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A FUNDAMENTAL CONSIDERATION ON THE SHIFT LOGIC OF AUTOMATIC TRANSMISSIONS FOR ELECTRIC VEHICLES

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Abstract

An electric vehicle is modeled from the viewpoint of gear-shifting, and a fundamental consideration on its shift logic is performed, attempting to clarify the essence of shift logic for automobiles.

This report proposes an improved shift logic which contains vehicle acceleration as one of its fundamental variables, while a conventional one is constructed from steady-state driving conditions. A computer simulation is executed to compare the efficiencies of the both shift logics. Finally, to clarify the relationship between gear-shifting and an accelerator, the shift logic which utilizes both inputs and outputs of a powertrain as its fundamental variables is constructed, and investigated on its difference from the shift logic whose fundamental variables are outputs of a powertrain alone.

1. Introduction

This report treats a shift logic problem of automatic transmissions for electric vehicles, and attempts to clarify the essence of shift logic for automobiles.

In recent years, demands for more improved performance and efficiency of automobiles require deeper studies and developements for powertrains[1,5]. In case of internal combustion (I. C.) engine vehicles, it is attempted to clarify the dynamics of I. C. engines from the side of control system theory[3,4], and to apply the optimal control theory to engine controls[2]. It is transmissions, however, that hold the key of displaying the performance of I. C. engines. And improvement in the performance and efficiency of automatic transmissions in recent years is remarkable as seen in the adoption of Overdrive mechanism[9] etc., or in the examination of optimal shift points[8]. In case of electric vehicles, automatic transmissions are newly equipped to some of them so as to replace their electric drives with smaller ones without loss of vehicle performance, resulting in lighter total vehicle weight[5,6]. As described above, automatic transmissions become more and more impotant.

On the both vehicles, however, the shift logics, which determine when automatic transmissions should shift under every circumstance, are constructed from steady-state driving conditions[6,8]. So it does not always seem adequate to use them on a general urban driving pattern which contains frequent vehicle accelerations and decelerations. Moreover some shift points are decided empirically, because a theoretical framework of shift logic is not necessarily completed. Therefore, it is important to reexamine the shift logic from the basic point of view, paying attention to the trend of the vehicle studies described above.

Electric vehicles are hoped to be an alternative of present-day automobiles, and the dynamics of an electric motor is well-known and simple over the whole driving region, compared with a I. C. engine. So an electric vehicle seems preferable for obtaining a fundamental insight into a shift logic problem. From these reasons, this report examines the shift logic problem of electric vehicles and attempts to clarify the nature of shift logic common to the both vehicles.

At first, an electric vehicle is modeled from the viewpoint of gear-shifting, and some model properties are noted on the similarities to I. C. engine vehicles. Secondly, an improved shift logic, which contains vehicle acceleration as one of its fundamental variables, is proposed, while a conventional one is constructed from steady-state driving conditions. A computer simulation is executed to compare the efficiencies of the both shift logics. Finally, to clarify the relationship between gear-shifting and an accelerator, the shift logic which utilizes both inputs and outputs of a powertrain as its fundamental variables is constructed, and investigated on its difference from the shift logic whose fundamental variables are outputs of a powertrain alone.

2. Model of Electric Vehicle

2. 1. Model of Powertrain of Electric Vehicle

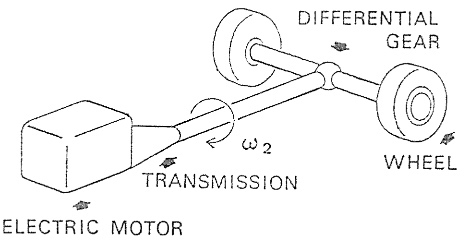
In the beginning, before obtaining a model of a powertrain of an electric vehicle, following assumptions or idealizations are set for the convenience of the fundamental consideration on the shift logic problem.

(1) Electric Motor: The prime mover is a D. C. motor controlled by an armature voltage, with a magnetic field of a permanent magnet. Such motor control circuits as a current restriction one and a regenerative one have ideally no energy losses, and their transfer functions are unity.

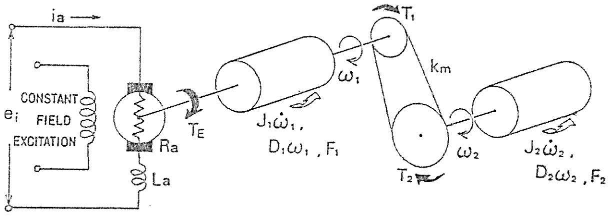
(2) Transmission: The transmission is a three-speed automatic transmission having no torque converter. Only when gear-shifting occures, a clutch is assumed to be present. This ideal transmission has also no energy loss and performs gear-shifting instantaneusly, which means that variations of a vehicle speed and a vehicle acceleration during gear-shifting are out of consideration.

(3) Driving Condition: Air resistance is assumed to be proportional to a vehicle speed on a low speed urban driving concerned in this report.

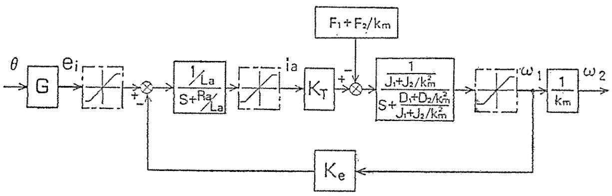
Now the model equations of the idealized electric vehicle are formulated as below, which are led by noting the transmission output RPM ω_2 and by dividing the powertrain into two sections, that is, the fore and rear sections of the power-train (Fig. 1-a, b).



