Title: CONTROL DEVICE FOR VEHICULAR HYDRAULIC CONTROL CIRCUIT

Abstract: The present invention provides a control device of vehicular hydraulic control circuit capable of reducing a load instantaneously applied to an electric oil pump disposed in the vehicular hydraulic control circuit upon starting the electric oil pump. Command control value setting means (120), based on a compared result between a line pressure PL (actual load value) and a target line pressure PL,wd (target load value) by the electric oil pump (72), sets a command rotation speed Nopmtg (command control value). Hence, with suppressing a high load applied to the electric oil pump (72) generated upon starting the electric oil pump (72), the responsiveness of the hydraulic pressure can be secured. As a result, an excessive current generated in an electronic control circuit for the control device controlling the electric oil pump (72) can be suppressed.
DESCRIPTION

CONTROL DEVICE FOR VEHICULAR HYDRAULIC CONTROL CIRCUIT

TECHNICAL FIELD
[0001] The present invention relates to a control device for vehicular hydraulic control circuit for supplying a hydraulic pressure required for a hydraulic actuator and the like driven by a hydraulic pressure of a vehicle. In particular, the hydraulic control circuit is provided with an electric oil pump driven by an electric power.

BACKGROUND ART
[0002] A hydraulic actuator such as a clutch for clutch provided for a vehicle is supplied with a suitable hydraulic pressure i.e. oil pressure regulated. Such hydraulic actuator, for instance, is supplied with a hydraulic pressure of the operation oil drawn up, for instance, by a mechanical oil pump or an electric oil pump as original pressure.
[0003] For instance, a vehicular hydraulic control circuit disclosed in Patent Document 1 (Japanese Patent Publication No. 2003-307271A) is equipped with a mechanical oil pump and a electric oil pump driven by an engine. During stopping of the engine, the electric oil pump is controllably driven instead of the mechanical oil pump stopped, and during driving of the mechanical oil pump, the electric oil pump is controlled so as to be stopped, by a control device. Patent Document 1 discloses a control technique to set, upon starting the electric oil pump, a target rotation speed further higher than the predetermined target rotation speed according to a vehicle state, thereby to improve the responsiveness of the hydraulic pressure.
[0004] Now, the vehicular hydraulic control circuit equipped with the electric oil pump including that in Patent Document 1 is generally a closed system. Hence, for securing the hydraulic pressure upon starting the electric oil pump from the stopping state of the electric oil pump, a load applied to the electric oil pump may become large. Particularly, when viscosity of the operation oil is high and the like, there was a fear that the load applied to the electric oil pump becomes instantaneously extremely large. Accompanied with this, there was a fear that an excess current is generated in an electronic control circuit of the control device controlling the electric power supplied to the electric oil pump.
DISCLOSURE OF THE INVENTION

[0005] The present invention has been made in view of the above described circumstances, and has an object to provide a control device of vehicular hydraulic control circuit, capable of reducing a load instantaneously applied to an electric oil pump upon starting the electric oil pump.

[0006] For achieving the above object, the present invention relate to a control device for vehicular hydraulic control circuit. In a first aspect, the vehicular hydraulic control circuit includes an electric oil pump. The control device is comprised of an actual load value detecting portion for obtaining or estimating an actual load value of the electric oil pump, a target load value setting portion for setting a target load value of the electric oil pump, and a command control value setting portion for setting a command control value of the electric oil pump. The command control value setting portion sets a command control value based on a compared result between the actual load value and the target load value of the electric oil pump.

[0007] In a second aspect of the invention, the command control value setting means sets the command control value based on a value obtained by multiplying a difference between the actual load value and the target load value of the electric oil pump by a proportionality constant, the proportionality constant being set based on an oil temperature of an operation oil.

[0008] In a third aspect of the invention, wherein the actual load value of the electric oil pump corresponds to a current value of the electric oil pump or a value of the hydraulic pressure of the operation oil discharged by the electric oil pump.

[0009] In a fourth aspect of the invention, the command control value of the electric oil pump is set based on the compared result between the target control value and the actual control value of the electric oil pump.

[0010] In a fifth aspect of the invention, the actual load value detecting portion estimates the actual load value of the electric oil pump based on a detection value correlated with the actual load value of the electric oil pump.

[0011] According to the control device for vehicular hydraulic control circuit of the invention in the first aspect, the command control value setting portion sets the command control value based on the compared result between the actual load value and the target load value of the electric oil pump. Hence, with suppressing a high load generated upon starting the electric oil pump and applied thereto, the responsiveness of
the hydraulic pressure can be secured. As a result, the excessive current generated in the electronic control circuit for controlling the electric oil pump can be suppressed.

[0012] According to the control device for vehicular hydraulic control circuit of the invention in the second aspect, the command control value setting portion sets the command control value based on the value obtained by multiplying the difference between the actual load value and the target load value of the electric oil pump by the proportionality constant set based on the oil temperature of the operation oil. Hence, an appropriate control can be performed according to the oil temperature of the operation oil. For instance, for the low oil temperature of the operation oil the proportionality constant is set high, and for the high oil temperature of the operation oil the proportionality constant is set low, so that with suppressing the high load applied to the electric oil pump, the responsiveness of the hydraulic pressure can be further secured efficiently.

[0013] According to the control device for vehicular hydraulic control circuit of the invention in the third aspect, the actual load value of the electric oil pump corresponds to the current value of the electric oil pump or the value of the hydraulic pressure of the operation oil discharged by the electric oil pump. Hence detecting the current value or hydraulic pressure enables easy detection of the actual load value at the electric oil pump.

[0014] According to the control device for vehicular hydraulic control circuit of the invention in the fourth aspect, the command control value of the electric oil pump is set based on the compared result between the target control value and the actual control value of the electric oil pump. Hence, a rapid change of the control value can be suppressed.

[0015] According to the control device for vehicular hydraulic control circuit of the invention in the fifth aspect, based on the detection value correlated with the actual load value of the electric oil pump, the actual load value of the electric oil pump is estimated. Hence, not only the actual load value of the electric oil pump can be easily estimated, but a relative error to the actual load value can be also suppressed in a range not affecting the present control.

As the detection value correlated with the actual load value of the electric oil pump, for instance, a temperature of the electric oil pump, a current value or the estimation value of the current value of the control device of the electric oil pump, a
temperature of the control device of the electric oil pump, a value or the estimated value of the hydraulic pressure of an engagement element, and the like are included.

[0016] The command control value setting portion preferably adds a value (feed forward value) variable depending on the oil temperature and the like of the operation oil to the command control value, when a difference between an actual load value and a target load value of the electric oil pump is equal to or more than the predetermined value. In this manner, the feed forward value can suppress a high load applied to the electric oil pump.

[0017] A power transmitting apparatus of the vehicle preferably includes an electric differential portion to control the operating state of the motor connected to the rotation element of the differential mechanism in a power-transmissive state, so that a differential state between the rotation speed of the input shaft to which the engine is connected and the rotation speed of the output shaft is controlled. In this manner, controlling the electric motor enables varying the speed change ratio continuously, thus obtaining a broad driving force range.

[0018] A mechanical oil pump additionally provided in the hydraulic control circuit is preferably connected to the engine. In this manner, driving the mechanical oil pump during driving the engine enables sufficient hydraulic pressure to be obtained. Further, no need for provision of a driving source for driving the mechanical oil pump separately can eliminate increase the number of parts.

An electrical differential portion is preferably configured by a planetary gear unit and two electric motors. By being so configured, the electric motor can controls the rotation speed of each rotation element of the planetary gear unit, and the structure thereof can be relatively made compact.

BRIEF DESCRIPTION OF THE DRAWINGS

[0019] Fig. 1 is a skeleton view explaining a structure of a hybrid vehicle drive apparatus to which is one of embodiments of the present invention.

Fig. 2 is an engagement operation table illustrating the relationship between a shifting operation, in which the hybrid vehicle drive apparatus, shown in Fig. 1, is placed in a continuously variable or step-variable shifting state, and the operation of a hydraulic-type frictional engaging device in combination.
Fig. 3 is a collinear chart illustrating the relative rotation speed of rotary elements in each of different gear positions when the hybrid vehicle drive apparatus, shown in Fig. 1, is caused to operate in the step-variable shifting state.

Fig. 4 is a view illustrating input and output signals to be input to or output from an electronic control device incorporated in the hybrid vehicle drive apparatus shown in Fig. 1.

Fig. 5 is a view showing one sample of a shift operating device for operating to select one of plural kinds of shift positions by a manual operation.

Fig. 6 is a functional block diagram illustrating a major control operation to be executed by the electronic control device shown in Fig. 4.

Fig. 7 is a view representing one example of a preliminarily stored shifting diagram, plotted on a two-dimensional coordinate in terms of parameters including a vehicle speed and output torque based on which the operation is executed whether to a shifting is executed in an automatic shifting portion; one example of preliminarily stored diagram based on which a shifting state of the shifting mechanism is switched; and one example of a preliminarily stored drive-force source switching diagram having a boundary line between an engine drive region and a motor drive region based on which an engine drive mode and a motor drive mode is switched.

Fig. 8 is a conceptual view, showing the preliminarily stored relationship, involving a boundary line, between a continuously variable control region and a step-variable control region, which is suitable for mapping a boundary between the continuously variable control region and the step-variable control region shown in broken lines in Fig. 7.

FIG. 9 is a circuit diagram relating to a linear solenoid valve for controlling an operation of each the hydraulic pressure actuators for clutches C and brakes B which are a part of the hydraulic control circuit, and a circuit diagram schematically showing configurations of a mechanical oil pump, an electric oil pump, and a regulator valve.

FIG. 10 is a view showing a relationship between a load applied to the electric oil pump and a rotation speed.

FIG. 11 is a flowchart for explaining a control operation for reducing the load applied to the electric oil pump upon starting the electric oil pump from a state where main parts of the control operation of an electronic control device, i.e. both the mechanical oil pump and the electric oil pump are in a stopped state.
FIG. 12 shows one example of the control operation shown in a flowchart of FIG. 11, and is a time chart showing a starting control upon starting the electric oil pump at the stopping time of the engine.

BEST MODE FOR CARRYING OUT THE INVENTION

[0020] Now, various embodiments of the present invention will be described below in detail with reference to accompanying drawings.

<Embodiment 1>

Fig. 1 is a skeleton view illustrating a shifting mechanism 10, forming part of a power transmitting apparatus for a hybrid vehicle, to which a control device of one embodiment according to the present invention is applied. As shown in Fig. 1, the shifting mechanism 10 includes an input shaft 14 serving as an input rotary member, a differential portion 11 directly connected to the input shaft 14 or indirectly connected thereto through a pulsation absorbing damper (vibration damping device) not shown, an automatic shifting portion 20 connected via a power transmitting member (corresponding to an output shaft of differential mechanism) 18 in series through a power transmitting path between the differential mechanism 11 and drive wheels 38 (see Fig. 6) to serve as a step-variable type transmission, and an output shaft 22 connected to the automatic shifting portion 20 as an output rotary member, all of which are disposed in a transmission casing 12 (hereinafter briefly referred to as a “casing 12”) serving as a non-rotary member connectedly mounted on a vehicle body.

[0021] The shifting mechanism 10, preferably applicable to a vehicle of FR type (front-engine rear-drive type), is disposed between a longitudinally mounted engine 8, i.e., an internal combustion engine such as a gasoline engine or a diesel engine serving as a drive force directly connected to the input shaft 14 or indirectly connected thereto via the pulsation absorbing damper, and a pair of drive wheels 38 (Fig. 6). This allows a vehicle drive force to be transmitted to the pair of drive wheels 38 on left and right in sequence through a differential gear device 36 (final speed reduction gear) and a pair of drive axles. Further, in present embodiment, differential portion 11 corresponds to an electric differential portion of the present invention.

[0022] With the shifting mechanism 10 of the present embodiment, the engine 8 and the differential portion 11 are connected to each other in a direct connection. As used herein, the term "direct connection" may refer to a connection, established without intervening any fluid-type transmitting device such as a torque converter or a fluid coupling, which
involves a connection established with the use of the vibration damping device. Upper and lower halves of the shifting mechanism 10 are structured in symmetric relation with respect to an axis of the shifting mechanism 10 and, hence, the lower half is omitted in the skeleton view of Fig. 1.

[0023] The differential portion 11 includes a first electric motor M1, a power distributing mechanism 16 which is a mechanical mechanism to distribute the output of the engine 8 input to the input shaft 14 mechanically, and which distributes the output of the engine 8 to the first electric motor M1 and the power transmitting member 18, and a second electric motor M2 unitarily rotatable with the power transmitting member 18.

Further, the second electric motor M2 may be disposed at any portion of the power transmitting path extending from the power transmitting member 18 to the drive wheels 38. Moreover, the first and second electric motors M1 and M2 are so-called motor/generators each having a function even as an electric power generator. The first electric motor M1 has at least one function as an electric power generator that generates a reactive force, and the second electric motor M2 has at least a function as an electric motor serving as a drive force source to generate a drive force to run the vehicle.

