Special Feature: Drivetrain and Braking Technology

Research Report

Robust Design Method for Automatic Transmission Clutch Control System

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■**ABSTRACT**‖ Due to recent requirements for fuel economy and drivability improvement, increased responsiveness and control accuracy are required in automatic transmission control systems. However, there are characteristic variations in every transmission, and nonlinearities that make control difficult. For this reason, it is important that the system is resistant to variations and nonlinearities; that is, a robust control system is required.

In this study, robust design methods are proposed, which include feedback and feedforward control methods that are generally applied to the clutch control systems of automatic transmissions. In the feedback control, we propose a loop-shaping method that aims to optimize a complementary sensitivity function of the control system so that the stability and regulation performance can be improved as much as possible. In the feedforward control, robust target patterns are first generated by our proposed "hierarchical and inverse directional design method". Once the target pattern is obtained, the control input can be systematically calculated from the target pattern using an inverse calculation. Each method is verified, and the validity is confirmed.

1. Introduction

Due to recent requirements for fuel economy and drivability improvement, increased responsiveness and control accuracy are needed in automatic transmission control systems. Because clutches are generally applied in control systems to regulate or change the rotational speed of gears within a fixed period, the performance of clutch controls has a great influence on fuel economy and shift shock.

However, there are several problems that make clutch control difficult. One is a change in driving conditions, such as vehicle speed, engine load, and oil temperature. In different conditions, the output responses of the system are often different because of the change in transfer characteristics, even if the same control input is applied.

Furthermore, there are characteristic variations in every transmission and nonlinearities. For example, the hydraulic system has characteristic variations of control valves in the manufacturing and clearance variations of clutch pack which cause shift shock in initial clutch engagement. An example of nonlinearities is response speed change of oil pressure due to the difference of

the sensitiveness between flow control mode where the clutch pack clearance is removed and oil pressure control mode where the clutch cramp force is adjusted in the hydraulic system. These variations cause the control system to become unstable, making calibration of the control inputs difficult. For this reason, it is important that the system is resistant to changes in driving conditions and characteristic variations; that is, a robust control system is required.

Previous studies on clutch control have been of two types. The first type is concerned with feedback control. Hiramatsu et al.[1] performed a stability analysis of a PI-based feedback control system, and determined a range of constants that ensure stability. Lee et al.[2] employed not only sliding mode control to ensure good tracking performance, but also ensured robustness against uncertainties such as unmodeled dynamics and disturbances. It is not clear, however, how a good damping performance (convergence) is ensured in Ref. (1). In Ref. (2), it is not sufficiently elucidated how the parameters for a controller should be determined if a frequency band, where unmodeled dynamics exist, is close to that of control.

The second type of studies involves feedforward

control, by which the control input is adjusted based on variations estimated using a model in an onboard computer.[3–5] However, the main adjustment is only an offset of the predetermined input value. Accordingly, the generation of an input value that determines the basic transient characteristics of the shift is still a problem.

In the present study, robust design methods are proposed, which include feedback and feedforward control methods that are generally applied to the clutch control systems of automatic transmissions. First, the feedback control, which uses information from sensors, provides a low-sensitivity response from a disturbance or a variation to the system output. One of the design methods is widely known as "formulas for mixed sensitivity problems".[6, 7] However, it was found that these do not offer sufficient stability. Therefore, we adopt the method described below, namely, loop shaping that aims to optimize a complementary sensitivity function of the control system so that the stability and the regulation performance can be improved as much as possible.

Secondly, in the feedforward control, robustness of the system should be achieved by the design of the control input pattern so that the desired output response can be maintained even though characteristic variations exist in the plant. In our proposed "hierarchical and inverse directional design method", the first step involves the generation of a robust target pattern, such that the system output pattern, which is generated to be equal to the target pattern, is not significantly affected by the characteristic variations of the system. Once the target pattern is obtained, the control input can be systematically calculated from the target input by an inverse calculation.

In Section 2, two typical clutch control systems are explained. In Section 3, our proposed robust design methods are described, and in Section 4, they are validated for the control systems of Section 2.

2. Systems and Requirements

A torque converter lock-up clutch slip control system and a shift control system are described as typical examples of feedback and feedforward controls.