(a) Transmission Output RPM ω_2



(b) Schematic Diagram of the Model



(c) Block Diagram of the Model

Fig. 1. Modeling of Electric vehicle.

$$e_i = G\theta \tag{1}$$

$$\dot{i}_a = (-R_a i_a - K_e \omega_1 + e_i)/L_a \tag{2}$$

$$T_E = K_T i_a \tag{3}$$

$$T_E = J_1 \dot{\omega}_1 + D_1 \omega_1 + F_1 + T_1 \tag{4}$$

$$T_2 = k_m T_1 \tag{5}$$

$$\omega_1 = k_m \omega_2 \tag{6}$$

$$T_2 = J_2 \dot{\omega}_2 + D_2 \omega_2 + F_2 \tag{7}$$

where, $(\dot{\ })$ means d( )/dt, and

$$J_2 = J_{2_0} + 2\pi r^2 W/(60 z^2 g) \tag{8}$$

$$D_2 = D_{2_0} + D_{2_a} \tag{9}$$

$$F_2 = F_{2_0} + (\mu \cos \alpha + \sin \alpha) Wr/z \tag{10}$$

Here the nomenclature is as follows:

(Electric Motor)

θ : Acceleration pedal opening [%]

G : Accelerator gain [V/%]

e_i : Input voltage [V]

i_a : Armature current [A]

T_E : Motor torque [kgw·m]

R_a : Armature resistance [Ω]

L_a : Armature inductance [H]

K_T : Torque coefficient [kgw·m/A]

K_e : Back EMF coefficient [kgw·m/rpm]

(Transmission Input)

ω_1 : Input speed (=Motor speed) [rpm]

T_1 : Input torque [kgw·m]

J_1 : Total inertia of the fore section of the powertrain [kgw·m·sec/rpm]

D_1 : Coefficient of viscosity in the fore section of the powertrain [kgw·m/rpm]

F_1 : Coulomb friction in the fore section of the powertrain [kgw·m]

(Transmission Output)

ω_2 : Output speed [rpm]

T_2 : Output torque [kgw·m]

J_{2_0}: Total inertia of the rear section of the powertrain [kgw·m·sec/rpm]

D_{2_0}: Coefficient of viscosity in the rear section of the powertrain [kgw·m/rpm]

D_{2_a}: Coefficient of air resistance (linearity assumption) [kgw·m/rpm]

F_{2_0}: Coulomb friction in the rear section of the powertrain [kgw·m]

(Vehicle)

W : Vehicle weight [kgw]

r : Effective radius of wheel [m]

α : Grade [deg]

g : Acceleration of gravity [m/sec^2]

k_m : Transmission gear ratio

z : Differential gear ratio

μ : Coefficient of rolling resistance

π : Ratio of circumference

Because J_2 and F_2 are influenced mainly by the vehicle weight W and the grade α respectively, J_2 and F_2 are called 'Vehicle weight' and 'Climbing resistance' respectively in this report, as long as there occures no misunderstanding. In addition, ω_2 and $\dot{\omega}_2$ are called 'Vehicle speed' and 'Vehicle acceleration' respectively from now on.

Eqs. (1)~(4) exibit the dynamics of the fore section of the powertrain and eqs. (7)~(10), the dynamics of the rear section of the powertrain. Torques and speeds of inputs and outputs of the transmission are connected to one another by the transmission gear ratio k_m (m=1, 2, 3) as shown in eqs. (5)~(6).

To protect the electric motor from failures, following three performance limits are laid.

(i) Restriction for maximum rpm

$$0 \leqq \omega_1 \leqq \omega_{1\max} \tag{11}$$

(ii) Restriction for armature current

$$-i_{a\max} \leqq i_a \leqq i_{a\max} \tag{12}$$

(iii) Restriction for input voltage

$$-e_{i\max} \leqq e_i \leqq e_{i\max} \tag{13}$$

For simplicity, $-i_{a_{max}}$ and $-e_{i_{max}}$ are used in place of $i_{a_{min}}$ and $e_{i_{min}}$ respectively in eqs. (12)～(13). The block diagram of the total vehicle system including these restrictions are depicted in Fig. 1-c.

2. 2. Energy Consumption of Powertrain

It is a significant subject how much energy automobiles consume while driving. Energy loss of an electric motor can be calculated by an approximate equation[6,7]. Using this approximation and taking into account the mechanical energy loss of a powertrain, total energy consumption V [kW] of the idealized electric vehicle can be expressed as below.

$$V(i_a,\ \omega_1)=(K_1 i_a+K_2 i_a^2+K_3\omega_1+K_4\omega_1^2+K_5)+AK_T i_a\omega_1 \tag{14}$$

Here, K_1 to K_5 are constant and A [kW/(kgw·m·rpm)] is a constant for unit exchanging. The parentheses term of the right hand side (R. H. S.) of eq. (14) means electrical energy losses of the motor, that is, 1st term: brush electrical loss etc., 2nd term: copper loss etc., 3rd and 4th terms: iron loss etc., 5th term: constant loss of battery and control circuit irrespective of current and revolutions. The last term of the R. H. S. of eq. (14) is the mechanical energy consumed while driving including such mechanical loss as friction loss.

Although it is a function of i_a and ω_1, V is affected through these two variables by driving conditions (e. g. F_2, ω_2, $\dot{\omega}_2$ etc.) and gear ratio k_m.

2. 3. Similarities to Internal Combustion Engine Vehicle

In this section some similarities between the electric vehicle model described above and a I. C. engine vehicle are listed to show that the consideration on the shift logic problem of electric vehicles in the following chapters can be expected to be applied to that of automobiles extendedly.