[0024] The power distributing mechanism 16, corresponding to the differential mechanism of the present invention, mainly includes a differential-portion planetary gear unit 24 of a single pinion type having a given gear ratio ρ0 of, for instance, about "0.418", a switching clutch C0 and a switching brake B0. The differential-portion planetary gear unit 24 includes rotary elements, such as a differential-portion sun gear S0, a differential-portion planetary gears P0, a differential-portion carrier CA0 supporting the differential-portion planetary gears P0 to be rotatable about its axis and about the axis of the differential-portion sun gear S0, and a differential-portion ring gear R0 meshing with the differential-portion sun gear S0 through the differential-portion planetary gears P0. With the differential-portion sun gear S0 and the differential-portion ring gear R0 assigned to have the numbers of teeth represented by ZS0 and ZR0, respectively, the gear ratio ρ0 is expressed as ZS0/ZR0.

[0025] With the power distributing mechanism 16 of such a structure, the differential-portion carrier CA0 is connected to the input shaft 14, i.e., to the engine 8; the differential-portion sun gear S0 is connected to the first electric motor M1; and the differential-portion ring gear R0 is connected to the power transmitting member 18. The switching brake B0 is disposed between the differential-portion sun gear S0 and the
casing 12, and the switching clutch C0 is disposed between the differential-portion sun gear S0 and the differential-portion carrier CA0. With both the switching clutch C0 and the switching brake B0 being disengaged, the power distributing mechanism 16 is rendered operative such that the differential-portion sun gear S0, the differential-portion carrier CA0 and the differential-portion ring gear R0, forming the three elements of the differential-portion planetary gear unit 24, are caused to rotate relative to each other to enable the operation in a differential action, i.e., in a differential state under which the differential action is effectuated.

[0026] Thus, the output of the engine 8 is distributed to the first electric motor M1 and the power transmitting member 18 with a part of the engine output distributed to the first electric motor M1 being used to generate electric energy to be stored in a battery or to drivably rotate the second electric motor M2. This renders the differential portion 11 (power distributing mechanism 16) operative as an electrically controlled differential device. Thus, the differential portion 11 is placed in a so-called continuously variable shifting state (electrically controlled CVT state), in which a rotation speed of the power transmitting member 18 varies in a continuous fashion regardless of the engine 8 operating at a given rotation speed. That is, as the power distributing mechanism 16 is placed in the differential state, the differential portion 11 is also placed in differential state. In this casing, the differential portion 11 is placed in the continuously variable shifting state to operate as the electrically controlled continuously variable transmission with a speed ratio γ0 (a ratio of rotation speed N_{IN} of the driving device input shaft 14 to the rotation speed N_{18} of the power transmitting member 18) continuously varying in a value ranging from a minimum value γ0_{min} to a maximum value γ0_{max}. By controlling a drive state of the first electric motor M1 and the second electric motor M2 which are respectively connected to differential-portion sun gear S0 and to differential-portion ring gear R0, the differential states of each of rotary elements of the differential portion 11 are controlled.

[0027] Under such a state, as the switching clutch C0 or the switching brake B0 is engaged, the power distributing mechanism 16 is disenabled to perform the differential action, i.e., placed in a non-differential state in which no differential action is effectuated. In particular, as the switching clutch C0 which functions as a locking mechanism of the present invention is engaged to cause the differential-portion sun gear S0 and the differential-portion carrier CA0 to be unitarily coupled to each other, the
power distributing mechanism 16 is placed in a locked state under which the
differential-portion sun gear S0, the differential-portion carrier CA0 and the
differential-portion ring gear R0, serving as the three elements of the differential-portion
planetary gear unit 24, are caused to rotate together, i.e., in a unitarily rotating state
under the non-differential state in which no differential action is effectuated. Thus, the
differential portion 11 is placed in the non-differential state. Therefore, the rotation
speeds of the engine 8 and the power transmitting member 18 coincide with each other,
so that the differential portion 11 (power distributing mechanism 16) is placed in a fixed
shifting state, i.e., a step-variable shifting state to function as a transmission with the
speed ratio γ0 connected to a value of "1".

[0028] Instead of the switching clutch C0, next, if the switching brake B0 is engaged to
connect the differential-portion sun gear S0 to the casing 12, then, the power distributing
mechanism 16 is placed in the locked state. Thus, the differential-portion sun gear S0 is
placed in the non-rotating state under the non-differential state in which no differential
action is initiated, causing the differential portion 11 to be placed in the non-differential
state.

Since the differential-portion ring gear R0 rotates at a speed higher than that of
the differential-portion carrier CA0, the power distributing mechanism 16 functions as a
speed-increasing mechanism. Thus, the differential portion 11 (power distributing
mechanism 16) is placed in the fixed shifting state, i.e., the step-variable shifting state to
perform a function as a speed-increasing transmission with the speed ratio γ0 connected
to a value smaller than "1", i.e., for example, about 0.7.

[0029] With the present embodiment, the switching clutch C0 and the switching brake
B0 selectively place the shifting state of differential portion 11 (power distributing
mechanism 16) in the differential state, i.e., the unlocked state and the non-differential
state, i.e., the locked state. That is, the switching clutch C0 and the switching brake B0
serves as a differential state switching device that selectively switches the differential
portion 11 (power distributing mechanism 16) in one of: the continuously variable
shifting state, operative to perform the electrically and continuously controlled variable
shifting operation, under which the differential portion 11 (power distributing
mechanism 16) is placed in the differential state (coupled state) to perform the function
as the electrically controlled differential device operative to function as the continuously
variable transmission with, for instance, the shifting ratio is continuously variable; and
the fixed shifting state under which the differential portion 11 (power distributing mechanism 16) is placed in the shifting state, disenabling the function of the electrically controlled continuously variable shifting operation, such as the locked state disenabling the function of the continuously variable transmission in which no continuously variable shifting operation is effectuated with a speed ratio being locked at a connected level.

In the locked state, the differential portion 11 (power distributing mechanism 16) is rendered operative as a transmission of a single-stage or a multi-stage with a speed ratio of one kind or speed ratios of more than two kinds to function in the fixed shifting state (non-differential state), disenabling the electrically controlled continuously variable shifting operation, under which the differential portion 11 (power distributing mechanism 16) operates as the transmission of the single-stage or the multi-stage with the speed ratio kept at a connected level.

[0030] The automatic shifting portion 20 structures a part of a power transmitting path between the differential portion 11 and the drive wheels 38, and includes a first planetary gear unit 26 of a single-pinion type, a second planetary gear unit 28 of a single-pinion type and a third planetary gear unit 30 of a single-pinion type. The first planetary gear unit 26 includes a first sun gear S1, first planetary gears P1, a first carrier CA1 supporting the first planetary gears P1 to be rotatable about its axis and about the axis of the first sun gear S1, and a first differential-portion ring gear R1 meshing with the first sun gear S1 via the first planetary gears P1, having a gear ratio $\rho_1$ of, for instance, about "0.562". The second planetary gear unit 28 includes a second sun gear S2, second planetary gears P2, a second carrier CA2 supporting the second planetary gears P2 to be rotatable about its axis and about the axis of the second sun gear S2, and a second ring gear R2 meshing with the second sun gear S2 via the second planetary gears P2, having a gear ratio $\rho_2$ of, for instance, about "0.425".

[0031] The third planetary gear unit 30 includes a third sun gear S3, third planetary gears P3, a third carrier CA3 supporting the third planetary gears P3 to be rotatable about its axis and about the axis of the third sun gear S3, and the third ring gear R3 meshing with the third sun gear S3 through the third planetary gears P3, having a gear ratio $\rho_3$ of, for instance, about "0.421". With the first sun gear S1, the first ring gear R1, the second sun gear S2, the second ring gear R2, the third sun gear S3 and the third ring gear R3 assigned to have the numbers of teeth represented by $ZS_1$, $ZR_1$, $ZS_2$, $ZR_2$, $ZS_3$
and ZR3, respectively, the gear ratios ρ1, ρ2 and ρ3 are represented by ZS1/ZR1, ZS2/ZR2, and ZS3/ZR3, respectively.

[0032] With the automatic shifting portion 20, the first sun gear S1 and the second sun gear S2 are integrally connected to each other and selectively connected to the power transmitting member 18 through a second clutch C2 while selectively connected to the casing 12 through a first brake B1. The first carrier CA1 is selectively connected to the casing 12 through a second brake B2 and the third ring gear R3 is selectively connected to the casing 12 through a third brake B3. The first ring gear R1, the second carrier CA2 and the third carrier CA3 are integrally connected to each other and also connected to the output shaft 22. The second ring gear R2 and the third sun gear S3 are integrally connected to each other and selectively connected to the power transmitting member 18 through the first clutch C1.

[0033] Thus, the automatic shifting portion 20 and the power transmitting member 18 are selectively connected to each other through the first clutch C1 or the second clutch C2 used for establishing a gear shift position in the automatic shifting portion 20. In other words, the first clutch C1 and the second clutch C2 collectively function as an engaging device for switching the operations of the power transmitting member 18 and the automatic shifting portion 20. That is, such an engaging device selectively switches a power transmitting path between the differential portion 11 (transmitting member 18) and the drive wheels 38 in a power transmitting state, enabling a power transfer through the power transmission path, and a power interrupting state (neutral state) to interrupting the power transfer through the power transmission path. That is, with at least one of the first clutch C1 and the second clutch C2 being engaged, the power transmitting path is placed in the power transmitting state. In contrast, with both the first clutch C1 and the second clutch C2 being disengaged, the power transmitting path is placed in the power interrupting state (neutral state).

[0034] The switching clutch C0, the first clutch C1, the second clutch C2, the switching brake B0, the first brake B1, the second brake B2 and the third brake B3 are hydraulic-type frictionally coupling devices used in a vehicular step-variable type automatic transmission of the related art. An example of the frictionally coupling device includes a wet-type multiple-disc type that includes a plurality of superposed friction plates pressed against each other with a hydraulic actuator or a band brake comprised of a rotary drum having an outer circumferential surface on which one band or two bands
are wound to be tightened at one ends with a hydraulic actuator to allow associated component parts, between which the rotary drum intervenes, to be selectively connected to each other.

[0035] With the shifting mechanism 10 of such a structure, as indicated in an engagement operation table shown Fig. 2, the switching clutch C0, the first clutch C1, the second clutch C2, the switching brake B0, the first brake B1, the second brake B2 and the third brake B3 are selectively engaged in operation. This selectively establishes either one of a 1st-speed gear position (1st-speed gear shift position) to a 5th-speed gear position (5th-speed gear shift position) or one of a reverse-drive gear position (reverse-drive gear shift position) and a neural position with a speed ratios $\gamma$ (input-shaft rotation speed $N_{IN}$/output-shaft rotation speed $N_{OUT}$) varying in nearly equal ratio for each gear position.

[0036] In particular, with the present embodiment, the power distributing mechanism 16 is comprised of the switching clutch C0 and the switching brake B0, either one of which is engaged in operation. This makes it possible to cause the differential portion 11 to be placed in the continuously variable shifting state enabling the operation as the continuously variable transmission while establishing the fixed shifting state enabling the transmission to operate with the speed ratio maintained at a fixed level. With either one of the switching clutch C0 and the switching brake B0 being engaged in operation, accordingly, the differential portion 11 is placed in the fixed shifting state to cooperate with the automatic shifting portion 20 to allow the shifting mechanism 10 to operate as the step-variable transmission placed in the step-variable shifting state. With both of the switching clutch C0 and the switching brake B0 being disengaged in operation, the differential portion 11 is placed in the continuously variable shifting state to cooperate with the automatic shifting portion 20 to allow the shifting mechanism 10 to operate as the electrically controlled continuously variable transmission placed in the continuously variable shifting state.

In other words, the shifting mechanism 10 is switched to the step-variable shifting state, upon engagement of either one of the switching clutch C0 and the switching brake B0, and the continuously variable shifting state with both of the switching clutch C0 and the switching brake B0 being brought into disengagement. In addition, it can be said that the differential portion 11 is the transmission that can also be switched to the step-variable shifting state and the continuously variable shifting state.
[0037] For example, as shown in Fig. 2, under a circumstance where the shifting mechanism 10 is caused to function as the step-variable transmission, engaging the switching clutch C0, the first clutch C1 and the third brake B3 results in the 1st-speed gear position with the speed ratio \( \gamma_1 \) having a maximum value of, for instance, about "3.357". Engaging the switching clutch C0, the first clutch C1 and the second brake B2 results in the 2nd-speed gear position with the speed ratio \( \gamma_2 \) of, for instance, about "2.180", which is lower than that of the 1st-speed gear position.

Engaging the switching clutch C0, the first clutch C1 and the first brake B1 results in the 3rd-speed gear position with the speed ratio \( \gamma_3 \) of, for instance, about "1.424", which is lower than that of the 2nd-speed gear position. Engaging the switching clutch C0, the first clutch C1 and the second clutch C2 results in the 4th-speed gear position with the speed ratio \( \gamma_4 \) of, for instance, about "1.000", which is lower than that of the 3rd-speed gear position.