2. 1 Lock-up Clutch Slip Control System

The lock-up clutch improves the efficiency of a transmission by providing a direct mechanical connection between a pump and a turbine of a torque converter. Unfortunately, it must be disengaged to prevent deterioration in the drivability of the vehicle, because engine torque fluctuations are transmitted to the driveline at low and medium vehicle speeds. A slip control system enhances the power transmission efficiency and improves fuel economy by engaging the clutch with a slight slip at those speeds where the clutch is disengaged. The slip speed needs to be accurately maintained to the target one so that the contradictory requirements can be satisfied at the same time. **Figure 1** shows the overall configuration of the control system. An electronic control unit (ECU) detects the slip speed of the clutch and also calculates and the outputs a command signal to a hydraulic control unit, so that the slip speed can be maintained at the target value. In response to a received command, the hydraulic control unit regulates the oil pressure so that the clutch applies the required pressure. The mechanical configuration is shown in Ref. (8).

Next, the characteristic variations of the control system in Fig. 1 are described. **Figure 2**(a) shows the response patterns for *Nslip* (Output signal), where a step signal is input as *u* (Input signal) in Fig. 1. According to the change in engine load, it is clear that the pattern for the slip speed (*Nslip*) varies significantly.

Transfer functions from the input *u* to the output *Nslip* are calculated, where the conditions of the ATF (automatic transmission fluid), oil temperature, throttle position, and turbine speed (input speed of the transmission) are changed so that the characteristic variations of the system can be investigated
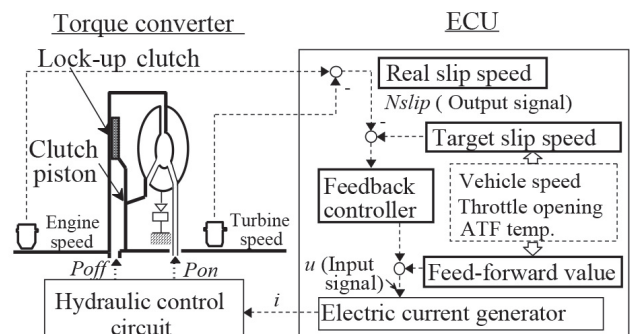

Fig. 1 Slip control system.

quantitatively. Therefore, the model shown in Eq. (1) is applied, and the model coefficients are determined by an experimental identification method.[9]
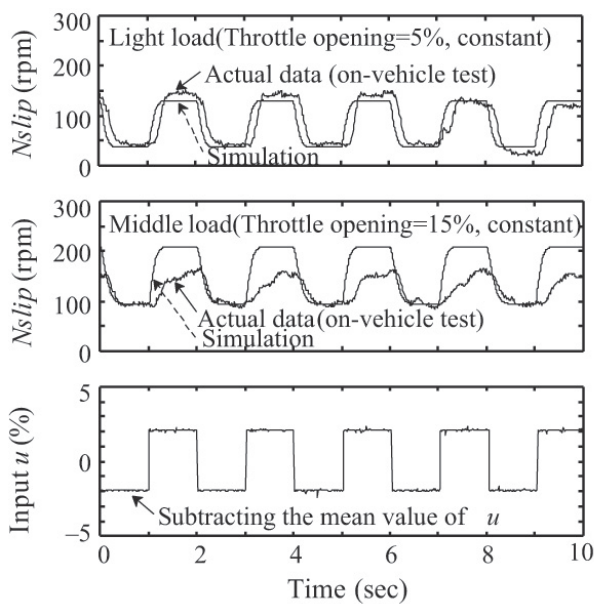
$$A(q) \cdot Nslip(t) = B(q) \cdot u(t - nk) + C(q) \cdot e(t), \quad (1)$$

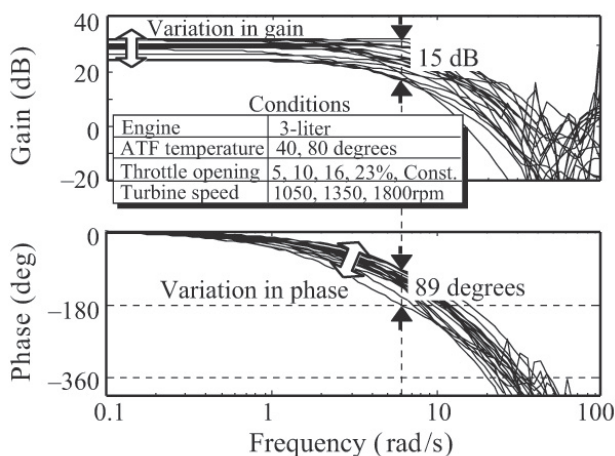where e is the prediction error, q^{-1} is the delay operator, and nk is the dead time.

$$A(q) = 1 + \sum_{i=1}^{na} a_i \cdot q^{-i}, B(q) = \sum_{i=1}^{nb} b_i \cdot q^{-i}, C(q) = 1 + \sum_{i=1}^{nc} c_i \cdot q^{-i}$$

In typical order,
$na = 5$, $nb = 5$, $nc = 2$, $nk = 3$

(a) Relation between model and actual unit

(b) Bode diagrams of the transfer functions Eq. (1).