(1) The energy consumption of a I. C. engine is depicted on a graph as 'Fuel Map'.[10] Although it does not have such an approximate equation as eq. (14), the graph suggests it can be approximated as a quadratic form of a torque and a speed just like the electric motor.

(2) The static characteristics of the electric motor can be expressed as the following equation derived from eqs. (1)～(3):

$$T_E=K_T(G\theta-K_e\omega_1)/R_a \tag{15}$$

So motor torque T_E is a (linear) function of motor speed ω_1 and acceleration pedal opening θ. The static characteristics of a I. C. engine are expressed on a graph as 'Engine Characteristic Curves'. The graph indicates that the engine torque is also a (nonlinear) function of an engine speed and acceleration pedal opening.

(3) The dynamics of the electric motor is expressed simply as eqs. (2)～(3). The dynamics of a I. C. engine will be displayed exactly by very complex nonlinear simultaneous differencial equations, when it is fully analized. But the linearly approximated dynamics of a I. C. engine vehicle is expressed similar to the dynamics of the model in this report[4], and a flow rate of a mixture of a I. C. engine may be considered to correspond to the armature current of the electric motor.

(4) In the model, acceleration pedal opening θ and input voltage e_i are ideally connected by the constant gain G as depicted in eq. (1). The throttle opening corresponds to e_i in case of an I. C. engine, in case of which the acceleration pedal

opening and the throttle opening are also in linear relation.

By reason of the similarities listed above, modifications by noting the differences between the both vehicles will enable the adaption of the following consideration to I. C. engine vehicles.

3. Shift Logic

3. 1. Purposes of Shifting

A determinative answer to the question, 'Why automobiles must do gear-shifting?', has not been necessarily obtained, and designers of automobiles deal the problem, 'When automobiles should shift?', differently from one another or empirically sometimes. The authors believe the purposes of shifting are following three items.

(1) Extention of drivable regions of automobiles
(The drivable regions of automobiles are considered on the output torque—vehicle speed diagram in this report.)

(2) Realization of low energy-loss driving

(3) Improvement of acceleration characteristics of automobiles
(For example, the shift-down operation at passing maneuver.)

The shift logic of the electric vehicle of Ref. (6) is obtained from investigating the items (1) and (2) under steady-state driving conditions. (It uses two variables, drive force and vehicle speed, the former of which corresponds to output torque T_2 in this report.) This report examines the items (1) and (2) under acceleration or deceleration driving conditions, and asserts vehicle acceleration is one of the fundamental variables of shift logic, proposing the shift logic which is named 'Acceleration/Deceleration Driving Shift Logic'. The item (3) is related to an accelerator of an automobile, and mentioned at Chapter 5 on the relationship between gear-shifting and an accelerator.

3. 2. Acceleration/Deceleration Driving Shift Logic

The items (1) and (2) under acceleration or deceleration driving conditions, are examined at following 3. 2. 1. and 3. 2. 2. respectively.

3. 2. 1. Shift by the Conditions of Motor Performance Limits (Item (1))

Some sorts of shift which are ascribed to the performance limits of the motor i. e. eqs. (11)∼(13), are described below in order of the equations. The drivable regions of the electric vehicle are determined by these equations.

At first, on eq. (11), as clear from eq. (6), maximum motor speed $\omega_{1\max}$ is transformed into following each maximum vehicle speed $\omega_{2\max(m)}$ by each gear ratio k_m:

$$\omega_{2\max(m)}=\omega_{1\max}/k_m \quad (m=1,\ 2,\ 3) \tag{16}$$

Secondly, on eq. (12), as calculated from eqs. (3)∼(6), the motor torque generated by maximum current $i_{a\max}$ is transformed into transmission output torque T_2 satisfying following each relation by each gear, associated with vehicle speed ω_2 and vehicle acceleration $\dot{\omega}_2$:

$$T_2 = k_m(K_T i_{a\max} - F_1) - D_1 k_m^2 \omega_2 - J_1 k_m^2 \dot{\omega}_2 \quad (m=1,\ 2,\ 3) \tag{17}$$

Finally, on eq. (13), as calculated from eqs. (2)～(7), at maximum input voltage $e_{i\max}$, the three variables T_2, ω_2 and $\dot{\omega}_2$ satisfy following each relation by each gear:

$$T_2 = k_m(K_T e_{i\max}/R_a - F_1) - (K_T K_e/R_a + D_1) k_m^2 \omega_2 - J_1 k_m^2 \dot{\omega}_2 \quad (m=1,\ 2,\ 3) \tag{18}$$

Here, $\dot{i}_a$ is neglected for simplicity in the calculation of eq. (18), as it is not so important for the succeeding discussion.

From the above eqs. (16)～(18), each drivable region is formed for each gear, and by gear-shifting one drivable region is converted to another. Or gear-shifting extends the total drivable region of a vehicle by choosing out a suitable region of the three.

3. 2. 2. Shift by the Condition of Energy Consumption (Item (2))

Above drivable regions have intersections on one another. As selection of gears is allowed in the intersections, efficient driving is possible there by choosing the least energy-lossing gear position. The intersections are devided into three parts by comparing energy consumptions as follows.