[0038] With the first clutch C1, the second clutch C2 and the switching brake B0 being engaged, the 5th-speed gear position is established with the speed ratio \( \gamma_5 \) of, for example, about "0.705", which is smaller than that of the 4th-speed gear position. With the second clutch C2 and the third brake B3 being engaged, further, the reverse-drive gear position is established with the speed ratio \( \gamma_R \) of, for example, about "3.209", which lies at a value between those of the 1st-speed and 2nd-speed gear positions. For the neutral "N" state to be established, for instance, all the clutches and the brakes C0, C1, C2, B0, B1, B2 and B3 are disengaged.

[0039] However, for the shifting mechanism 10 to function as the continuously variable transmission, both the switching clutch C0 and the switching brake B0 are disengaged as indicated in the engagement operation table shown in Fig. 2. With such operation, the differential portion 11 is rendered operative to function as the continuously variable transmission and the automatic shifting portion 20, connected thereto in series, is rendered operative to function as the step-variable transmission. This causes the rotation speed input to the automatic shifting portion 20, i.e., the rotation speed of the power transmitting member 18 to be continuously varied for each of the 1st-speed gear position, the 2nd-speed gear position, the 3rd-speed gear position and the 4th-speed gear position. This allows each of the various gear positions to be established in an infinitely variable shifting ratio. Accordingly, a speed ratio can be continuously variable across the
adjacent gear positions, making it possible for the shifting mechanism 10 as a whole to obtain an infinitely variable total speed ratio (overall speed ratio) γT.

[0040] Fig. 3 shows a collinear chart plotted in straight lines that can represent a correlation among the rotation speeds of the various rotary elements available to accomplish clutch engagement states in different modes depending on the gear positions of the shifting mechanism 10 comprised of the differential portion 11, functioning as the continuously variable shifting portion or the first shifting portion, and the automatic shifting portion 20 functioning as the step-variable shifting portion or the second shifting portion. The collinear chart of Fig. 3 is a two-dimensional coordinate system having the horizontal axis, representing the correlation among the gear ratios ρ established with the planetary gear units 24, 26, 28 and 30, and the vertical axis representing relative rotation speeds of the rotary elements. The lowermost line X1 of three horizontal lines indicates the rotation speed laying at a value of "0". An upper horizontal line X2 indicates the rotation speed laying at a value of "1.0", that is, a rotation speed NE of the engine 8 connected to the input shaft 14. The uppermost horizontal line XG indicates the rotation speed of the power transmitting member 18.

[0041] Starting from the left, three vertical lines Y1, Y2 and Y3, corresponding to the three elements of the power distributing mechanism 16 forming the differential portion 11, respectively, represent relative rotation speeds of the differential-portion sun gear S0 corresponding to a second rotary element (second element) RE2, the differential-portion carrier CA0 corresponding to a first rotary element (first element) RE1, and the differential-portion ring gear R0 corresponding to a third rotary element (third element) RE3. A distance between adjacent ones of the vertical lines Y1, Y2 and Y3 is determined in accordance with the gear ratio ρ0 of the differential-portion planetary gear unit 24.

Starting from the left, five vertical lines Y4, Y5, Y6, Y7 and Y8 for the automatic shifting portion 20 represent relative rotation speeds of the first and second sun gears S1 and S2 corresponding to a fourth rotary element (fourth element) RE4 and connected to each other, the first carrier CA1 corresponding to a fifth rotary element (fifth element) RE5, the third ring gear R3 corresponding to a sixth rotary element (sixth element) RE6, the first ring gear R1 and the second and third carriers CA2 and CA3 corresponding to a seventh rotary element (seventh element) RE7 and connected to each other, and the second ring gear R2 and the third sun gear S3 corresponding to an eighth rotary element.
(eighth element) RE8 and connected to each other, respectively. A distance between the adjacent ones of the vertical lines Y4 to Y8 is determined based on the gear ratios ρ1, ρ2 and ρ3 of the first, second and third planetary gear units 26, 28 and 30.

[0042] In the correlation between the vertical lines on the collinear chart, if an interval between the sun gear and the carrier is assigned to a distance corresponding to a value of "1", an interval between the carrier and the ring gear is assigned to a distance corresponding to the gear ratio ρ of the planetary gear unit. That is, for the differential portion 11, an interval between the vertical lines Y1 and Y2 is assigned to a distance corresponding to a value of "1" and an interval between the vertical lines Y2 and Y3 is assigned to a distance corresponding to a value of "ρ0". For each of the first, second and third planetary gear units 26, 28 and 30 of the automatic shifting portion 20, further, an interval between the sun gear and the carrier is assigned to a distance corresponding to a value of "1" and an interval between the carrier and the ring gear is assigned to a distance corresponding to the gear ratio "ρ".

[0043] Expressing the structure using the collinear chart shown in Fig. 3, the shifting mechanism 10 of the present embodiment takes the form of a structure including the power distributing mechanism 16 (differential portion 11). With the power distributing mechanism 16, the differential-portion planetary gear unit 24 has the first rotary element RE1 (differential-portion carrier CA0) connected to the input shaft 14, i.e., the engine 8, while selectively connected to the second rotary element RE2 (differential-portion sun gear S0) through the switching clutch C0, the second rotary element RE2 connected to the first electric motor M1 while selectively connected to the casing 12 through the switching brake B0, and the third rotary element RE3 (differential-portion ring gear R0) connected to the power transmitting member 18 and the second electric motor M2. Thus, the rotation of the input shaft 14 is transmitted (input) to the automatic shifting portion (step-variable shifting portion) 20 through the power transmitting member 18. An inclined straight line L0, passing across an intersecting point between the lines Y2 and X2, represents the correlation between the rotation speeds of the differential-portion sun gear S0 and the differential-portion ring gear R0.

[0044] For example, as the switching clutch C0 and the switching brake B0 are disengaged, the shifting mechanism 10 is switched to the continuously variable shifting state (differential state). In this case, controlling the rotation speed of the first electric motor M1 causes the rotation speed of the differential-portion sun gear S0, represented
by an intersecting point between the straight line L0 and the vertical line Y1, to increase or decrease. Under such a state, if the rotation speed of the differential-portion ring gear R0, bound with the vehicle speed V, remains at a nearly fixed level, then, the rotation speed of the differential-portion carrier CA0, represented by the intersecting point between the straight line L0 and the vertical line Y2, is caused to increase or decrease.

[0045] With the switching clutch C0 being engaged to couple the differential-portion sun gear S0 and the differential-portion carrier CA0 to each other, the power distributing mechanism 16 is brought into the non-differential state where the three rotary elements are caused to integrally rotate as a unitary unit. Thus, the straight line L0 matches the lateral line X2, so that the power transmitting member 18 is caused to rotate at the same rotation speed as the engine rotation speed NE. In contrast, with the switching brake B0 being engaged to halt the rotation of the differential-portion sun gear S0, the power distributing mechanism 16 is brought into the non-differential state to function as the speed-increasing mechanism. Thus, the straight line L0 describes a state as shown in Fig. 3, under which the rotation of the differential-portion ring gear R0, i.e., the power transmitting member 18, represented by an intersecting point between the straight line L0 and the vertical line Y3, is input to the automatic shifting portion 20 at a rotation speed higher than the engine rotation speed NE.

[0046] With the automatic shifting portion 20, the fourth rotary element RE4 is selectively connected to the power transmitting member 18 through the second clutch C2 and selectively connected to the casing 12 through the first brake B1. The fifth rotary element RE5 is selectively connected to the casing 12 through the second brake B2 and the sixth rotary element RE6 is selectively connected to the casing 12 through the third brake B3. The seventh rotary element RE7 is connected to the output shaft 22 and the eighth rotary element RE8 is selectively connected to the power transmitting member 18 through the first clutch C1.

[0047] As shown in Fig.3, with the automatic shifting portion 20, upon engagement of the first clutch C1 and the third brake B3, the rotation speed of the output shaft 22 for the 1st-speed gear position is represented by an intersecting point between the inclined straight line L1 and the vertical line Y7 representing the rotation speed of the seventh rotary element RE7 connected to the output shaft 22. Here, the inclined straight line L1 passes across an intersecting point between the vertical line Y8, indicative of the rotation speed of the eighth rotary element RE8, and the horizontal line X2, and an
intersecting point between the vertical line Y6, indicative of the rotation speed of the sixth rotary element RE6, and the horizontal line X1.

[0048] Similarly, the rotation speed of the output shaft 22 for the 2nd-speed gear position is represented by an intersecting point between an inclined straight line L2, determined upon engagement of the first clutch C1 and the second brake B2, and the vertical line Y7 indicative of the rotation speed of the seventh rotary element RE7 connected to the output shaft 22. The rotation speed of the output shaft 22 for the 3rd-speed gear position is represented by an intersecting point between an inclined straight line L3, determined upon engagement of the first clutch C1 and the first brake B1, and the vertical line Y7 indicative of the rotation speed of the seventh rotary element RE7 connected to the output shaft 22. The rotation speed of the output shaft 22 for the 4th-speed gear position is represented by an intersecting point between a horizontal line L4, determined upon engagement of the first and second clutches C1 and C2, and the vertical line Y7 indicative of the rotation speed of the seventh rotary element RE7 connected to the output shaft 22.

[0049] For the 1st-speed to 4th-speed gear positions, the switching clutch C0 remains engaged. Therefore, a drive force is applied from the differential portion 11, i.e., the power distributing mechanism 16 to the eighth rotary element RE8 at the same rotation speed as that of the engine rotation speed NE. However, in place of the switching clutch C0, if the switching clutch B0 is engaged, then, the drive force is applied from the differential portion 11 to the eighth rotary element RE8 at a higher rotation speed than the engine rotation speed NE. Thus, an intersecting point between a horizontal line L5 and the vertical line Y7 represents the rotation speed of the output shaft 22 for the 5th-speed gear position. Here, the horizontal line L5 is determined upon engagement of the first clutch C1, the second clutch C2 and the switching brake B0 and the vertical line Y7 represents the rotation speed of the seventh rotary element RE7 connected to the output shaft 22.

[0050] Fig. 4 exemplarily shows various input signals applied to an electronic control device 40, serving as a control device for controlling the shifting mechanism 10 forming part of the hybrid vehicle drive apparatus according to the present invention, and various output signals delivered from the electronic control device 40. The electronic control device 40 includes a so-called microcomputer incorporating a CPU, a ROM, a RAM and an input/output interface. With the microcomputer operated to perform signal processing
according to programs preliminarily stored in the ROM while utilizing a temporary data storage function of the RAM, hybrid drive controls are conducted to control the engine 8 and the first and second electric motors M1 and M2, while executing drive controls such as shifting controls of the automatic shifting portion 20.

[0051] The electronic control device 40 is applied with the various input signals from various sensors and switches shown in Fig. 4. These input signals include a signal indicative of an engine cooling water temperature TEMPW, a signal indicative of a selected shift position SP, a signal indicative of a rotation speed N_{M1} of the first electric motor M1, a signal indicative of a rotation speed N_{M2} of the second electric motor M2, a signal indicative of the engine rotation speed NE representing the rotation speed of the engine 8, a signal indicative of a set value of gear ratio row, a signal commanding an M-mode (manually shift drive mode), and an air-conditioning signal indicative of the operation of an air conditioner, etc.

[0052] Besides the input signals described above, the electronic control device 40 is further applied with other various input signals. These input signals include a signal indicative of the vehicle speed V corresponding to the rotation speed N_{OUT} of the output shaft 22, a working oil temperature signal indicative of a working oil temperature of the automatic shifting portion 20, a signal indicative of a side brake being operated, a signal indicative of a foot brake being operated, a catalyst temperature signal indicative of a catalyst temperature, an accelerator opening signal indicative of a displacement value ACC of an accelerator pedal corresponding to an output demand value required by a driver, a cam angle signal, a snow mode setting signal indicative of a snow mode being set, an acceleration signal indicative of a fore and aft acceleration of the vehicle, an auto-cruising signal indicative of the vehicle running under an auto-cruising mode, a vehicle weight signal indicative of a weight of the vehicle, a drive wheel velocity signal indicative of a wheel velocity of each drive wheel, a signal indicative of an air-fuel ratio A/F of the engine 8, and a signal indicative of a throttle valve opening \theta_{TH}, etc.

[0053] The electronic control device 40 generates various control signals to be applied to an engine output control device 43 (refer to Fig. 6) for controlling the engine output. These control signals include, for instance, a drive signal applied to a throttle actuator 97 for controlling an opening degree \theta_{TH} of a throttle valve 96 disposed in an intake manifold 95 of the engine 8, a fuel supply quantity signal to be applied to a fuel injection device 98 for controlling the amount to fuel to be supplied to each cylinder of
the engine 8, an ignition signal to be applied to an ignition device 99 for commanding an
ignition timing of the engine 8, a supercharger pressure regulating signal for adjusting a
supercharger pressure level, an electric air-conditioner drive signal for actuating an
electric air conditioner, and command signals for commanding the operations of the first
and second electric motors M1 and M2.

[0054] Besides the control signals described above, the electronic control device 40
generates various output signals. These output signals include a shift-position (selected
operating position) display signal for activating a shift indicator, a gear-ratio display
signal for providing a display of the gear ratio, a snow-mode display signal for providing
a display of a snow mode under operation, an ABS actuation signal for actuating an ABS
actuator for preventing slippages of the drive wheels during a braking effect, an M-mode
display signal for displaying the M-mode being selected, valve command signals for
actuating electromagnet valves incorporated in a hydraulically operated control circuit
42 (see Fig. 6) to control the hydraulic actuators of the hydraulically operated frictional
engaging devices of the differential portion 11 and the automatic shifting portion 20,
drive command signals for actuating an electric i.e. electrically-controlled hydraulic
pump 72 serving as a hydraulic pressure source of the hydraulically operated control
circuit 42, a signal for driving an electric heater, and signals applied to a cruise-control
computer, etc.