Fig. 2   Characteristic variations.

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Bode diagrams for the transfer functions in Eq. (1) from the input to the output, which are determined for all conditions, are shown in Fig. 2(b). It is considered that the variations shown in the figure affect the stability of the feedback control system because the variations of the gain phase are large in the frequency range where the phase is below 180 degrees.

From the above, the requirements for the lock-up clutch slip system are as follows:
[Requirements]
(i) High regulation performance of the slip speed under all driving conditions (quick response)
(ii) Stability under all driving conditions (robust stability)
(iii) Convergence to the target (good damping performance).

Requirement (iii) is for good drivability. If an undamped oscillation is transmitted to the vehicle body, the drivability declines.

2. 2 Shift Control System

As another example, we consider a 2 to 3 upshift in a shift control system and explain the shift action. The 2nd clutch is released, and the 3rd clutch is engaged. Therefore the transmitted pass of engine power is changed and a pair of meshing gears is switched to another. The clutch release or engagement is controlled by each clutch hydraulic system. **Figure 3** shows a time series of the oil pressure, torque and revolution speed of each part during the shift.

In the figure, lines of nominal and disturbed conditions are each shown by solid and dashed lines. The clutch transmitted torque is switched from 2nd clutch to 3rd following an increase of the oil pressure of the engaged clutch. Here the oil pressure of the released clutch is controlled to zero at the moment when all torque is switched to the 3rd clutch (end of torque phase). If the timing of the clutch release is too early the engine revolution speed increases quickly, and in the case of late timing a shock occurs. At this point the turbine revolution speed is still equal to 2nd synchronizing speed ω_{o2} (Fig. 3). Therefore the oil pressure of the engaged clutch is increased further so that the revolution is decreased to 3rd synchronizing speed ω_{o3} (inertia phase). The inertia phase (shift) is completed at the time when the turbine revolution speed is equal to the 3rd synchronizing speed.

The dashed lines show the patterns where hydraulic

variation exists. Shift shock is found in the output torque graph. Shock is also found in the case where variations occur in the clutch friction coefficient because these variations lead to disturbances in the clutch transfer torque.

The requirements for the shift control system are expressed as the objective functions in Eqs. (2) to (4). Objective functions I and II in Eqs. (2) and (3) are for suppressing the shift shock and time, respectively. These are expressed in terms of torque change factors A, B and shift time factors C, D, which are typical characteristic factors of the output torque and turbine revolution speed patterns, as shown in **Fig. 4**. Smaller values of the objective functions are better, that is, smaller output torque in the inertia phase and smaller shift time.

Objective function III in Eq. (4) is the heat load. The clutch converts energy from differences in rotation speed into heat energy. If the energy is too large, the clutch breaks. Therefore, again, smaller values of the objective function are better. Here, the transfer torque

T_{c3}, whose pattern determines objective function III, also determines the output torque pattern. This means that objective function III is related to objective functions I and II and can be expressed in terms of factors A, B, C, and D.

[Requirements]

Objective function I: Shift shock $J_s = A + B$ (2)

Objective function II: Shift time $J_t = C + D$ (3)

Objective function III: Clutch heat load

$$J_q = Q = \int_0^{tc} T_{c3} \cdot \Delta\omega_{c3} dt \qquad (4a)$$

where

$$tc = C + D \qquad (4b)$$

$$\Delta\omega_{c3} = \omega_t - \omega_{o3} = (\omega_{o3} - \omega_{o2}) / D \cdot t + \omega_{o2} - \omega_{o3} \qquad (4c)$$

$\Delta\omega_{c3}$: Slip speed of engaged clutch

$$T_{c3} = \mu_3 (\Delta\omega_{c3}) \cdot A_3 \cdot R_3 \cdot P_{c3} \qquad (4d)$$