At first, eqs. (3)～(6) generate next equation.

$$i_a = (J_1 k_m \dot{\omega}_2 + D_1 k_m \omega_2 + F_1 + T_2/k_m)/K_T \tag{19}$$

Substituting eqs. (19) and (6) into energy consumption equation (14), two different energy consumptions $V(T_2,\ \omega_2,\ \dot{\omega}_2;\ k_m)$ and $V(T_2,\ \omega_2,\ \dot{\omega}_2;\ k_n)$ are calculated for the two different gears k_m and k_n (assumed $m<n$) respectively for the same driving condition, $(T_2,\ \omega_2,\ \dot{\omega}_2)$. Therefore, by looking for the whole allowable domain of $(T_2,\ \omega_2,\ \dot{\omega}_2)$ satisfying

$$V(T_2,\ \omega_2,\ \dot{\omega}_2;\ k_m) \leqq V(T_2,\ \omega_2,\ \dot{\omega}_2;\ k_n), \tag{20}$$

the region can be obtained in which the energy consumption by the gear k_m is fewer than that by the gear k_n. The result after somewhat tedious calculation is as follows.

$$T_2 \geqq [-\alpha_3 + \sqrt{R(\omega_2,\ \dot{\omega}_2)}]/(2\alpha_4) \tag{21}$$

where

$$R(\omega_2,\ \dot{\omega}_2) = \alpha_3^2 + 4\alpha_4[\alpha_2 \omega_2^2 + (\alpha_1 + \overline{\alpha_1}(\dot{\omega}_2))\omega_2 + \overline{\alpha_0}(\dot{\omega}_2)]$$

$$\alpha_1 = (K_1 D_1/K_T + 2K_2 D_1 F_1/K_T^2 + K_3 + AF_1)(k_m - k_n)$$

$$\alpha_2 = (K_2 D_1^2/K_T{}^2 + K_4 + AD_1)(k_m^2 - k_n^2)$$

$$\alpha_3 = -(K_1/K_T + 2K_2 F_1/K_T{}^2)(1/k_m - 1/k_n)$$

$$\alpha_4 = -K_2/K_T{}^2 .\ (1/k_m^2 - 1/k_n^2)$$

$$\overline{\alpha_0}(\dot{\omega}_2) = (K_1/K_T + 2K_2 F_1/K_T{}^2) J_1 (k_m - k_n)\dot{\omega}_2 + K_2 J_1^2 (k_m^2 - k_n^2)/K_T{}^2 .\ \dot{\omega}_2^2$$

$$\overline{\alpha_1}(\dot{\omega}_2)=(2K_2D_1/K_T{}^2+A)J_1(k_m^2-k_n^2)\dot{\omega}_2$$

α_1 to α_4 are constants to be determined from vehicle data. $\overline{\alpha_0}(\dot{\omega}_2)$ and $\overline{\alpha_1}(\dot{\omega}_2)$ are functions of $\dot{\omega}_2$. If the driving state, $(T_2, \omega_2, \dot{\omega}_2)$, satisfies eq. (21), then the drive by the gear k_m is efficient on energy consumption. Contrary, if it does not satisfy eq. (21), then it is better to shift the gear to k_n.

Equations (21) for $m=1$, $n=2$ and for $m=2$, $n=3$, devide the intersections into three parts for three gears.

3. 2. 3. Figures of Calculation Results

Each gear-region drivable and efficient, determined by the above considerations, is depicted in Fig. 3. This figure expresses the Acceleration/Deceleration Driving Shift Logic. On the other hand, Fig. 2 shows the special case, $\dot{\omega}_2=0$ in Fig. 3, that is, the case for the steady-state driving conditions alone. This figure is named 'Steady-State Driving Shift Logic' in this report.

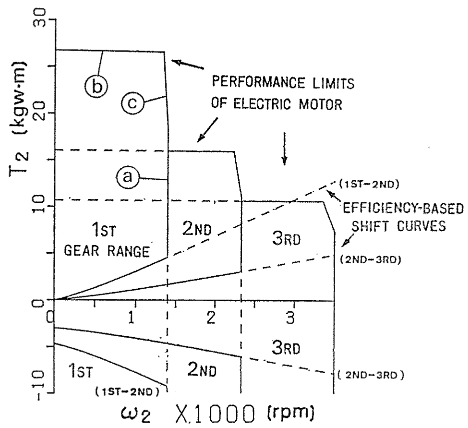


Fig. 2. Steady-state Driving Shift Logic.

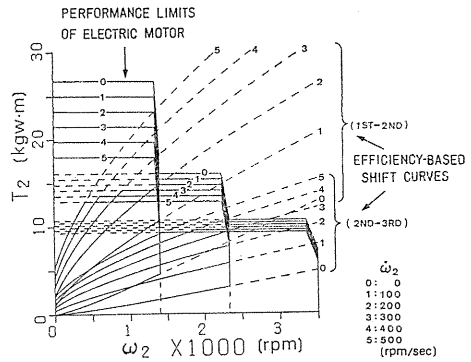


Fig. 3. Acceleration/Deceleration Driving Shift Logic.

In Fig. 2, segments a, b and c show eqs. (16), (17) and (18) respectively for the 1st gear under the steady-state conditions. Two efficiency-based shift curves in the positive torque area imply eqs. (21) for 1st-2nd gears and for 2nd-3rd gears at the condition, $\dot{\omega}_2=0$. Fig. 2 is essentially the same to the figure in Ref. (6). As for the negative torque area in Fig. 2, see Appendix 1.

In acceleration time, as seen in Fig. 3, the limits due to maximum current, eq. (17) and maximum voltage, eq. (18) on the each drivable region decrease downward as vehicle acceleration increases. This is because the motor inertia consumes a part of motor torque in acceleration, and torque T_2 usable as the transmission output torque decreases as much. On the contrary, efficiency-based shift curves increse upward as vehicle acceleration increases, so that lower speed gear region becomes shrunk. This can be realized that when a relatively low torque enables a high speed and high acceleration driving (e. g. when the vehicle accelerates on a gentle downward slope), the drive by a higher speed gear is better in efficiency.