[0055] Fig. 5 is a view showing one sample of a shift operating device 48, serving as a
switching device, which is manually operated to select one of the shift positions SP of
multiple kinds. The shift operating device 48 includes a shift lever 49 mounted aside, for
example, a driver’s seat to be manually operated to select one of the shifting positions
SP of the plural kinds.

[0056] The shift lever 49 has a structure arranged to be selectively shifted in manual
operation to be set to one of a parking position "P" (Parking) under which the shifting
mechanism 10, i.e., the automatic shifting portion 20, is placed in the neutral state
interrupting the power transmitting path of the shifting mechanism 10, i.e., the automatic
shifting portion 20, a reverse drive running position "R" (Reverse) for the vehicle to run
in a reverse drive mode, a neutral position "N" (Neutral) for the neutral state to be
established under which the power transmitting path of the shifting mechanism 10 is
interrupted, a forward drive automatic shift position "D" (Drive) for an automatic shift
control to be executed within a varying range of the total speed ratio □T that can be
shifted with the shifting mechanism 10, and a forward drive manual shift position "M" (Manual) under which a manual shift running mode (manual mode) is established to set a so-called shift range that limits the shift gear positions in a high speed range during the execution of the automatic shift control.

[0057] In conjunction with the shift lever 49 being manually operated to each of the shift positions SP, for instance, the hydraulic control circuit 42 is electrically switched in such a way to establish each of the gear shift positions such as the reverse drive position "R", the neutral position "N" and the forward drive position "D" as shown in the engagement operation table shown in Fig. 2.

[0058] Among the various shift positions SP covering "P" to "M" positions, the "P" and "N" positions represent the non-running positions selected when no intention is present to run the vehicle. For the "P" and "N" positions to be selected, both the first and second clutches C1 and C2 are disengaged, as shown in, for example, the engagement operation table of Fig. 2, and non-drive positions are selected to place the power transmitting path in the power cutoff state. This causes the power transmitting path of the automatic shifting portion 20 to be interrupted, disenabling the vehicle to be driven.

[0059] The "R", "D" and "M" positions represent running positions selected when the vehicle is caused to run. These shift positions also represent drive positions selected when switching the power transmitting path to the power transmitting state under which at least one of the first and second clutches C1 and C2 is engaged as shown in, for instance, the engagement operation table of Fig. 2. With such shifting positions are selected, the power transmitting path of the automatic shifting portion 20 is connected to enable the vehicle to be driven.

[0060] More particularly, with the shift lever 49 manually operated from the "P" position or the "N" position to the "R" position, the second clutch C2 is engaged so that the power transmitting path of the automatic shifting portion 20 is switched from the power cutoff state (neutral state) to the power transmitting state. With the shift lever 49 manually operated from the "N" position to the "D" position, at least the first clutch C1 is engaged, causing the power transmitting path of the automatic shifting portion 20 to be switched from the power cutoff state to the power transmitting state.

With the shift lever 49 manually operated from the "R" position to the "P" position or the "N" position, the second clutch C2 is disengaged, causing the power transmitting path of the automatic shifting portion 20 to be switched from the power
transmitting state to the power cutoff state. With the shift lever 49 manually operated from the "D" position to the "N" position, the first and second clutches C1 and C2 are disengaged, causing the power transmitting path of the automatic shifting portion 20 to be switched from the power transmitting state to the power cutoff state.

[0061] The "M" position is located at the same position as the "D" position in the longitudinal direction of the vehicle, and is adjacent thereto in the lateral direction of the same. The shift lever 49 is operated to the "M" position, for manually selecting one of the above-indicated "D" through "L" positions. Specifically, for the "M" position, an upshift position "+ " and a downshift position " - " are provided in the front-rear direction of the vehicle. The shift lever 49 is manipulated to the upshift position "+ " and the downshift position " - " to select any of the "D" range to the "L" range. For example, the five shifting ranges of the "D" range to the "L" range selected at the "M" position correspond to, in the changeable range of the overall speed ratio $\gamma T$ which can control the shifting mechanism 10 automatically, different kinds of shifting ranges in which the overall speed ratio $\gamma T$ at higher speed side (minimum gear ratio side) are different. Also, these five shifting ranges limit the shifting range i.e., scope of the shifting position (gear position) so that the maximum side shifting position which can control the shifting of the automatic shifting portion 20 is different.

[0062] The shift lever 49 is urged by urge means such as a spring from the upshift position "+ " and the downshift position " - " to be automatically returned to the "M" position. In addition, the shift operation device 48 is provided with a shift position sensor (not shown) for detecting each of the shift positions of the shift lever 49, to output a signal representing the shift position of the shift lever 48, and the number of manipulation at the "M" position to the electronic control device 40.

[0063] When the "M" position is selected by manipulation of the shift lever 49, the automatic shift control is executed within the total speed ratio $\gamma T$ in which the shifting mechanism 10 can be shifted in each of the shifting ranges thereof, so as not to exceed the highest speed side shifting position or the shifting ratio of the shifting range. For example, in the step variable shifting running in which the shifting mechanism 10 is switched to the step variable shifting state, the automatic shift control is executed within the total speed ratio $\gamma T$ in which the shifting mechanism 10 can be shifted in each of the shifting ranges thereof.
In the continuously variable shifting running in which the shifting mechanism 10 is switched to the continuously variable shifting state, the automatic shift control is executed within the total speed ratio $\gamma T$ in which the shifting mechanism 10 can be shifted in each of the shifting ranges thereof, and which is obtained by continuously variable shift width i.e. spread of the power distributing mechanism 16, and each of the gear positions of the automatic shifting portion 20 to be automatically controlled corresponding to each of the shifting ranges within the changeable shifting positions. This "M" position corresponds to a shift position for selecting a manually shifting running mode (manual mode) i.e. a control style in which the shifting mechanism 10 is subjected to the manual shifting control.

[0064] Fig. 6 is a functional block diagram illustrating an essential part of a control function to be performed with the electronic control device 40. In Fig. 6, step-variable shifting control means 54 functions as shifting control means for the shifting the automatic shifting portion 20. For instance, the step-variable shifting control means 54 discriminates whether to execute the shifting in the automatic shifting portion 20 on the basis of a vehicle condition represented by the vehicle speed $V$ and the demanded output torque $T_{OUT}$ for the automatic shifting portion 20 by referring to the relationships (including the shifting diagram and the shifting map), preliminarily stored in memory means 56, which are plotted in solid lines and single dot lines as shown in Fig. 7. That is, the step-variable shifting control means 54 discriminates a shifting position to be shifted in the automatic shifting portion 20, thereby causing the automatic shifting portion 20 to execute the shifting so as to obtain the discriminated shifting position. When this takes place, the step-variable shifting control means 54 outputs a command (shifting output command) to the hydraulic control circuit 42 for engaging and/or disengaging the hydraulic-type frictionally coupling devices, excepting the switching clutch C0 and the switching brake B0, so as to achieve a desired shifting position in accordance with, for instance, the engagement operation table shown in Fig. 2.

[0065] Hybrid control means 52 renders the engine 8 operative in an operating region at high efficiency under the infinitely variable shifting state of the shifting mechanism 10, i.e., the differential state of the differential portion 11. At the same time, the hybrid control means 52 causes the engine 8 and the second electric motor M2 to deliver drive forces at varying distributing rates while causing the first electric motor M1 to generate electric power at a varying rate for a reactive force to be generated at an optimum value,
thereby controlling the speed ratio $\gamma_0$ of the differential portion 11 placed in the electrically controlled continuously variable transmission.

For instance, during the running of the vehicle at a current vehicle speed, the hybrid control means 52 calculates a target (demanded) output of the vehicle by referring to the displacement value Acc of the accelerator pedal and the vehicle speed $V$ that collectively represents the output demanded value intended by the driver. Then, the hybrid control means 52 calculates a demanded total target output based on the target output and a charging request value of the vehicle. In order to obtain the total target output, the hybrid control means 52 calculates a target engine output with taking account of the transmitting a loss, loads on auxiliary units and assisting torque of the second electric motor M2, etc. Then, the hybrid control means 52 controls the engine 8 so as to provide the engine rotation speed $N_E$ and engine torque $T_E$ such that the target engine output is obtained, while controlling the first electric motor M1 to generate electric power at a proper power rate.

[0066] The hybrid control means 52 executes a hybrid control with taking account of the gear position of the automatic shifting portion 20 so as to obtain power performance and improved fuel consumption. During such a hybrid control, the differential portion 11 is rendered operative to function as the electrically controlled continuously variable transmission for the purpose of matching the engine rotation speed $N_E$, determined for the engine 8 to operate at a high efficiency, to the rotation speed of the power transmitting member 18 determined based on the vehicle speed $V$ and the selected gear position of the automatic shifting portion 20.

[0067] To this end, the hybrid control means 52 preliminarily stores therein an optimum fuel economy curve (including a fuel economy map and relevant relationship) of the engine 8 preliminarily determined on an experimental basis such that, during the running of the vehicle under the continuously variable shifting state, the vehicle has drivability and fuel economy performance in compatibility on a two-dimensional coordinate with parameters including, for instance, the engine rotation speed $N_E$ and output torque (engine torque) $T_E$ of the engine 8. In order to cause the engine 8 to operate on such an optimum fuel economy curve, a target value on the total speed ratio $\gamma_T$ of the shifting mechanism 10 is determined so as to obtain engine torque $T_E$ and the engine rotation speed $N_E$ for the demanded engine output to be generated so as to satisfy, for instance, the target output (total target output and demanded drive force). To achieve
such a target value, the hybrid control means 52 controls the speed ratio $\gamma_0$ of the
differential portion 11, while controlling the total speed ratio $\gamma_T$ within a variable
shifting range at a value, for instance, ranging from 1.3 to 0.5.

[0068] During such hybrid control, the hybrid control means 52 allows electric energy,
generated by the first electric motor M1, to be supplied to a battery 60 and the second
electric motor M2 through an inverter 58. This allows a major part of the drive force,
delivered from the engine 8, to be mechanically transmitted to the power transmitting
member 18 and the rest of the drive force of the engine 8 is delivered to the first electric
motor M1 to be consumed thereby for conversion to electric power. The resulting
electric energy is supplied through the inverter 58 to the second electric motor M2,
which in turn is driven to provide a drive force for delivery to the power transmitting
member 18. Equipments, involved in the operation of generating electric energy and the
operation causing the second electric motor M2 to consume electric energy, establish an
electric path in which the part of the drive force, delivered from the engine 8, is
converted to electric energy which in turn is converted into mechanical energy.

[0069] The hybrid control means 52 functionally includes engine output control means
for executing an output control of the engine 8 so as to provide the demanded engine
output. The engine output control means allows the throttle actuator 97 to perform a
throttle control so as to controllably open or close the electronic throttle valve 96. In
addition, the engine output control means outputs commands to the engine output
control device 43 so as to cause the fuel injection device 98 to control the fuel injection
quantity and fuel injection timing for performing a fuel injection control while
permitting the ignition device 99, such as an igniter or the like, to control an ignition
timing for an ignition timing control. These commands are output in a single mode or a
combined mode. For instance, the hybrid control means 52 drives the throttle actuator 97
in response to the acceleration opening signal Acc by fundamentally referring to the
preliminarily stored relationship, not shown, so as to execute the throttle control such
that the greater the accelerator opening Acc, the greater will be the throttle valve
opening $\theta_{TH}$.

[0070] A solid line A, shown in Fig. 7, represents a boundary line between an engine
drive region and a motor drive region for the engine 8 and an electric motor, i.e., for
instance, the second electric motor M2 to be selectively switched as a drive force source
for the vehicle to perform a startup/running (hereinafter referred to as "running"). In
other words, the boundary line is used for switching a so-called engine drive mode, in which the engine 8 is caused to act as a running drive force source for starting up/running (hereinafter referred to as "running") the vehicle, and a so-called motor drive mode in which the second electric motor M2 is caused to act as a drive force source for running the vehicle.

The preliminarily stored relationship, having the boundary line (in the solid line A) shown in Fig. 7 for the engine drive region and the motor drive region to be switched, represents one example of a drive-force source switching diagram (drive force source map), formed on a two-dimensional coordinate, which includes parameters such as the vehicle speed V and output torque $T_{OUT}$ representing a drive force correlation value. Memory means 56 preliminarily stores such a drive-force source switching diagram together with the shifting diagram (shifting map) designated by, for instance, the solid line and the single dot line in Fig. 7.

[0071] The hybrid control means 52 determines which of the motor drive region and the engine drive region is to be selected based on the vehicle condition, represented by the vehicle speed V and demanded torque output $T_{OUT}$ by referring to, for instance, the drive-force source switching diagram shown in Fig. 7, thereby executing the motor drive mode or the engine drive mode. Thus, the hybrid control means 52 executes the motor drive mode at relatively low output torque $T_{OUT}$, i.e., low engine torque TE, at which an engine efficiency is generally regarded to be lower than that involved a high torque region, or a relatively low vehicle speed range of the vehicle speed V, i.e., under a low load region as will be apparent from Fig. 7.