A_3: Area of engaged clutch

R_3: Representative radius of engaged clutch

3. Proposed Robust Design Method

3. 1 Robust Design of Feedback Control System

In this section a feedback control system is studied. H_∞ control design which enables the achievement of
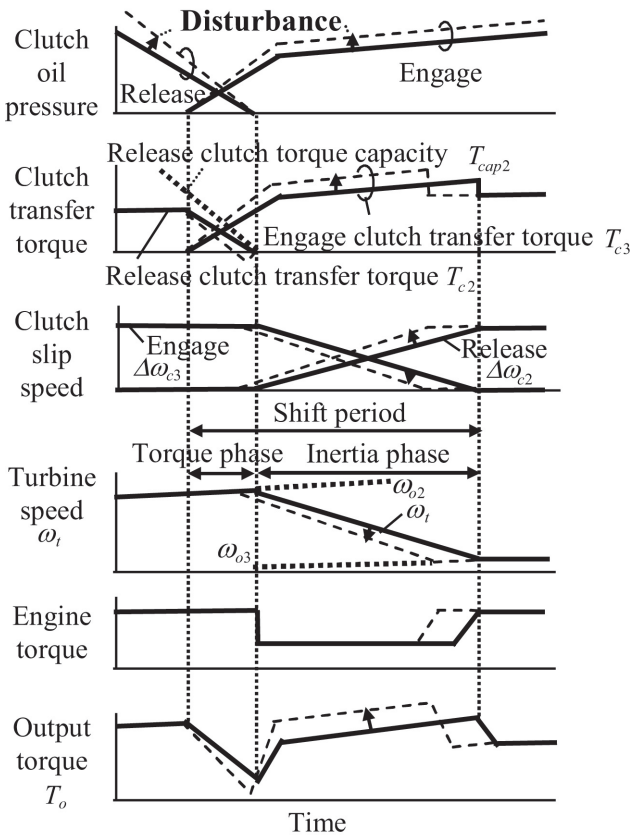

Fig. 3   Time series of 2 to 3 up-shift.

Fig. 4   Requirements for AT shift control.

loop shaping, which is shaping a frequency response of a closed loop system, is applied. This allows us to propose a method for ensuring the robust stability of the characteristic variations and enhancing the response performance to the maximum degree possible. First, we describe so called mixed-sensitivity problem formulation,[7] and the issues to be solved, and then move on to our proposed method.

The lock-up clutch slip system, which is given by Eq. (1) of Section 2.1, can also be expressed by Eq. (5). The nominal model is characteristic in typical driving conditions, and all of the other conditions are expressed by perturbations , $\Delta(s)$.

$$P(s) = (1 + \Delta(s)) \cdot P_0(s) \tag{5}$$

where

$P_0(s)$：Nominal model, s：Laplace operator
$P(s)$：Model in disturbed condition

First, the mixed-sensitivity problem formulation,[7] is applied so that the controller satisfies the above requirements. The configuration of the feedback control system is shown in **Fig. 5**. In order to satisfy the requirement (i) shown in the Section 2.1, the controller is needed to contain an integrator so that a deviation from the target value is not caused. Therefore, a sensitivity function $S_0(s)$ from the target slip speed r to the tracking error e between the target and the plant output is determined to satisfy the relationship defined by Eq. (6), where the weighting function $\gamma W_1(s)$ acts as a first order integrator.

$$\left| S_0(j\omega) \right| < \left| W_1(j\omega)^{-1}/\gamma \right| \quad \forall \omega \tag{6}$$

where

$$S_0(s) \equiv (1 + P_0(s) \cdot C(s))^{-1}$$

Fig. 5   Feedback control system.

On the other hand, requirements (ii) and (iii) shown in the section 2.1 are satisfied as follows. That is, a complementary sensitivity function $T_0(s)$ from target slip speed r to the slip speed is determined to satisfy the relationship defined by Eq. (8) by using a weighting function $W_2(s)$, which satisfies Eq. (7) for all characteristic variations.

$$\left| \Delta(j\omega) \right| \leq \left| W_2(j\omega) \right| \quad \forall \omega \tag{7}$$

$$\left| T_0(j\omega) \right| < \left| W_2(j\omega)^{-1} \right| \quad \forall \omega \tag{8}$$

where

$$T_0(s) \equiv P_0(s) \cdot C(s)(1 + P_0(s) \cdot C(s))^{-1}$$

These are the formulations for the mixed sensitivity problem.

There is, however, a problem with the above formulations. That is, Eq. (7) does not ensure convergence to the target (good damping performance) under disturbed conditions. In practice, the larger the value of γ in Eq. (6) for quick response, the greater the output oscillation becomes. This is because peak characteristics appear in the transfer function $T(s)$ of Eq. (9) under disturbed conditions (**Fig. 6**).

$$Nslip = T(s) \cdot r \tag{9}$$

where

$$T(s) \equiv P(s) \cdot C(s)(1 + P(s) \cdot C(s))^{-1} \tag{10}$$

Therefore, we propose a new method for a loop-shaping design where the peak characteristics in Fig. 6 are also considered (only the peak is suppressed). Specifically, the weighting function $W_2(j\omega)$ in Eq. (8) is modified so that only the peak in the figure is suppressed.