3. 3. Shift Surfaces

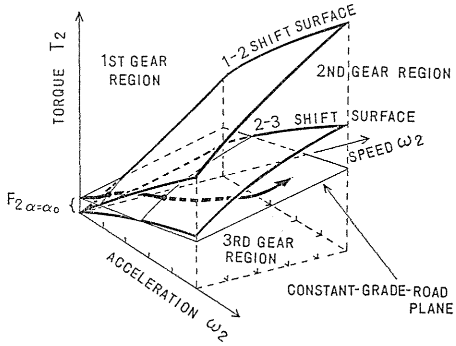


Fig. 4. Schematic Diagram of Shift Surfaces.

Fig. 3 is outlined in Fig. 4, which depicts only the surfaces derived from the efficiency-based shift curves, eq. (21), and omits the planes determined by the performance limits, eqs. (16) ~ (18). The surfaces are called 'Shift Surfaces' in this report. A plane which crosses the shift surfaces expresses the equation (7) of the driving resistances on a constant-grade road (for example, level road), $\alpha=\alpha_0$. As long as the grade is consistent, the vehicle motion is constrained on this plane, and the vehicle is required to shift at the moments it collides with the shift surfaces. If the grade changes, then the plane moves parallel. But, even if the plane moves, the shift surfaces themselves are unchanged.

Acceleration/Deceleration Driving Shift Logic, Fig. 3 (or Fig. 4) needs sensing output torque, vehicle speed and vehicle acceleration, all of which are the quantities on vehicle motions. But the shift surfaces remain uninfluenced for any grade and any acceleration, that is to say, whatever the conditions of the vehicle may be. And the surfaces are decided in advance only by the performances of the prime mover.

As already described, shift surfaces are in the space of three variables, T_2, ω_2 and $\dot{\omega}_2$. Eq. (7) shows that the climbing resistance F_2 and these three valiables are linearly dependent. Therefore, in place of $\dot{\omega}_2$, F_2 can construct, cooperating with T_2 and ω_2, shift surface in T_2-ω_2-F_2 space. Though this report asserts it is an important variable of shift logic, the vehicle acceleration $\dot{\omega}_2$ is not an absolutely exclusive variable having no alternative, but F_2 can be an alternative. A figure of shift surfaces in T_2-ω_2-F_2 space can be also depicted corresponding to Fig. 3, but is omitted here.

The similar discussion can be made in i_a-ω_2-$\dot{\omega}_2$ space, namely, T_2 can be replaced with i_a. This is because eq. (19) indicates that i_a and other three variables are linearly dependent. The detailed calculation of shift surfaces in i_a-ω_2-$\dot{\omega}_2$ space is in the same way as that of 'e_i-ω_2-$\dot{\omega}_2$ Shift Diagram' which is to be stated later. (See Chadter 5 and Appendix 2.)

A current is a practical, easily measurable signal, so the electric vehicle of Ref. (6) utilizes in practice the two signals, current and vehicle speed for the shift logic. The current is a substitute for the drive force which corresponds to output torque T_2 in this report. But vehicle acceleration is not considered as a fundamental variable of shift logic.

The shift logic of a I. C. engine vehicle is constructed from two variables, generally, acceleration pedal opening and vehicle speed. The acceleration pedal opening is a substitute for output torque. In this report, to investigate the item (3) and the relationship between gear-shifting and an accelerator, output torque T_2 is replaced with input voltage e_i, which corresponds to the throttle opening in case of a I. C. engine vehicle. And e_i-ω_2-$\dot{\omega}_2$ Shift Diagram is obtained in Chapter 5.

4. Simulation of Example

Computer simulation for the efficiency comparison between
(A) Steady-State Driving Shift Logic (Fig. 2) and
(B) Acceleration/Deceleration Driving Shift Logic (Fig. 3),
was executed to see how much energy consumptions of the identical vehicle differ owing to the both shift logics. Fig. 5 shows the simulation result which is led under the same condition of a level road and along the same driving pattern (so called, '10-Mode Driving Pattern'). Neither disturbances nor parameter deviations are assumed to occur. But it is only the shift points that differ as the condition for the comparison. The data used are in Table 1. They are typical values approvable for the electric vehicle of the gross weight $W=1,200$ [kgw]. (The figures in this report are based on the data.)

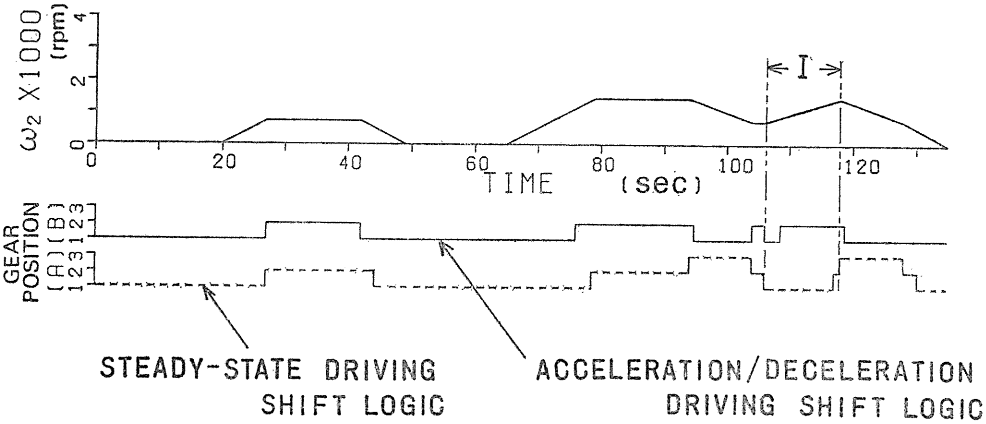


Fig. 5. Computer Simulation for Efficiency Comparison between the Both Shift Logics.