[0072] During such a motor drive mode, the hybrid control means 52 renders the differential portion 11 operative to perform an electrical CVT function (differential function) for controlling the first-motor rotation speed $N_{M1}$ at a negative rotation speed, i.e., at an idling speed to maintain the engine rotation speed NE at a zeroed or nearly zeroed level, thereby minimizing a drag of the engine 8, remained under a halted state, for providing improved fuel economy.

[0073] Further, even under the engine drive region, the hybrid control means 52 may execute the operation to allow the second electric motor M2 to be supplied with electric energy, generated by the first electric motor M1, and/or electric energy delivered from the battery 60 via the electric path mentioned above. This causes the second electric motor M2 to be driven for performing a torque assisting operation to assist the drive
force of the engine 8. Thus, for the illustrated embodiment, the term "engine drive mode" may refer to an operation covering the engine drive mode and the motor drive mode in combination.

[0074] Further, the hybrid control means 52 can cause the differential portion 11 to perform the electrical CVT function through which the engine 8 can be maintained under the operating state regardless of the vehicle left in a halted condition or a low speed condition. For instance, if a drop occurs in a state of charge SOC of the battery 60 during the halt of the vehicle with a need occurring on the first electric motor M1 to generate electric power, the drive force of the engine 8 drives the first electric motor M1 to generate electric power with an increase in the rotation speed of the first electric motor M1. Thus, even if the second-motor rotation speed N_{M2}, uniquely determined with the vehicle speed V, is zeroed (nearly zeroed) due to the halted condition of the vehicle, the power distributing mechanism 16 performs the differential action, causing the engine rotation speed NE to be maintained at a level beyond an autonomous rotation speed.

[0075] The hybrid control means 52 executes the operation to cause the differential portion 11 to perform the electrical CVT function for controlling the first-motor M1 rotation speed N_{M1} and the second-motor M2 rotation speed N_{M2} to maintain the engine rotation speed NE at an arbitrary level regardless of the vehicle remaining under the halted or running state. As will be understood from the collinear chart shown in Fig. 3, for instance, when raising the engine rotation speed NE, the hybrid control means 52 executes the operation to maintain the second-motor M2 rotation speed N_{M2}, bound with the vehicle speed V, at a nearly fixed level while raising the first-motor M1 rotation speed N_{M1}.

[0076] In placing the shifting mechanism 10 in the step-variable shifting state, increasing-speed gear-position determining means 62 determines which of the switching clutch C0 and the switching brake B0 is to be engaged. To this end, the increasing-speed gear-position determining means 62 executes the operation based on, for instance, the vehicle condition according to the shifting diagram, shown in Fig. 7, which is preliminarily stored in the memory means 56, to determine whether or not a gear position to be shifted in the shifting mechanism 10 is an increasing-speed gear position, i.e., for instance, a 5th-speed gear position.

[0077] The switching control means 50 switches the engaging and/disengaging states of the differential state switching device (switching clutch C0 and switching brake B0)
based on the vehicle condition, thereby selectively executing a switchover between the continuously variable shifting state and the step-variable shifting state, i.e., between the differential state and the locked state. For instance, the switching control means 50 executes the operation based on the vehicle condition, represented with the vehicle speed \( V \) and demanded output torque \( T_{\text{OUT}} \), by referring to the relationships (shifting diagram and shifting map), preliminarily stored in the memory means 56, which are shown in the broken line and the double dot line in Fig. 7, thereby determining whether to switch the shifting state of the shifting mechanism 10 (differential portion 11). That is, the operation is executed to determine whether there exist a continuously variable shifting control region for the shifting mechanism 10 to be placed in the continuously variable shifting state or a step-variable shifting control region for the shifting mechanism 10 to be placed in the step-variable shifting state. This allows the operation to be executed for determining the shifting state to be switched in the shifting mechanism 10, thereby executing the operation to selectively switch the shifting state to one of the continuously variable shifting state and the step-variable shifting state.

[0078] More particularly, if the determination is made that the shifting mechanism 10 lies in the step-variable shifting control region, then, the switching control means 50 outputs a signal to the hybrid control means 52 for disenabling or interrupting the hybrid control or the continuously variable shifting control, while permitting the step-variable shifting control means 54 to perform the shifting for the step-variable shifting operation that has been preliminarily determined. When this takes place, the step-variable shifting control means 54 allows the automatic shifting portion 20 to perform the automatic shifting in accordance with, for instance, the shifting diagram shown in Fig. 7 and preliminarily stored in the memory means 56.

For instance, the engagement operation table, shown in Fig. 2 and preliminarily stored in the memory means 56, represents the operations in combination of the hydraulically operated frictional engaging devices, that is, the clutches C0, C1 and C2 and the brakes B0, B1, B2 and B3 to be selected in such a shifting operation. That is, a whole of the shifting mechanism 10, i.e., the differential portion 11 and the automatic shifting portion 20, functions as a so-called step-variable automatic transmission, thereby establishing the gear positions according to the engagement operation table shown in Fig. 2.
[0079] For instance, if the increasing-speed gear-position determining means 62 determines that the 5th-gear position is to be selected, the shifting mechanism 10 as a whole can obtain a so-called overdrive-gear position on an increasing-speed gear position with a speed ratio less than "1.0" as a whole. To this end, the switching control means 50 outputs a command to the hydraulic control circuit 42 for disengaging the switching clutch C0 and engaging the switching brake B0 to allow the differential portion 11 to function as an auxiliary power transmission with a fixed speed ratio $\gamma_0$, i.e., for instance, the speed ratio $\gamma_0$ equal to "0.7".

If the increasing-speed gear-position determining means 62 determines that no 5th-gear position is to be selected, the shifting mechanism 10 as a whole can obtain a decreasing-speed gear position with a speed ratio of "1.0" or more. To this end, the switching control means 50 outputs another command to the hydraulic control circuit 42 for engaging the switching clutch C0 and disengaging the switching brake B0 to allow the differential portion 11 to function as the auxiliary power transmission with the fixed speed ratio $\gamma_0$, i.e., for instance, the speed ratio $\gamma_0$ equal to "1".

Thus, the switching control means 50 causes the shifting mechanism 10 to be switched in the step-variable shifting state under which the operation is executed to selectively switch the gear positions of two kinds to either one gear position. With the differential portion 11 rendered operative to function as the auxiliary power transmission while the automatic shifting portion 20, connected to the differential portion 11 in series, is rendered operative to function as the step-variable transmission, the shifting mechanism 10 as a whole is rendered operative to function as the so-called step-variable automatic transmission.

[0080] On the contrary, if the switching control means 50 determines that the shifting mechanism 10 remains in the continuously variable shifting control region to be switched in the continuously variable shifting state, the shifting mechanism 10 as a whole can obtain the continuously variable shifting state. To this end, the switching control means 50 outputs a command to the hydraulic control circuit 42 for disengaging both the switching clutch C0 and the switching brake B0 so as to place the differential portion 11 in the continuously variable shifting state to enable an infinitely variable shifting operation to be executed. Simultaneously, the switching control means 50 outputs a signal to the hybrid control means 52 for permitting the hybrid control to be executed, while outputting a given signal to the step-variable shifting control means 54.
As used herein, the term "given signal" refers to a signal, by which the shifting mechanism 10 is fixed to a gear position for a predetermined continuously variable shifting state, or a signal for permitting the automatic shifting portion 20 to perform the automatic shifting according to, for instance, the shifting diagram, shown in Fig. 7, which is preliminarily stored in the memory means 56.

[0081] In this case, the step-variable shifting control means 54 performs the automatic shifting upon executing the operation excepting the operations to engage the switching clutch C0 and the switching brake B0 in the engagement operation table shown in Fig. 2. This causes the switching control means 50 to switch the differential portion 11 to the continuously variable shifting state to function as the continuously variable transmission, while rendering the automatic shifting portion 20, connected to the differential portion 11 in series, operative to function as the step-variable transmission. This allows a drive force to be obtained with an appropriate magnitude. Simultaneously, the rotation speed input to the automatic shifting portion 20, i.e., the rotation speed of the power transmitting member 18 is continuously varied for each gear position of the 1st-speed, 2nd-speed, 3rd-speed and 4th-speed positions of the automatic shifting portion 20, enabling the respective gear positions to be obtained in infinitely variable speed ratio ranges. Accordingly, since the speed ratio is continuously variable across the adjacent gear positions, the shifting mechanism 10 as a whole can obtain the overall speed ratio γT in an infinitely variable mode.

[0082] Now, Fig. 7 will be described more in detail. Fig. 7 is a view showing the relationships (shifting diagram and shifting map), preliminarily stored in the memory means 56, based on which the shifting of the automatic shifting portion 20 is determined, and representing one example of the shifting diagram plotted on a two-dimensional coordinate with parameters including the vehicle speed V and demanded output torque TOUT indicative of the drive force correlation value. In Fig. 7, the solid lines represent upshift lines and single dot lines represent downshift lines.

[0083] In Fig. 7, the broken lines represent a determining vehicle speed V1 and a determining output torque T1 for the switching control means 50 to determine the step-variable control region and the continuously variable control region. That is, the broken lines in Fig. 7 represent a high vehicle-speed determining line, forming a series of a determining vehicle speed V1 representing a predetermined high-speed drive determining line for determining a high speed running state of a hybrid vehicle, and a
high-output drive determining line, forming a series of determining output torque T1 representing a predetermined high-output drive determining line for determining the drive force correlation value related to the drive force of the hybrid vehicle. As used herein, the term "drive force correlation value" refers to determining output torque T1 that is preset for determining a high output drive for the automatic shifting portion 20 to provide output torque T_OUT at a high output.

[0084] A hysteresis is provided for determining the step-variable control region and the continuously variable control region as indicated by a double dot line in Fig. 7 in contrast to the broken line. That is, Fig. 7 represents a shifting diagram (switching map and relationship), preliminarily stored in terms of the parameters including the vehicle speed V, including the determining vehicle speed V1 and determining output torque T1, and output torque T_OUT, based on which the switching control means 50 executes the determination on a region as to which of the step-variable control region and the continuously variable control region belongs to the shifting mechanism 10.

The memory means 56 may preliminarily store the shifting map, inclusive of such a shifting diagram. Moreover, the shifting diagram may be of the type that includes at least one of the determining vehicle speed V1 and determining output torque T1 and may include a preliminarily stored shifting diagram with a parameter taking any of the vehicle speed V and output torque T_OUT.

[0085] The shifting diagram, the switching diagram or the drive-force source switching diagram or the like may be stored not in the map but in a determining formula for making comparison between a current vehicle speed V and a determining vehicle speed V1, and another determining formula or the like for making comparison between output torque T_OUT and determining output torque T1. In this casing, the switching control means 50 places the shifting mechanism 10 in the step-variable shifting state when the vehicle condition such as, for instance, an actual vehicle speed exceeds the determining vehicle speed V1. In addition, the switching control means 50 places the shifting mechanism 10 in the step-variable shifting state when the vehicle condition such as, for instance, output torque T_OUT of the automatic shifting portion 20 exceeds determining output torque T1.

[0086] When a malfunction or functional deterioration occurs in electrical control equipment such as an electric motor or the like used for rendering the differential portion 11 operative as the electrically controlled continuously variable transmission,
the switching control means 50 may be configured to place the shifting mechanism 10 in the step-variable shifting state on a priority basis for the purpose of ensuring the running of the vehicle even if the shifting mechanism 10 remains in the continuously variable control region. As used herein, the term "malfunction or functional deterioration in electrical control equipment" refers to a vehicle condition in which: functional degradation occurs in equipment related to the electrical path involved in the operation of the first electric motor M1 to generate electric energy and the operation executed in converting such electric energy to mechanical energy; that is, failures or functional deteriorations, caused by a breakdown or low temperature, occur in the first electric motor M1, the second electric motor M2, the inverter 58, the battery 60 and transmission paths interconnecting these component parts.

[0087] As used herein, the term "drive force correlation value" described above refers to a parameter corresponding to the drive force of the vehicle in one-to-one relation. Such a parameter may include not only drive torque or drive force delivered to the drive wheels 38 but also: output torque T_{OUT} of the automatic shifting portion 20; engine output torque TE; an acceleration value of the vehicle; an actual value such as engine output torque TE calculated based on, for instance, the accelerator operating or the throttle valve opening \( \theta_{TH} \) (or an intake air quantity, an air/fuel ratio or a fuel injection amount) and the engine rotation speed NE; or an estimated value such as engine output torque TE or the demanded output torque T_{OUT} for the automatic shifting portion 20 or demanded vehicle drive force calculated based on a displacement value of the accelerator pedal actuated by the driver or the throttle valve operating or the like. In addition, the drive torque may be calculated upon taking a differential ratio and a radius of each drive wheel 38 into consideration by referring to output torque T_{OUT} or the like or may be directly detected using a torque sensor or the like. This is true for each of other torques mentioned above.