3.2 Robust Design of Feedforward Control System

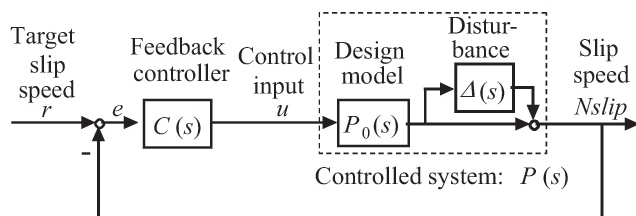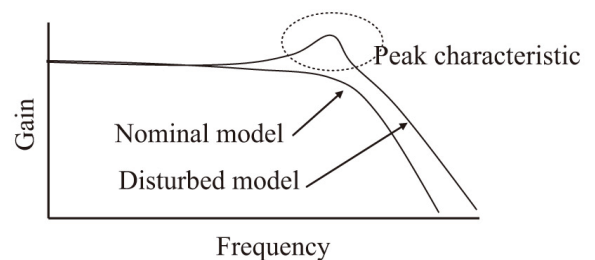In this section, a proposed design method for a

Fig. 6   Frequency responses of co-sensitivity functions $T(s)$.

feedforward control system is explained using **Fig. 7**. This figure shows the basic design flow. Step 1 of the design is the preparation of robust target patterns, such as the output torque. However, patterns that ensure robustness cannot be determined in the early stages of the design. Therefore, all possible patterns are considered.

Next, the most robust target pattern is determined in Steps 2 to 4. However, this cannot be done with just the target patterns. Therefore, target oil pressures corresponding to the target output patterns are initially generated with inverse calculations in Step 2.

In Step 3, the characteristic variations are added to the plant model. Simulations are performed with the above target oil pressures, and the result that is most robust and satisfies the requirements is determined. The target output pattern corresponding to the selected results is the optimal pattern (Step 4).

Finally, in Step 5, a control signal is generated with an inverse calculation based on the target output pattern of Step 4.

The abovementioned steps represent the basic flow of the proposed method. However, there are countless possible target patterns in Step 1, and it is difficult to carry out the design flow. Therefore, a method based on the Taguchi Method[10] is applied. With this method, it is sufficient if only 18 target patterns are considered in Step 1. The above design process should be applied to the 18 patterns in the same way. However, it is not necessary to apply it to all patterns, that is, with every possible combination of values for A, B, C, and D
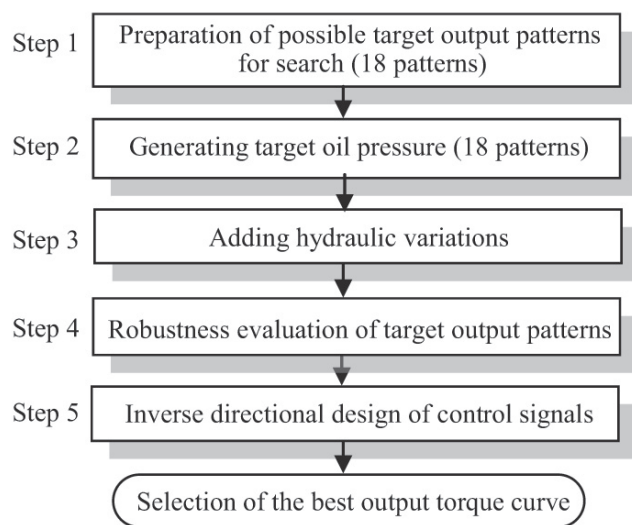
satisfying $A_1 \le A \le A_3$, $B_1 \le B \le B_3$, $C_1 \le C \le C_3$, $D_1 \le D \le D_3$ (where A_1, $A_3 \cdots$ are defined in **Fig. 13**.), because the general results can be obtained by estimating from the results obtained for the 18 target patterns. Furthermore, the method is based on the signal-to-noise (S/N) ratio,[10] so efficient quantitative investigations can be performed.

4. Application Example

4. 1 Application Example of Clutch Slip Control System

The design method described in Section 3. 1 is next applied to a slip control system. First, the problem with the conventional method is shown with three typical linear models in **Fig. 8**. Model A in the figure is the design model, model B is for the high-load condition, and model C has a typical characteristic under low loads and with age-related deterioration of the ATF and clutch. Models B and C are assumed to be models for the disturbed condition. **Figure 9** shows $\Delta(s)$ of Eq. (5) and the decided weighting function $W_2(s)$ (solid line). A simulation result for the step response and the frequency response (Eq. (9)) are shown in **Figs. 10**(a) and (b), where the controller is designed based on the H_∞ control theory,[6,7] which uses $W_2(s)$ in Fig. 9. The responses of models B and C show oscillations that result in bad convergence to the target. That is, it is not possible to satisfy all the requirements with only the mixed sensitivity problem formulation. The reason
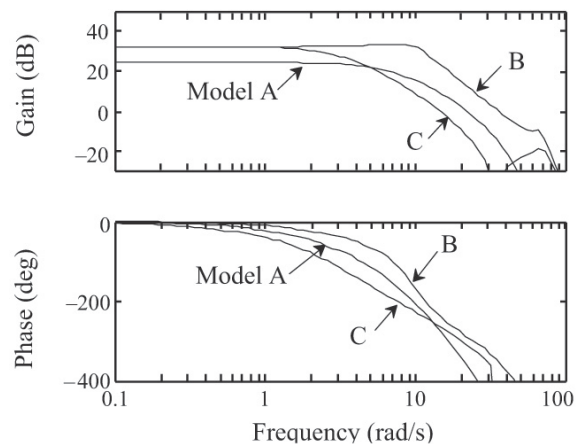

