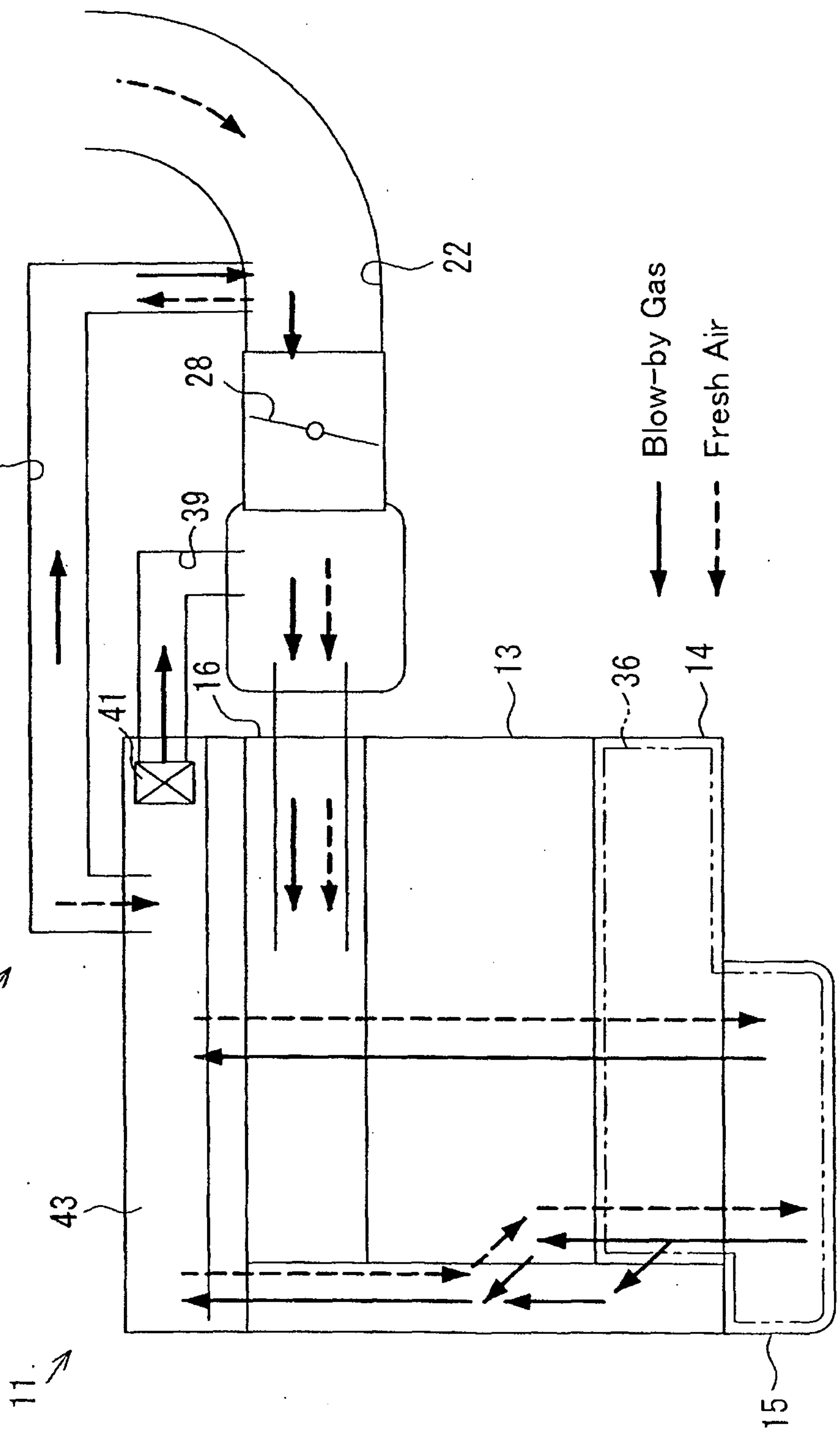


Fig. 3

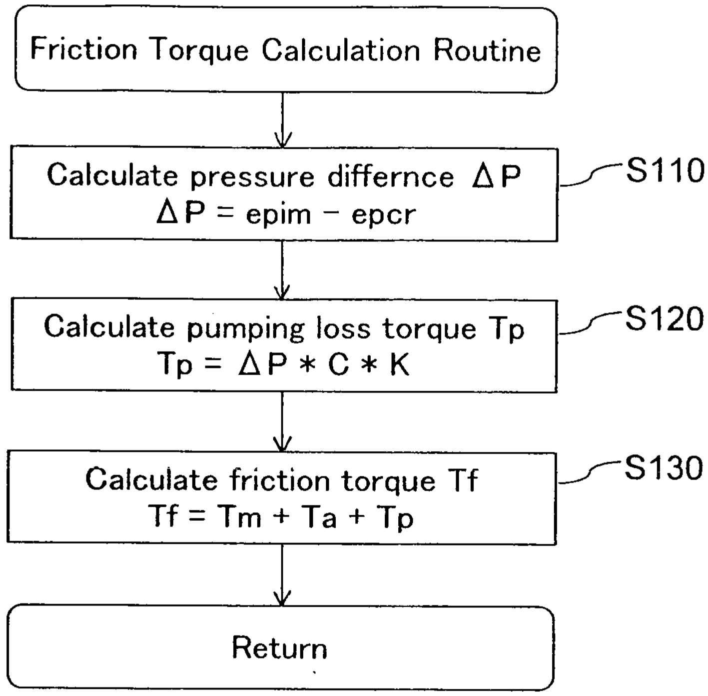
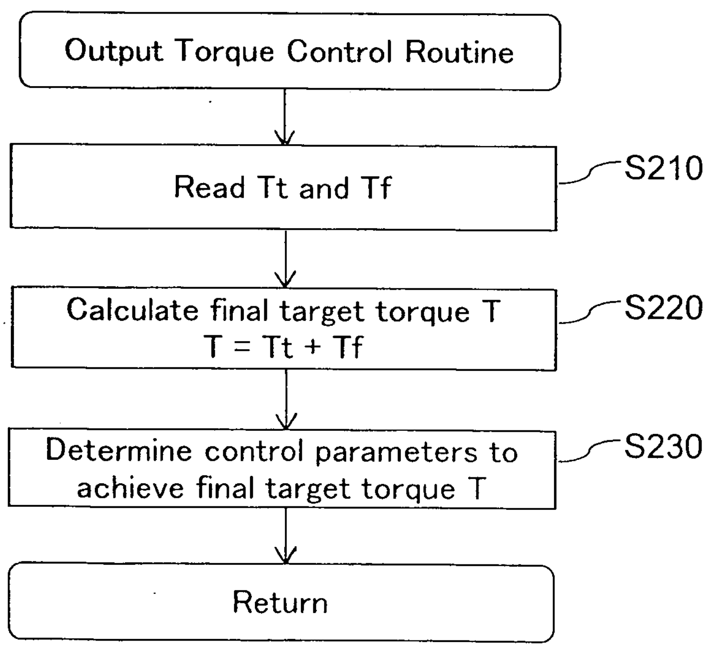

Friction Torque Calculation Routine
Calculate pressure differnce ΔP
ΔP = epim − epcr
S110
Calculate pumping loss torque Tp
Tp = ΔP * C * K
S120
Calculate friction torque Tf
Tf = Tm + Ta + Tp
S130
Return

Fig. 4


Output Torque Control Routine
Read Tt and Tf
S210
Calculate final target torque T
T = Tt + Tf
S220
Determine control parameters to
achieve final target torque T
S230
Return