Table 1. Date of System Parameters.

J_1	7.0×10^{-4}	K_T	0.0135	k_3	2.0
J_2	0.072	K_e	0.015	K_1	6.4×10^{-3}
D_1	5.9×10^{-6}	$e_{i\max}$	115.0	K_2	3.6×10^{-5}
D_2	7.4×10^{-4}	$i_{a\max}$	400.0	K_3	5.0×10^{-7}
F_1	0.05	$\omega_{1\max}$	7.0×10^{3}	K_4	7.9×10^{-9}
R_a	0.036	k_1	5.0	K_5	0.42
L_a	4.3×10^{-4}	k_2	3.0	A	1.026×10^{-3}

The total energy consumptions are 332.6 [kJ] for the case (A), and 313.4 [kJ] for the case (B). The percentage of improvement by (B) is 5.8%.

As seen in Fig. 5, in acceleration time, the vehicle shifts somewhat earlier by

(B) than by (A), because the shift surfaces in Fig. 4 increase upward as the acceleration increases. Improvement by (B) between the restricted interval I in acceleration time in Fig. 5, is 2.0 %. Energy-saving by altering shift points will not be negligible if such accelerations are repeated time after time.

In deceleration time, the both shift logics enable energy restoration through regenerative braking, but the difference of the gear positions causes the difference of the quantities of the energy lost without restoration. In Fig. 5, Shift Logic (A) is apt to shift to higher speed gears due to the negative low torque in deceleration time, while Shift Logic (B) puts the motor to rotate in high speed by lower speed gears to increase the energy recovery.

This simulation shows the effectiveness of Shift Logic (B) especially for energy recovery in deceleration time. On the contrary, the improvement in acceleration time is smaller than expected. The authors believe, however, that the percentage 2.0 % is not meaningless; because, for example, if the fuel consumption rate of a gasoline compact car were improved from 13 to 14 [km/l] in 10-Mode Driving Pattern only by the alternation of shift points, then it would be an astonishing matter, in case of which the improved percentage is 7.6 %.

In this simulation the vehicle runs on a level road, but Acceleration/Deceleration Driving Shift Logic is usable for any grade roads, always maintaining the highest effectiveness for energy consumption among the three gears.

5. Relationrhip between Gear-Shifting and Accelerator

Acceleration/Deceleration Driving Shift Logic (Fig. 3 or 4) gives an impression that gear-shifting is independent of an accelerator. But construction of the shift logic whose variables are e_i, ω_2 and $\dot{\omega}_2$ (interchanging T_2 with e_i) clarifies that gear-shifting is closely related to an accelerator. This shift logic is called 'e_i-ω_2-$\dot{\omega}_2$ Shift Diagram' in this report. Using this diagram, the relationship between gear-shifting and an accelerator is considered below.

The calculation of e_i-ω_2-$\dot{\omega}_2$ Shift Diagram follows basically that of Chapter 3. But as the variables are chosen from both the input and output of the transmission, it results that troublesome simultaneous equations must be solved. Details are summarized in Appendix 2 and the result is shown in Fig. 6.

Fig. 6 contains two shift diagrams, (a) and (b) in a pair, resembling in appea rence the pair of both the shift-up and shift-down diagrams of a I. C. engine vehicle. In acceleration time, the shift diagram for lower speed gears (Fig. 6-a) is applied to the gear before shifting, and the shift diagram for higher speed gears (Fig. 6-b) is applied to the gear after shifting. In deceleration time, they are reversed. Both figures, (a) and (b) show that efficiency-based shift curves gradually change as vehicle acceleration increases.

Fig. 7 gives an outline of how the vehicle moves and performs shifting on e_i-ω_2-$\dot{\omega}_2$ Shift Diagram. The efficiency-based shift curves for acceleration $\dot{\omega}_2=a_0$ and deceleration $\dot{\omega}_2=-b_0$ are brought from Fig. 6 (a_0 and b_0 are some positive constants), showing the shift points on the driving condition that constant vehicle acceleration on a level road is continued from the point a to the point f in the figure then, likewise, constant deceleration is succeeded to the point k. In acceleration time, between the points b and c and between the points d and e, the vehicle