Fig. 7 Flow of robust design of feedforward control system.

Fig. 8 Characteristic variation of three linear models.

is that the formulation of Eq. (7) does not ensure convergence to the target (good damping performance) under disturbed conditions, as previously mentioned.

Therefore, the second step in the design is the suppression of the peak characteristic. A second-order transfer function is added to $W_2(s)$. For example, $\delta\Delta_B(s)$, $\delta\Delta_C(s)$ of Eq. (11) are applied for the peak characteristics of models B and C, shown in Fig. 10(b), so that the gain of $W_2(s)$ is large only within that frequency range in which the peak emerges.

$$
\begin{aligned}
W_2(s) &= k\left(\frac{c' \cdot s + 1}{c \cdot s + 1}\right)^2 \\
\delta W_B(s) &= \frac{s^2 + b \cdot \omega_n \cdot s + \omega_n^2}{s^2 + a \cdot \omega_n \cdot s + \omega_n^2} \\
\delta W_C(s) &= \frac{s^2 + b' \cdot \omega_n \cdot s + \omega_n'^2}{s^2 + a' \cdot \omega_n \cdot s + \omega_n'^2} \\
W_2'(s) &= W_2(s) \cdot \delta W_B(s) \cdot \delta W_C(s)
\end{aligned}
\tag{11}
$$

where

ω_n, ω_n': Frequency corresponding to the peak of the complementary sensitivity function $T(s)$

a, b, a', b': Coefficients determining of the magnitude and the shape of the peak ($a < b$, $a' < b'$)

The redesigned weighting function $W_2'(s)$ is shown by a dashed line in Fig. 9. **Figure 11** shows the frequency response of the controller as redesigned in this step, in comparison with that in the first step. The gain of redesigned controller in the high-frequency domain is suppressed by the loop shaping of the peak. As a result, the controller enables an increase in the gain margin and enhances the stability of model B, which has a high gain in the high-frequency domain. On the other hand, in the low-frequency domain, the phase margin of model C, which has a large phase lag, increases because the phase of the controller is leading.

Actual vehicle test results are shown in **Fig. 12**, where the proposed method is compared with
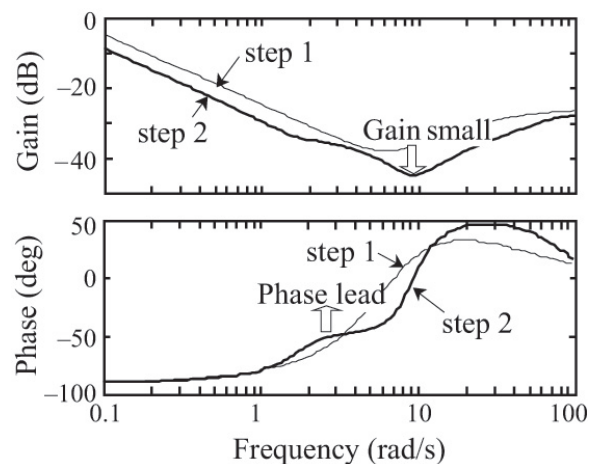

Fig. 9 Weighting function $W_2(s)$ and $\Delta(s)$.

(a) Simulation results

(b) Frequency responses of co-sensitivity functions $T(s)$

Fig. 10 Problem of the conventional method.

Fig. 11 Comparison between step1 and step 2 of H_∞ controller design.

conventional PID control. The response under PID control is the same as that for the proposed method under nominal conditions (which correspond to the conditions for model A). The responses are shown in Fig. 12(a). On the other hand, it is found that the proposed method is more robust than the conventional method under disturbed conditions, where the DC gain of the plant is intentionally increased to 2.5 times the original value (Fig. 12(b)).

4. 2 Application Example of Shift Control System

The design step of the proposed method mentioned in Section 3. 2 is next applied to the case of 2 to 3 upshift control.