[0088] For instance, the operation of the shifting mechanism 10 under the continuously variable shifting state during the running of the vehicle at the high speed turns out a consequence of deterioration in fuel economy. The determining vehicle speed V1 is determined to a value that can render the shifting mechanism 10 operative in the step-variable shifting state during the running of the vehicle at the high speed so as to address such an issue. Further, determining torque T1 is determined to a value that prevents reactive torque of the first electric motor M1 from covering a high output.
region of the engine during the running of the vehicle at a high output. That is, determining torque T1 is determined to such a value depending on, for instance, a characteristic of the first electric motor M1 that is possibly mounted with a reduced maximum output in electric energy for miniaturizing the first electric motor M1.

[0089] Fig. 8 represents a switching diagram (switching map and relationship), preliminarily stored in the memory means 56, which has an engine output line in the form of a boundary line to allow the switching control means 50 to determine a region based on the step-variable control region and the continuously variable control region using parameters including the engine rotation speed NE and engine torque TE. The switching control means 50 may execute the operation based on the engine rotation speed NE and engine torque TE by referring to the switching diagram shown in Fig. 8 in place of the switching diagram shown in Fig. 7. That is, the switching control means 50 may determine whether the vehicle condition, represented with the engine rotation speed NE and engine torque TE, lies in the step-variable control region or the continuously variable control region.

Further, Fig. 8 is also a conceptual view based on which the broken line in Fig. 7 is to be created. In other words, the broken line in Fig. 7 is also a switching line rewritten on a two-dimensional coordinate in terms of the parameters including the vehicle speed V and output torque \( T_{\text{OUT}} \) based on the relational diagram (map) shown in Fig. 8.

[0090] As indicated on the relationships shown in Fig. 7, the step-variable control region is set to lie in a high torque region, where output torque \( T_{\text{OUT}} \) is greater than the predetermined determining output torque T1, or a high vehicle speed region where the vehicle speed V is greater than the predetermined determining vehicle speed V1. Therefore, a step-variable shift drive mode is effectuated in a high drive torque region, where the engine 8 operates at relatively high torque, or the vehicle speed remaining in a relatively high speed region. Further, a continuously variable shift drive mode is effectuated in a low drive torque region, where the engine 8 operates at relatively low torque, or the vehicle speed remaining in a relatively low speed region, i.e., during a phase of the engine 8 operating in a commonly used output region.

[0091] As indicated by the relationship shown in Fig. 8, similarly, the step-variable control region is set to lie in a high-torque region with engine torque TE exceeding a predetermined given value TE1, a high-speed rotating region with the engine rotation
speed NE exceeding a predetermined given value NE1, or a high output region where the engine output calculated, based on engine torque TE and the engine rotation speed NE, is greater than a given value. Therefore, the step-variable shift drive mode is effectuated at relatively high torque, relatively high rotation speed or relatively high output of the engine 8. The continuously variable shift drive mode is effectuated at relatively low torque, relatively low rotation speed or relatively low output of the engine 8, i.e., in the commonly used output region of the engine 8. The boundary line, shown in Fig. 8, between the step-variable control region and the continuously variable control region corresponds to a high vehicle-speed determining line which is a series of a high vehicle-speed determining line and a high-output drive determining value which is a series of a high-output drive determining value.

[0092] With such a boundary line, for instance, during the running of the vehicle at a low/medium speed and low/medium output, the shifting mechanism 10 is placed in the continuously variable shifting state to ensure the vehicle to have improved fuel economy performance. During the running of the vehicle at a high speed with an actual vehicle speed V exceeding the determining vehicle speed V1, the shifting mechanism 10 is placed in the step-variable shifting state to act as the step-variable transmission. In this moment, the output of the engine 8 is transferred to the drive wheels 38 mainly through a mechanical power transmitting path. Thissuppresses a loss in conversion between the drive force and electric energy, generated when the shifting mechanism 10 is caused to act as the electrically controlled continuously variable transmission, providing improved fuel consumption.

[0093] During the running of the vehicle on the high output drive mode with the drive force correlation value, such as output torque TOUT or the like, which exceeds determining torque T1, the shifting mechanism 10 is placed in the step-variable shifting state to act as the step-variable transmission. In this moment, the output of the engine 8 is transferred to the drive wheels 38 mainly through the mechanical power transmitting path. In this case, the electrically controlled continuously variable transmission is caused to operate in the low/medium speed running region and the low/medium output running region of the vehicle. This enables a reduction in the maximum value of electric energy to be generated by the first electric motor M1, i.e., electric energy to be transmitted by the first electric motor M1, thereby causing the first electric motor M1 per se or a
vehicle drive apparatus including such a component part to be further miniaturized in structure.

[0094] According to another viewpoint, further, during the running of the vehicle on such a high output drive mode, the driver places more emphasis on a requirement for the drive force and less emphasis on a requirement for a mileage and, thus, the shifting mechanism 10 is switched to the step-variable shifting state (fixed shifting state) rather than to the continuously variable shifting state. With such a switching operation, the driver can enjoy a fluctuation in the engine rotation speed NE, i.e., a rhythmical variation in the engine rotation speed NE caused by the upshifting in the step-variable automatic shift running mode.

[0095] Fig. 9 shows, of the hydraulically operated control circuit 42, a circuit diagram relating to the linear solenoid valves SL1 to SL5 which control the operations of each of hydraulic actuators (hydraulic cylinders) AC1, AC2, AB1, AB2 and AB3 for the clutches C1 and C2 and brakes B1 to B3, in the present embodiment. Further, the Fig. 9 is also a circuit diagram showing structures of the mechanical oil pump 70, the electric oil pump 72 and the regulator valve 76 which supply a line pressure PL to each hydraulic actuator of these parts in the present embodiment.

[0096] In Fig. 9, the line hydraulic PL is respectively regulated to the engagement pressures PC1, PC2, PB1, PB2, and PB3 by the linear solenoid valves SL1 to SL5 according to the command signals from the electronic control device 40, and the engagement pressures PC1, PC2, PB1, PB2, and PB3 are directly supplied to each of the hydraulic actuators AC1, AC2, AB1, AB2, and AB3.

[0097] Any of the linear solenoid valves SL1 to SL5 are basically of the same configuration, and are independently excited and non-excited by the electronic control device 40. The hydraulic pressure of each of the hydraulic actuators AC1, AC2, AB1, AB2, and AB3 is independently regulated and controlled, so that the engagement pressures PC1, PC2, PB1, PB2, and PB3 for the clutches C1 to C4 and the brakes B1 and B2 are controlled. In the automatic shifting portion 20, each gear shift position, for instance, as shown in the engagement operation table of Fig. 2, is established by engaging the predetermined engaging devices with each other. In the shifting control of the automatic shifting portion 20, a so-called clutch to clutch shifting is executed in which, for instance, the release and engagement of the clutch C and the brake B involved in the shifting are simultaneously controlled.
[0098] The present embodiment includes two oil pumps, i.e. the mechanical oil pump 70 connected to the engine 8 to be synchronously driven therewith, and the electric oil pump 72 driven by the electric power. The mechanical oil pump 70 mechanical-controlled oil pump 70 is configured as a gear type oil pump formed by a driven gear and a drive gear (not shown). The mechanical oil pump 70 is connected to an output shaft of the engine 8 to be driven thereby. As a result, the mechanical oil pump 70 is driven during driving (rotating) the engine 8, and the mechanical oil pump 70 is stopped during stopping the engine 8. The electrical oil pump 72 composed of a fixed capacity type gear type pump, is driven by an oil pump motor 74 functioning as a drive source and capable of controlling the rotation speed. Controlling the rotation speed of the oil pump motor 74, can control a discharge amount of the electric oil pump 72.

[0099] The mechanical oil pump 70 and the electric oil pump 72 are arranged in parallel with each other, and by driving either one thereof or both of them, the operation oil stored in an oil pan (not shown) is drawn up through a strainer 78. This operation oil is regulated to the line pressure PL by a regulator valve 76 arranged at the downstream side of these oil pumps. The regulator valve 76 is a relief type pressure regulator valve, in which upon absence of the hydraulic pressure supply, a spool 79 moves to the full-closed position by the elastic force of a spring 77 to bring the regulator valve 76 into a valve closed state (non-operating state).

[0100] When the mechanical oil pump 70 or the electric oil pump 72 is driven, the regulator valve 76 is supplied with the hydraulic pressure so as to move the spool 79 to a valve opening position, thereby opening the regulator valve 76. The regulator valve 76 regulates an original hydraulic pressure as the line hydraulic pressure PL, to a value corresponding to an engine load and the like represented by opening of an accelerator or opening of a throttle, by the electric oil pump 72 and the mechanical oil pump 70 rotationally driven by the engine 8.

[0101] Returning to Fig. 6, the oil pump control means 110 controls a drive state of the mechanical oil pump 70 and the electric oil pump 72 according to the vehicle condition. For instance, during driving the engine, the mechanical oil pump 70 is driven by the engine 8, and the oil pump control means 110 stops the electric oil pump 72. Further, in a motor running state without driving the engine 8, due to non-driving of the mechanical oil pump 70, the oil pump control means 110 drives the electric oil
pump 72 to generate the hydraulic pressure. Further, even in the motor driving state upon non-shifting of the automatic shifting portion 20, due to no request of the hydraulic pressure, the oil pump control means 110 stops the electric oil pump 72.

[0102] Here, for instance, when the vehicle is started from a vehicle stopped state, the mechanical oil pump 70 and the electric oil pump 72 are operated to generate the hydraulic pressures from the stopping state. At this time, since the vehicle is usually started by the motor (second electric motor M2) and the engine 8 is not started, the hydraulic pressure is generated by the electric oil pump 72. The regulating valves such as the linear solenoid valves SL1 to SL5 and the regulator valve 76 are all closed (non-operated) to establish a closed hydraulic circuit state. When the electric oil pump 72 is started in this state, the line pressure PL instantaneously rises in the transition period to the opened (operated) state of these regulating valves, whereby a large load is applied to the electric oil pump 72.

Particularly, when the operation oil is low in the oil temperature, i.e., its viscosity is high, due to further increased operation resistance of the valve and the like, a high load is likely to be applied to the electric oil pump 72. Further, accompanied with this high load, there was a fear that an excessive current is generated in the electronic control circuit for supplying the electric power to the oil pump motor 74 of the electric oil pump 72.

[0103] In view of this, when starting the electric oil pump 72 from a state where both of the mechanical oil pump 70 and the electric oil pump 72 are stopped, the oil pump control means 110 executes a control of reducing the load applied to the electric oil pump 72. Hereinafter, a description will be focused on this control.

The oil pump control means 110 is comprised of a target control value setting means 112, an actual control value detecting means 114, a target load value setting means 116, an actual load value detecting means 118, and a command control value setting means 120.

[0104] The target control value setting means 112 operates to set a rotation speed Npomfwd of the oil pump motor 74 serving as a target upon starting the electric oil pump 72. This target rotation speed Npomfwd is, for instance, set to a suitable value using a mode map or a relational equation set in advance based on an accelerator opening Acc, a gear position, a vehicle speed V and the like. Controlling the oil pump motor 74 in the target rotation speed Npomfwd contributes to obtain a suitable line
pressure PL in consideration of the fuel consumption and the like of the vehicle. Further, the target rotation speed Nopmfwd of the present embodiment corresponds to the target control value of the present invention.

[0105] The actual control value detecting means 114 successively operates to detect an actual rotation speed Nopm of the oil pump motor 74 which is for instance successively detected by a rotation speed sensor 80 provided in the oil pump motor 74 for detecting the actual rotation speed Nopm. The actual rotation speed Nopm of the present embodiment corresponds to the actual control value of the present invention.

[0106] The target load value setting means 116 operates to set a target load value of the electric oil pump 72. Here, due to the relationship where as the line pressure PL rises the load value of the electric oil pump 72 becomes high, the magnitude of the line pressure PL corresponds to the load value of the electric oil pump 72. Hence, in the present embodiment, for the target load value of the electric oil pump 72, a line pressure (target line pressure) PLfwd serving as a target is set after starting the electric oil pump 72.

This target line pressure PLfwd is, similarly to the target rotation speed Nopmfwd of the oil pump motor 74, set to a suitable value using a mode map or a relational expression set in advance based on an accelerator opening Acc, a gear shift, a vehicle speed V and the like. For instance, due to the relationship where as the accelerator opening Acc becomes large the engine torque becomes large, the engagement torques of the clutches C1 and C2 and the brakes B1 to B3 are required to be made large to transmit the engine torque. Thus, the target line pressure PLfwd is set in the relatively higher value. The target line pressure PLfwd of the present embodiment corresponds to the target load value of the present invention.

[0107] The actual load value detecting means 118 operates to obtain an actual load value of the electric oil pump 72. Here, due to the relationship where as the line pressure PL rises the load value of the electric oil pump 72 becomes high, the magnitude of the line pressure PL corresponds to the load value of the electric oil pump 72. Hence, in the present embodiment, as the actual load value, the actual line pressure PL supplied to the hydraulic control circuit 42 is detected, that is, for instance, directly detected by a hydraulic pressure sensor 82 arranged therein. Upon starting the electric oil pump 72, the regulator valve 76 for regulating the line pressure PL is in the operation transition time and does not function as a regulating valve. Thus, the
hydraulic pressure of the operation oil discharged by the electric oil pump 72 functions as the line pressure PL. The line pressure PL of the present embodiment corresponds to the actual load value of the present invention.