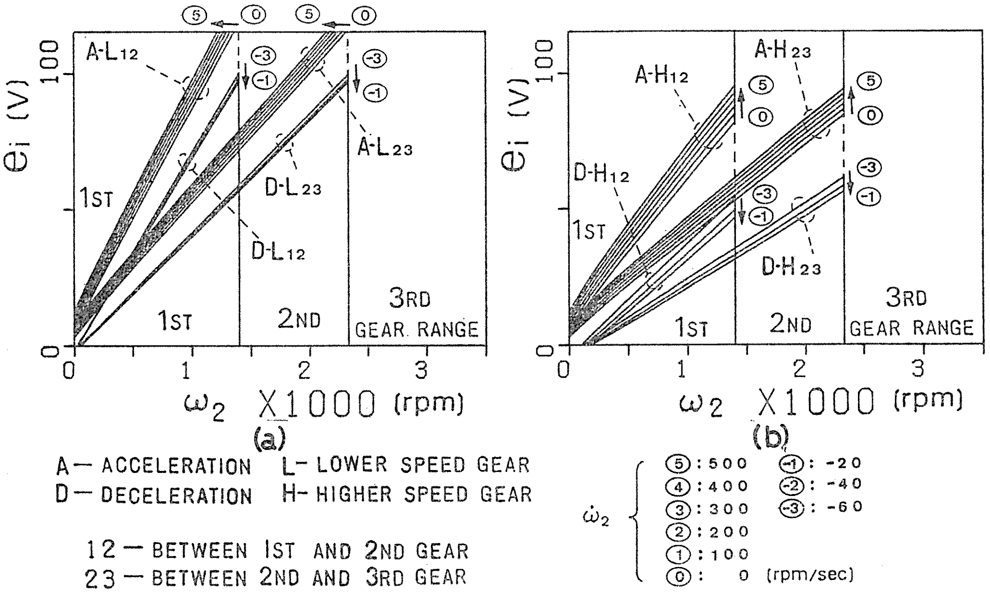


Fig. 6. Shift Diagrams of Acceleration/Deceleration Driving Shift Logic.
(a) Shift Diagram for Lower Speed Gears.
(b) Shift Diagram for Higher Speed Gears.

shifts from 1st to 2nd gear and from 2nd to 3rd gear respectively, and at the same time input voltage is controlled adequately, which produces the unviolated acceleration after shifting. In deceleration time, the similar statement is concluded.

Fig. 6 and 7 indicate that control of input voltage e_i is needed to maintain the commanded vehicle acceleration after gear-shifting, simultaneously pursuing efficiency (namely, economy requirment). The command of vehicle acceleration is put in by a driver through acceleration pedal opening θ. For the constant gain G of an accelerator it is impossible to practice the control of e_i, then an actively controlling accelerator is necessary, which decides the required input voltage after shifting. The role of the accelerator is recognized as the device that harmonizes the connection between a driver's acceleration (or speed) command and the system states changes of the power-train caused by gear-shifting.

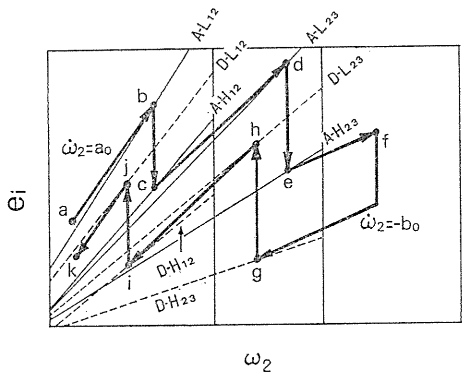


Fig. 7. Trajectory of Constant Acceleration and Constant Deceleration Driving.

Conventional shift logics for both electric and I. C. engine vehicles set themselves the hysteresis, which avoids a certain unstability of shifting, namely 'hunting', preparing two shift diagrams for shift-up and shift-down at the sacrifice of efficiency. This phenomenon is closely related to the difference of the dynamics of the powertrain arisen from the difference of the gear positions before and after shifting (see Fig. 1-c), and to the driver's response of stepping the acceleration pedal. The authors believe that the phenomenon can be overcome by controlling gear-shifting and the accelerator at the same time adequately as shown in Fig. 7, and that the sacrifice of efficiency becomes unnecessary.

The shift logic containing vehicle acceleration as one of its fundamental variables, and the Actively Controlling Accelerator mentioned above enable vehicle driving to satisfy the items (2) and (3) in Section 3.1 at a time.

6. Conclusion

6. 1. discussion

The conventional shift logic which infers shift points by the use of output force (or torque) and output speed of a transmission, seems adequate from the viewpoint that multiplication of the both variables implies the output power. But, as becomes clear from Chapter 3, vehicle's driving states are not completely determined by these two variables alone, and one more variable (vehicle acceleration, for example) is necessary. All the information necessary and sufficient to identify the vehicle's driving states enables the best selection of the gear positions at any time, as seen in Fig. 4. It is noticed in this case, that the information of the prime mover is not required.

As becomes clear from Chapter 5, the shift logic which utilizes both input and output signals of a powertrain like e_i-ω_2-$\dot{\omega}_2$ Shift Diagram, requires from the both sides the number of variables necessary and sufficient to identify the operating states of the powertrain completely (more precisely, to prepare the conditions for efficiency comparison and to determine the energy consumption, eq. (14) completely). Generally it becomes larger than the number of the variables of the shift logic which utilizes only the output states like Fig. 4. It is because the former shift logic has to take into consideration the dynamics of the powertrain (including the prime mover), whose freedom is large even if the output states of the vehicle are fixed. In the model of this report, the dynamics of the electric motor treats only the terms, $\dot{i}_a$ (change rate of current) and $\dot{\omega}_1$ (motor acceleration). In fig. 6, the terms of $\ddot{\omega}_2$ and $\dot{F}_2$, which are two of the variables derived from the calculation of $\dot{i}_a$, are neglected for simplicity, as they are small in ordinary driving conditions. And only $\dot{\omega}_2$ (vehicle acceleration) is added to the shift logic variables. If the effect of $\dot{i}_a$ had to be considered precisely, e_i-ω_2-$\dot{\omega}_2$ Shift Diagram would be incomplete and must take $\dot{i}_a$ (otherwise, $\ddot{\omega}_2$ and $\dot{F}_2$) into consideration besides the three variables.