Step 1: Preparation of possible target patterns
The 18 output torque and turbine revolution speed patterns, which are set as the possible target patterns, are shown in **Fig. 13**. Factors A, B, C, and D, which are typical characteristics of the output torque, are related to the three objective functions (objective functions I, II, and III in Section 2. 2). Therefore, the possible target output torque patterns are generated as combinations of specific values of each of factors A, B, C, and D; in particular, combinations of the 18 patterns are made with the $L18$ orthogonal tables in Fig. 13.[10]

Step 2: Generating oil pressure and engine torque patterns with inverse directional design
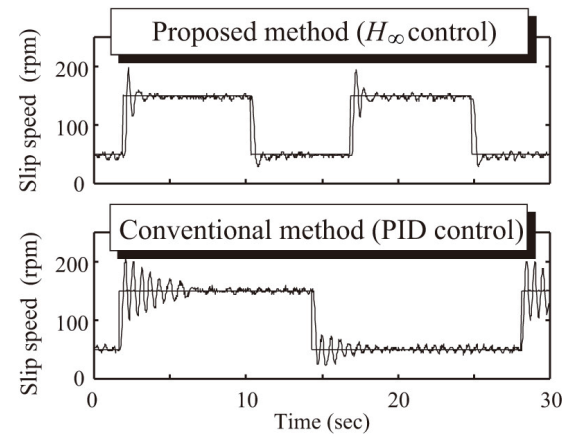We calculate the target oil pressure and engine torque patterns which are each related to the 18 target output torque and turbine revolution speed patterns set in step 1 with the iterative simulation of the gear train system (in **Fig. 14** above).

Step 3: Adding hydraulic variations (upper and lower values)
To verify the robustness, ±15 [kPa] hydraulic variations are added to each target oil pressure pattern P_{c3} calculated in the previous step, and the effect on the objective functions is investigated. There exist clutch friction coefficient variations, which are replaced with equivalent hydraulic variations. The variation ΔT of the clutch transmitted torque T, which affects the output torque, can be expressed by the variation $\Delta\mu$ of the friction coefficient μ or the variation ΔP of the oil pressure P, as shown in Eqs. (12a) to (12c).
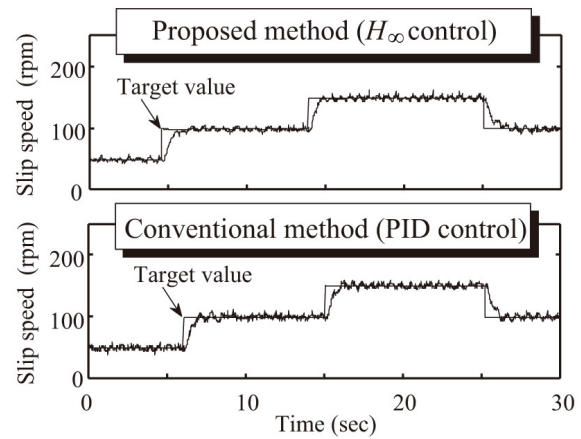
(a) Nominal condition: Light load (Throttle opening = 5%, constant)

(b) DC-gain variation: High-load (Throttle opening = 23%, constant)

Fig. 12   Experimental results (on-vehicle).
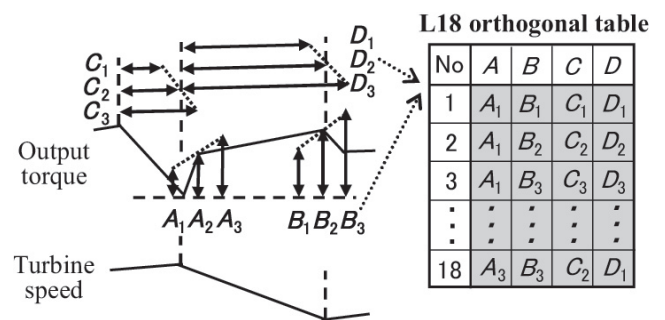
Fig. 13   Preparation of target patterns.

$$T + \Delta T = (\mu + \Delta\mu)AR \cdot (P + \Delta P) \qquad (12a)$$

$$\Delta T \approx AR(\Delta\mu \cdot P + \mu \cdot \Delta P) \equiv AR \cdot \mu \cdot \Delta P' \qquad (12b)$$

$$\Delta P' = \Delta P + (P / \mu)\Delta\mu \qquad (12c)$$

Step 4: Robustness evaluation

The characteristic variations of Step 3 are added to the plant model, and simulations are performed. Each objective function is evaluated using the S/N ratio[10] η in Eqs. (13a) to (13c).