[0108] Instead of directly detecting the line pressure PL by the actual load value detecting means 118, the current value of the electric oil pump 72, specifically, the current value supplied to the oil pump motor 74 may be detected. Since this current value become large as the load applied to the electric oil pump 72 becomes large, similarly to the line pressure PL, the magnitude of this current value corresponds to the load value applied to the electric oil pump 72. For this reason, the actual load value of the electric oil pump 72 can be detected by the current value.

[0109] Further, the actual load value detecting means 118 may estimate the actual load value of the electric oil pump 72 based on the detection value correlated with the actual load value of the electric oil pump 72. Specifically, the actual load value of the electric oil pump 72 can be estimated, for instance, based on temperature of the electric oil pump 72, current value or the estimation value thereof of the control device of the electric oil pump 72, temperature of the control device of the electric oil pump 72, and engagement pressure or the estimation value thereof of the hydraulically operated frictional engaging device in the automatic shifting portion 20.

[0110] For instance, temperature of the electric oil pump and temperature of the control device of the electric oil pump have tendency to become high as the actual load value of the electric oil pump 72 increases. The actual load value of the electric oil pump 72 can be estimated based on this relation. Further, current value or estimation value thereof of the electronic control circuit (control device) for controlling the electric oil pump 72 has tendency to become large as the actual load value of the electric oil pump 72 increases. The actual load value of the electric oil pump 72 can be estimated based on this relation.

Engagement pressure or estimation value thereof of the hydraulically operated frictional engaging device in the automatic shifting unit 20 has tendency to become large as the actual load value of the electric oil pump 72 increases. The actual load value of the electric oil pump 72 can be estimated based on this relation. Even when the present embodiment is executed based on the detection value correlated with these actual load values of the electric oil pump 72, the same effect as the present control can be obtained.
[0111] The command control value setting means 120 sets the command rotation speed Nopmtg of the oil pump motor 74 of the electric oil pump 72, based on following factors. The factors include a target rotation speed Nopmfwd set by the target control value setting means 112, an actual rotation speed Nopm detected by the actual control value detecting means 114, a target line pressure PLfwd set by the target load value setting means 116, and the line pressure PL detected by the actual load value detecting means 118. The command rotation speed Nopmtg of the present embodiment corresponds to the command control value of the present invention.

[0112] Here, the command rotation speed Nopmtg is successively set based on an equation (1) shown below. "Kopm", being a proportionality constant (rotation speed gain) to be multiplied by the difference between the target rotation speed Nopmfwd and the rotation speed Nopm, is set to an optimum value in advance through experiments and the like. "Kpl" shows a proportionality constant (load gain) to be multiplied by the difference between the target line pressure PLfwd and the line pressure PL. When the target line pressure PLfwd and the line pressure PL are substituted by the current value of the electric oil pump 72, the load gain Kpl is changed to the current value gain Ki.

\[
\text{Nopmtg} = \text{Nopmfwd} + \text{Kopm} \times (\text{Nopmfwd} - \text{Nopm}) + \text{Kpl} \times (\text{PLfwd} - \text{PL}) \quad \text{(1)}
\]

[0113] In the right term of the equation (1), a first term (Nopmfwd), being a target rotation speed Nopmfwd, corresponds to a rotation speed value (feed forward value) controlled so as to be attained finally. A second term \((\text{Kopm} \times (\text{Nopmfwd} - \text{Nopm}))\) is a so-called feedback term, and is set to be zero in its difference based on compared result between the target rotation speed Nopmfwd and the rotation speed Nopm. Further, a third term \((\text{Kpl} \times (\text{PLfwd} - \text{PL}))\) is a feedback term using the difference of the load (line pressure) as the essential part of the present invention, and specifically, is set to be zero in its difference based on comparison between the target line pressure PLfwd and the line pressure PL.

[0114] When the feedback term shown in the third term is not present, the line pressure PL rapidly rises upon starting the electric oil pump 72 as shown by the broken line of Fig. 12, which imposes a high load instantaneously to the electric oil pump 72. However, adding the feedback term (third term) based on the difference of the line
pressure PL can eliminate the rapid rise of the PL pressure, which suppresses the load applied to the electric oil pump 72. Specifically, when the line pressure PL becomes larger than the target line pressure PL fwd, the third term has a negative value, which makes the command rotation speed Nopmtg in the left term small. In this manner, the command rotation speed Nopmtg of the electric oil pump 72 is feedback-controlled in the direction to suppress the rapid rise of the line pressure PL.

[0116] Here, in the present embodiment, both values of the rotation speed gain Kopm and the load gain KpL (or the current value gain Ki) are set based on the oil temperature of the operation oil flowing inside the hydraulic control circuit 42. For instance, as the oil temperature of the operation oil is lowered, each of the gain values is set large. The gain value is set to a suitable value by a map or a relational equation showing a relationship between the gain values and the oil temperature set in advance through the experiment and the like. The oil temperature of the operation oil is, for instance, detected by an oil temperature sensor 84 provided inside the hydraulic control circuit 42.

[0117] Now, a rotation speed command of the oil pump motor 74, specifically, a voltage signal is output from an electronic control device to be positioned upstream, based on which a current energization command is output from the electronic control device for controlling the electric oil pump 72. That is, the current value flowing to the electric oil pump 72 (oil pump motor 74) is determined according to the load applied to the electric oil pump 72.

An equation (2) shows a relationship between a load torque and a current value applied to the electric oil pump 72. Here "τ" indicates a load applied to the electric oil pump 72, "Kt" a coefficient of the load (or the current), and "Ia" the current value of the electric oil pump 72 (oil pump motor 74), respectively.

The rotation speed of the electric oil pump 72 (oil pump motor 74) is shown by an equation (3), wherein "N" indicates a rotation speed of the oil pump motor 74, "Vt" a voltage, "Ke" a constant of the dielectric voltage (or rotation speed), and "Ra" a motor resistance, respectively. The upstream electronic control device and the electronic control device for controlling the electric oil pump 72 are conceptually distinguished, and they can be both built into the same electronic control device 40, or they can be separately configured as a device.

[0118] \[ \tau = Kt \times Ia \]  \hspace{1cm} \textit{*****}(2)
\[ N = V_t/K_e - (R_a/(K_t \times K_e)) \times \tau \text{  (3)} \]

[0119] As apparent from the equation (3), if the load \( \tau \) is constant, the voltage \( V_t \) and the rotation speed \( N \) of the oil pump motor 74 have a proportional relation. Further, in the equation (1), in an equation of the rotation speed omitting the third term \((=K_p \times (PLfwd-PL))\), the load \( \tau \) is considered as constant. Thus, the voltage \( V_t \) proportional to the rotation speed \( N \) is output. In Fig. 10 showing a relationship between the load \( \tau \) and the rotation speed \( N \), the conventional relation is shown by the broken line. Here, upon starting the oil pump motor 74, the high load is imposed due to the small actual rotation speed \( N \), which proportionally increases the current value \( I_a \). In contrast to this, controlling the command rotation speed \( Nopmtg \) along the equation (1) can reduce the load \( \tau \), by reducing the command rotation speed \( Nopmtg \) as shown by the solid line in Fig. 10.

[0120] Returning to Fig. 6, an engine stopping determining means 122 determines whether or not the vehicle is under the motor running state, that is, whether or not the engine 8 is in the stopped state. The stopped determination of the engine 8 is performed, for instance, based on whether or not the rotation speed \( NE \) of the engine 8 is zero. The stopped determined of the engine 8 means that the mechanical oil pump 70 is not driven.

[0121] An electric oil pump starting request determining means 124 (hereinafter, described as a "pump starting request determining means 124") determines whether or not the request for starting the electric oil pump 72 is arisen. For instance, when a start button is turned on after insertion of an ignition key in a vehicle stopping state, or when the running state is switched from the engine running state to the motor running state and the like during driving, requirement for starting the electric oil pump 72 is determined.

[0122] The oil pump control means 110 executes in a predetermined case a feedback control of the electric oil pump 72 based on the command rotation speed \( Nopmtg \) calculated by the command control value setting means 120. The predetermined case means occasion when the engine stopping determining means 122 determines the stopping of the engine 8 and the pump starting request determining means 124 determines the starting requirement of the electric oil pump 72.

[0123] Fig. 11 is a flowchart describing a principal part of the control operation by the electronic control device 40. That is, the flowchart describes a control operation for
reducing the load applied to the electric oil pump 72 upon starting the electric oil pump 72 from the state in which the mechanical oil pump 70 and the electric oil pump 72 are both stopped. Fig. 12 shows one example of the control operation of the flowchart shown in Fig. 11, and is a time chart showing an operation control for starting the electric oil pump 72 upon stopped state of the engine 8.

[0124] First, in step SA1 (hereinafter, "step" will be omitted) corresponding to the engine stopping determining means 122, it is determined whether or not the motor is under running, that is, whether or not the engine 8 is held in a stopped state. When SA1 is disaffirmed, other controls are executed in SA5. When SA1 is affirmed, in SA2 corresponding to the pump starting request determining means 124, it is determined whether or not a starting request for the electric oil pump 72 is arisen. When SA2 is disaffirmed, other controls are executed in SA5. When SA2 is affirmed, in SA3 corresponding to the actual load value detecting means 118, the load actually applied to the electric oil pump 72 is detected. The load is detected based on the current value applied to the electronic control device for controlling the electric oil pump 72, the line pressure PL, the engagement pressure or the temperature of the electric oil pump 72, and the like.

[0125] In SA4 corresponding to the command control value setting means 120, the target control value setting means 112, the actual control value detecting means 114, the target load value setting means 116, and the oil pump control means 110, the starting control of the electric oil pump 72 is executed based on the load of the electric oil pump 72. Specifically, the command control value setting means 120 sequentially calculates the command rotation speed Nopmtg, based on which the discharge amount of the electric oil pump 72 is controlled.

In SA4, when a difference between the line pressure PL and the target line pressure PLfwd, or a difference between the load value (current value) and the target load value (current value) of the electric oil pump 72 is equal to or more than the predetermined value set through the experiments and the like, a feed forward term variable depending on the oil temperature of the operation oil can be added to the command rotation speed Nopmtg.

[0126] In Fig. 12, broken lines show a state in which the feedback term by the difference of the load of the electric oil pump 72 is absent, and solid lines show a state in which the feedback term (the third term of the equation (1)) by the difference of the
load of the electric oil pump 72 is added. When the electric oil pump 72 is started at a
time point, i.e., timing t1 with no feedback term by the load difference, the line
pressure PL instantaneously rises as shown by the broken line, which may cause an
excessive current to flow in the electronic control device that controls the electric oil
pump 72.

Meanwhile, thanks to addition of the feedback term by the load difference, the
command rotation speed Nopmtg gradually increases at a time point t1 as shown by the
solid line, so that the instantaneous rise of the line pressure PL is suppressed. As a
result, suppressing the load applied to the electric oil pump 72 can suppress the
excessive current generated in the electronic control device.

[0127] As described above, according to the present embodiment, the command
control value setting means 120 sets, based on the comparison between the line
pressure PL (actual load value) by the electric oil pump 72 and the target line pressure
PLfwd (target load value), the command rotation speed Nopmtg (command control
value). Hence, with suppressing the high load applied to the electric oil pump 72 upon
starting time thereof, the responsiveness of the hydraulic pressure can be secured. As a
result, the excessive current generated in the electronic control circuit for controlling
the electric oil pump 72 can be suppressed.

[0128] According to the present embodiment, the command control value setting
means 120 sets, based on the value calculated by multiplying the difference between
the line pressure PL (actual load value) by the electric oil pump 72 and the target line
pressure PLfwd (target load value) by the load gain Kpl (proportionality constant), the
command rotation speed Nopmtg (command control value). Since the load gain Kpl is
set based on the oil temperature of the operation oil, the appropriate control can be
performed according to the oil temperature of the operation oil. For instance, the load
gain Kpl is set high for the low oil temperature, and it is set low for the high oil
temperature. As a result, with suppressing the high load applied to the electric oil
pump 72, the further effective responsiveness of the hydraulic pressure can be secured.

[0129] According to the present embodiment, the actual load value of the electric oil
pump 72 corresponds to the current value of the electric oil pump 72, or the hydraulic
pressure value of the operation oil discharged from the electric oil pump 72. Hence,
detecting the current value or the hydraulic pressure enables easy detection of the
actual load value of the electric oil pump 72.
According to the present embodiment, since the command rotation speed Nopmtg (command control value) of the electric oil pump 72 is set based on the compared result between the target rotation speed Nopmfwd (target control value) and the actual rotation speed Nopm (actual control value) of the electric oil pump 72, the rapid variation of the rotation speed of the electric oil pump 72 can be suppressed.

[0130] According to the present embodiment, the command control value setting means 120 adds, a value (feed forward value) variable depending on the oil temperature and the like of the operation oil, to the command control value. This is performed when the difference between the line pressure PL (actual load value) by the electric oil pump 72 and the target line pressure PLfwd (target load value) is equal to or more than the predetermined value. As a result, the feed forward value can suppresses the high load applied to the electric oil pump 72.