The difference of the gear positions impels a prime mover to operate in the different states, even though the output states of a vehicle do not change before and after gear-shifting, so gear-shifting is closely related to an accelerator. To satisfy both economy requirements and acceleration command, the Actively Controlling Accelerator is necessary, as e_i-ω_2-$\dot{\omega}_2$ Shift Diagram explicitly shows.

6. 2. conclusion

In this report, a fundamental consideration on the shift logic for electric vehicles is performed, attempting to clarify the essence of shift logic for automobiles. As a result, the shift logic which contains vehicle acceleration as one of its fundamental variables, is proposed, while a conventional one is constructed under steady-state driving conditions. This report shows that the shift logic proposed in this report enables the improvement of a transmission as seen in Chapter 4, and that there is the possibility of its application to I. C. engine vehicles.

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Appendixes

Appendix 1

Fig. 2 partly shows each gear-region also in the negative torque area. The region is decided by the same calculation as the one in the positive torque area. The efficiency-based shift curves in the negative torque area are calculated by the following equation:

$$T_2 \leqq [-\alpha_3 - \sqrt{R(\omega_2,\ \dot{\omega}_2)}\,]/(2\alpha_4) \tag{A1}$$

An automobile ascends and decends hills, and even if the acceleration pedal is kept on stepping on a level road, it decelerates when the resistances such as air and rolling resistances overcome the drive force. So, likewise to the above comment on Fig. 2, the remaining three portions not appearing in Fig. 3 (namely, the portions where (i) $\omega_2 > 0$, $\dot{\omega}_2 > 0$ and $T_2 < 0$, (ii) $\omega_2 > 0$, $\dot{\omega}_2 < 0$ and $T_2 > 0$, (iii) $\omega_2 > 0$, $\dot{\omega}_2 < 0$ and $T_2 < 0$) can also have shift logic. The calculation of each gear-region in these portions uses eqs. (21) and (A1) for efficiency-based shift curves. The figure is omitted here because of the complexity.

Appendix 2

Derivation of e_i-ω_2-$\dot{\omega}_2$ Shift Diagram is summarized below.

At first, i_{am}, e_{im} and V_m are defined as armature current, input voltage and energy consumption respectively on the driving condition (ω_2, $\dot{\omega}_2$, F_2) at the gear position k_m. They satisfy following three relations.

$$i_{am} = [(J_1 k_m + J_2/k_m)\dot{\omega}_2 + (D_1 k_m + D_2/k_m)\omega_2 + F_1 + F_2/k_m]/K_T \tag{A2}$$

$$(-R_a i_{am} - K_e k_m \omega_2 + e_{im})/L_a = (D_1 k_m + D_2/k_m)/K_T \cdot \dot{\omega}_2 \tag{A3}$$

$$V_m = (K_1 i_{am} + K_2 i_{am}^2 + K_3 k_m \omega_2 + K_4 K_m^2 \omega_2^2 + K_5) + A K_T i_{am} k_m \omega_2 \tag{A4}$$

In eq. (A3), the terms $\ddot{\omega}_2$ (change rate of acceleration) and $\dot{F}_2$ (change rate of climbing resistance, caused by the grade change) are neglected here, because they are small in ordinary driving conditions. The same discussion below in this Appendix is possible including them, though the calculation becomes more intricated.

Secondly, i_{an}, e_{in} and V_n are similarly defind as the three variables on the same driving condition supposed above, but at the gear position k_n (assumed $m < n$). And three equations like (A2)$\sim$(A4) follows.

Now, assume the gear is changed from k_m to k_n. Both the above six equations and an inequality equation

$$V_m \leqq V_n \tag{A5}$$

form simultaneous equations. The solution in terms of e_{im}, ω_2 and $\dot{\omega}_2$ expresses the region where energy consumption by the gear k_m is fewer than that by the gear k_n for the transmission states before shifting (Fig. 6-a). Exchanging eq. (A5) for the following one:

$$V_m \geqq V_n \tag{A6}$$

and solving renewed simultaneous equations in terms of e_{in}, ω_2 and $\dot{\omega}_2$, we can obtain the region where energy consumption by the gear k_n is fewer than that by the gear k_m for the transmission states after shifting (Fig. 6-b).

Fig. 6-a and -b is a pair of Shift Diagrams. Necessity of e_{im} and e_{in} for the calculation, results from the difference of the operating states of the powertrain owing to the difference of the gear positions.

In Fig. 7, the trajectory of a constant-acceleration driving on a constant-grade road is a straight line. This is because the following equation holds from eqs. (2) and (A2):

$$e_i=[R_aD_1/K_T+K_e)k_m+R_aD_2/(K_Tk_m)]\omega_2+[(R_aJ_1+L_aD_1)k_m/K_T + (R_aJ_2+L_aD_2)/(K_Tk_m)]\dot{\omega}_2+R_a(F_1+F_2/k_m)/K_T \quad (A7)$$

By the same reason in the section 3. 3, e_i-ω_2-$\dot{\omega}_2$ Shift Diagram can be transformed into 'e_i-ω_2-F_2 Shift Diagram' by replacing $\dot{\omega}_2$ with F_2.

Furthermore, 'i_a-ω_2-$\dot{\omega}_2$ Shift Diagram' can be constructed in the same manner. (The calculation is a little simpler than that of e_i-ω_2-$\dot{\omega}_2$ Shift Diagram.) In this case, currents i_{am}, i_{an} are necessary as well input voltages e_{im}, e_{in} are in e_i-ω_2-$\dot{\omega}_2$ Shift Diagram.