[0131] Further, according to the present embodiment, the differential portion 11 is provided to control the differential state between the rotation speeds of the input shaft 14 connected with the engine 8 and the power transmitting member 18. This is performed by controlling the drive condition of the first electric motor M1 connected to the rotary element of the power distribution mechanism 16 in the power transmissive state. Therefore, controlling the first electric motor M1 can continuously vary the speed ratio, whereby a wide range of the drive force can be obtained.

[0132] According to the present embodiment, since the electric oil pump 72 directly connected to the engine 8 is driven during driving thereof, to thereby render sufficient hydraulic pressure. Further, there is no need to separately provide the drive source for driving the electric oil pump 72, which avoids the increase in the number of parts.

The present embodiment includes the differential-portion planetary gear unit 24, and the first and second electric motors M1 and M2. The first and second electric motors M1 and M2 can control the rotation speed of each of the rotation elements of the differential-portion planetary gear unit 24. The structure including one unit and the two motors can be made relatively compact.

According to the present embodiment, the actual load value of the electric oil pump 72 is estimated based on the detection value correlated with the actual load value of the electric oil pump 72. Therefore, not only the actual load value of the electric oil pump 72 can be easily estimated, but the error of the estimated value relative to the actual load value can be suppressed within the range not affecting the present control.
[0133] Heretofore, the embodiment of the present invention has been described in detail with reference to the drawings, but the present invention is also applicable to other modes.

[0134] For instance, in the above described embodiment, determining means, not executing the present control when the oil temperature of the operation oil becomes higher than the predetermined temperature and the like, can be added to execute the flowchart of Fig. 11 only when the operation oil is low in the temperature thereof. In this manner, the load necessary for the control can be reduced.

[0135] In the above described embodiment, when the difference between the line pressure PL (actual load value) by the electric oil pump 72 and the target line pressure PL_fwd (target load value) is more than the predetermined value, the value (feed forward value) variable depending the oil temperature and the like of the operation oil can be added to the command control value. However, the feed forward control is not necessarily executed, and executing the feedback control alone can render the effect of the present invention.

[0136] In the above described embodiment, the mechanical oil pump 70 is synchronously driven with the engine 8, but the present invention is not particularly restricted to such a configuration. The mechanical oil pump 70 may be driven by other drive sources. Further, the invention is not restricted to the configuration in which the mechanical oil pump 70 and the electric oil pump 72 are provided in parallel, but the configuration including two electric oil pumps or only one the electric oil pump 72, can be adopted.

[0137] In the illustrated embodiment, the second electric motor M2 is connected to the power transmitting member 18 in series. However, the connecting position of the second electric motor M2 is not necessarily limited to connecting arrangement described above, but the second electric motor M2 may be directly connected to the power transmitting path between the differential portion 11 or indirectly connected thereto via a transmission or the like.

In the illustrated embodiment, while the differential portion 11 is configured to function as the electrically controlled continuously variable transmission in which the speed ratio $\gamma 0$ is continuously varied from the minimal value $\gamma 0_{\text{min}}$ to the maximal value $\gamma 0_{\text{max}}$, the present invention may be applied even to a case wherein the speed ratio $\gamma 0$ of
the differential portion 11 is not continuously varied but pretended to vary step-by-step with the use of a differential action.

[0138] With the power distribution mechanisms 16 of the illustrated embodiments, the first carrier CA1 is connected to the engine 8; the first sun gear S1 is connected to the first electric motor M1; and the first ring gear R1 is connected to the power transmitting member 18. However, the present invention is not necessarily limited to such connecting arrangement, and the engine 8, first electric motor M1 and power transmitting member 18 have no objection to be connected to either one of the three elements CA1, S1 and R1 of the first planetary gear set 24.

[0139] Although the illustrated embodiment has been described with reference to the engine 8 directly connected to the input shaft 14, these component parts may suffice to be operatively connected via, for instance, gears, belts or the like. No need may arise for the engine 8 and the input shaft 14 to be necessarily disposed on a common axis.

Further, while the illustrated embodiment has been described with reference to the first electric motor M1 and the second electric motor M2 wherein the first electric motor M1 is coaxially disposed with the drive apparatus input shaft 14 and connected to the first sun gear S1 upon which the second electric motor M2 is connected to the power transmitting member 18. However, no need arises for these component parts to be necessarily placed in such connecting arrangement. For example, the first electric motor M1 may be connected to the first sun gear S1 through gears, a belt or the like, and the second electric motor M2 may be connected to the power transmitting member 18.

[0140] In the illustrated embodiment, further, the hydraulic-type frictionally coupling devices such as the first and second clutches C1, C2 may include magnetic type clutches such as powder (magnetic powder) clutches, electromagnetic clutches and meshing type dog clutches, and electromagnetic type and mechanical coupling devices. For instance, with the electromagnetic clutches being employed, the hydraulic control circuit 42 may not include a valve device for switching hydraulic passages and may be replaced with a switching device or electromagnetically operated switching device or the like that are operative to switch electrical command signal circuits for electromagnetic clutches.

[0141] While the illustrated embodiment has been described above with reference to the automatic transmission portion 20 that is connected to the differential portion 11 in series via the power transmitting member 18, a countershaft may be provided in parallel to the input shaft 14 to allow the automatic transmission portion 20 to be coaxially
disposed on an axis of the countershaft. In this case, the differential portion 11 and the automatic transmission portion 20 may be connected to each other in power transmitting capability via a set of transmitting members structured of, for instance, a counter-gear pair acting as the power transmitting member 18, a sprocket and a chain.

Further, the power distributing mechanism 16 of the illustrated embodiment may include, for instance, a differential gear set in which a pinion, rotatably driven with the engine, and a pair of bevel gears, held in meshing engagement with the pinion, are operatively connected to the first electric motor M1 and the power transmitting member 18 (second electric motor M2).

[0142] The power distributing mechanism 16 of the illustrated embodiment having been described above as including one set of planetary gear units, may include two or more sets of planetary gear units that are arranged to function as a transmission having three or more speed positions under a non-differential state (fixed shifting state). In addition, the planetary gear unit is not limited to the single-pinion type, but may be of a double-pinion type. Following structure can be adopted. When the power distributing mechanism 16 is comprised of two or more sets of planetary gear units, the engine 8, first and second electric motors M1 and M2, and power transmitting member 18 can be connected to each of rotary elements of the planetary gear units in the power transmissive state. Further, the step variable shifting and the continuously variable shifting state can be switched by controlling the clutch C and brake B connected to each of rotary elements of the planetary gear unit.

[0143] In the illustrated embodiment, although the engine 8 and the differential portion 11 are directly connected with each other, such connecting mode is not essential. The engine 8 and the differential portion 11 can be connected via the clutch etc.

In the illustrated embodiment, the differential portion 11 and the automatic shifting portion 20 are connected to each other in series. However, the present invention can be applied to a structure even if the differential portion 11 and the automatic shifting portion 20 are mechanically independent from each other, provided that a whole of the shifting mechanism 10 has a function to achieve an electrically controlled differential action enabling a differential state to be electrically varied, and a function to perform a shifting on a principle different from the function of the electrically controlled differential action. Also the connecting position and the connecting arrangement of the differential portion 11 and the automatic shifting portion 20 are not necessarily limited,
but may be selected freely. Further, the present invention can be applied to the shifting mechanism which has the functions to perform an electrically controlled differential action and a shifting action, even if a part of structure is overlapped or a whole of structure is common.

[0144] In the illustrated embodiment, the shift operating device 48 includes a shift lever 49 which is operated to select one of the shifting positions SP of the plural kinds. However, the shift operating device 48 may include a switch which selects shift positions SP of the plural kinds, for instance, a push-button switch, a slide switch or the like, or a device which is reacted to a driver’s voice independent from the manual operation to switch shift positions SP of the plural kinds, or a device which is actuated by driver’s foot which switches a shift positions SP of the plural kinds, or the like in place of the shift lever 49.

In the illustrated embodiment, the varying range i.e. shifting range is selected by the shift lever 49 to be operated to select a position "M". However, setting of the gear position, i.e., setting of the highest speed gear position of each varying range as the gear position may be adopted. In this case, the speed shifting is executed by switching the gear position in the automatic shifting portion 20. For instance, if the shift lever 49 is manually operated to select the upshift position "+" or the downshift position "-" in the position "M", either one of a 1st-speed gear position or a 5th-speed gear position is selected by the operation of the shift lever 49 in the automatic shifting portion 20.

[0145] In the illustrated embodiment, the automatic shifting position 20 adopts a step-variable transmission which enables to have four speed positions. However, the gear shift position of the automatic shifting portion 20 is not limited to four speed positions, but may be to five speed positions or the like. Further, a structure of the automatic shifting position 20 is not limited to that in the illustrated embodiment, but may be changed freely.

The foregoing merely illustrates the embodiments for illustrating the principles of the present invention. It will be appreciated by those skilled in the art that various modifications and alternatives to those details could be developed in the light of the overall teachings of the disclosure.
CLAIMS

1. A control device for vehicular hydraulic control circuit, wherein
   the vehicular hydraulic control circuit includes an electric oil pump; and
   the control device is comprised of an actual load value detecting portion for
   obtaining or estimating an actual load value of the electric oil pump, a target load value
   setting portion for setting a target load value of the electric oil pump, and a command
   control value setting portion for setting a command control value of the electric oil
   pump,
   wherein the command control value setting portion sets a command control
   value based on a compared result between the actual load value and the target load
   value of the electric oil pump.

2. The control device for vehicular hydraulic control circuit according to claim 1,
   wherein the command control value setting means sets the command control value
   based on a value obtained by multiplying a difference between the actual load value
   and the target load value of the electric oil pump by a proportionality constant, the
   proportionality constant being set based on an oil temperature of an operation oil.

3. The control device for vehicular hydraulic control circuit according to claim 1 or 2,
   wherein the actual load value of the electric oil pump corresponds to a current value of
   the electric oil pump or a value of the hydraulic pressure of the operation oil
   discharged by the electric oil pump.

4. The control device for vehicular hydraulic control circuit according to any one of
   claims 1 to 3, wherein the command control value of the electric oil pump is set based
   on the compared result between the target control value and the actual control value of
   the electric oil pump.

5. The control device for vehicular hydraulic control circuit according to any one of
   claims 1 to 4, wherein the actual load value detecting portion estimates the actual load
   value of the electric oil pump based on a detection value correlated with the actual load
   value of the electric oil pump.
6. The control device for vehicular hydraulic control circuit according to claim 2, wherein the command control value setting portion adds a value variable depending on the oil temperature of the operation oil to the command control value, when a difference between the actual load value and the target load value of the electric oil pump is equal to or more than a predetermined value.

7. The control device for vehicular hydraulic control circuit according claim 1, wherein the target control value is a rotation speed of an oil pump motor serving as a target upon starting the electric oil pump, and the actual control value is an actual rotation speed of the oil pump motor successively detected.

8. The control device for vehicular hydraulic control circuit according claim 1, wherein a power transmitting apparatus of the vehicle include a electric differential portion in which a differential state between a rotation speed of an input connected to an engine and a rotation speed of an output shaft is controlled by controlling an operation state of an electric motor connected to a differential mechanism.

9. The control device for vehicular hydraulic control circuit according to claim 8, wherein the vehicular hydraulic control circuit further including a mechanical oil pump connected to the engine.
FIG. 1

FIG. 2

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☒ ENGAGED
☒ ENGAGED UPON STEP-VARIABLE
RELEASED UPON CONTINUOUSLY-VARIABLE
FIG. 5

SWITCHING SAMPLE OF M POSITION

- D RANGE
- 4 RANGE
- 3 RANGE
- 2 RANGE
- L RANGE
FIG. 11

START

SA1
ENGINE STOPPED?

No

Yes

SA2
STARTING REQUEST FOR ELECTRIC OIL PUMP ARisen?

No

Yes

SA3
EXECUTE LOAD MEASUREMENT/CALCULATION OF ELECTRIC OIL PUMP

SA4
EXECUTE STARTING CONTROL OF ELECTRIC OIL PUMP

SA5
EXECUTE OTHER CONTROLS

RETURN
FIG. 12

- **Electric Oil Pump Started**
  - **Solid Line**: With Feedback Control Depending on Load
  - **Broken Line**: Without Feedback Control Depending on Load

- **Command Rotation Speed of Electric Oil Pump (Nomprm)**
  - **Engine Rotation Speed (Ne)**
  - **Line Pressure (PL)**

**Time**
A. CLASSIFICATION OF SUBJECT MATTER

INV. F16H61/00

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)
F16H

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)
EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

<table>
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<td>EP 0 952 349 A (NISSAN MOTOR [JP]) 27 October 1999 (1999-10-27) abstract; figures 2-4</td>
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[X] Further documents are listed in the continuation of Box C. 
[X] See patent family annex.

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  *A* document defining the general state of the art which is not considered to be of particular relevance
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Date of the actual completion of the international search:
21 November 2008

Date of mailing of the international search report:
03/12/2008

Name and mailing address of the ISA/
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Authorized officer

Masset, Candie
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