Shift shock

$$\eta_s(i) = -10 \log (J(i)_{s0}^2 + J(i)_{s+}^2 + J(i)_{s-}^2) \qquad (13a)$$

Shift time

$$\eta_t(i) = -10 \log (J(i)_{t0}^2 + J(i)_{t+}^2 + J(i)_{t-}^2) \qquad (13b)$$

Heat load

$$\eta_q(i) = -10 \log (J(i)_{q0}^2 + J(i)_{q+}^2 + J(i)_{q-}^2) \qquad (13c)$$

where $i = 1, \ldots, 18$, J_0, $J+$, J_-: no variation, upper variation, lower variation

Next, the relationship between the S/N ratios and the characteristic factors A, B, C and D of the target patterns are represented as graphs of factorial effects,[10] shown in **Fig. 15**. For example, as shown in Eq. 13d, the S/N ratio of the parameter A_1 is equal to an average of η for No.1, 2, 3, 10, 11 and 12 that correspond to A_1 of the orthogonal table in Fig. 13.

$$\eta_s(A_1) = \frac{\eta_{s1} + \eta_{s2} + \eta_{s3} + \eta_{s10} + \eta_{s11} + \eta_{s12}}{6} \qquad (13d)$$

The better the evaluation result is, and the smaller the magnitude of characteristic variation, the larger the calculated S/N ratio becomes. This graph of factorial effects enables us to find the characteristic factor, which mostly affects each objective function, and select the most robust level of each characteristic factor. However, the two best levels of characteristic factors may change due to the design weighting to shift shock and weighting to clutch heat load, as in the characteristic factor D in the figure. Therefore weighting to each objective function is performed and the level of the characteristic factor is selected for each shift quality according to the weight. Concretely at first, the graphs of factorial effects are approximated with quadratic functions (as shown by the dashed lines in the figure), secondly each function is weighted (α_1, α_2, α_3) so that one quadratic function can be made from each weighted function, and last, the levels are selected so that the value of the function can be a maximum. Examples of the target torque patterns are shown in **Fig. 16** where the designs are (a) weight (0.7, 0.2, 0.1) to shift shock and (b) weight (0.1, 0.2, 0.7) to clutch heat load. It is obvious from the factor $(C + D, B)$ of (a) weight to shift shock in the figure that the shift time is long and the factor B of torque change is small.
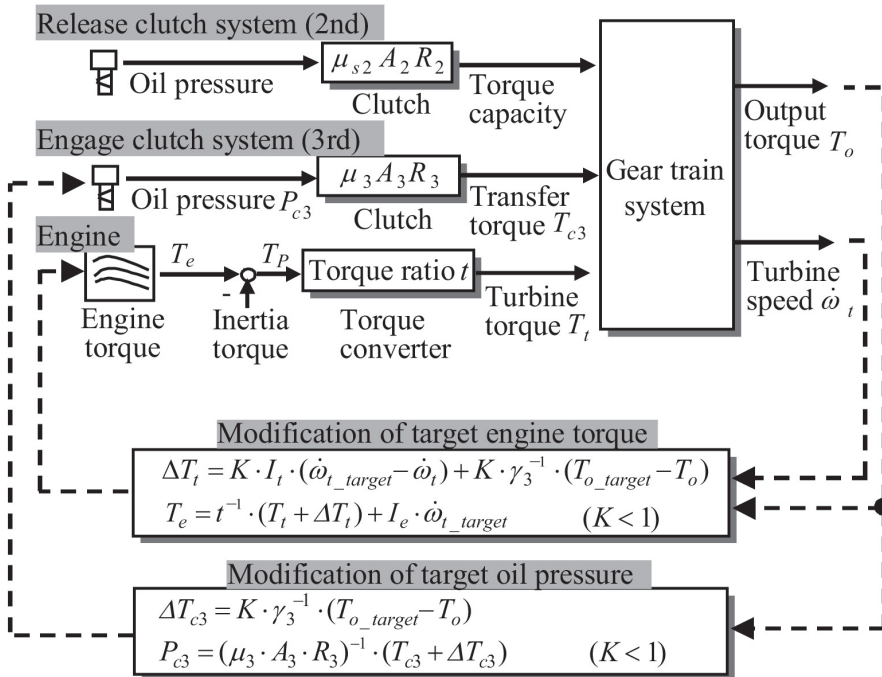
Step 5: Inverse directional design of control signals

First, the target oil pressure pattern is calculated from the target output torque, and a control signal for the oil pressure is then calculated from the target oil pressure. The flow is shown in **Fig. 17**. An iterative calculation method is applied because of the nonlinear elements included in the model.

To verify the robustness, two oil pressure patterns were designed. One design weighted the shift shock requirement, and the other
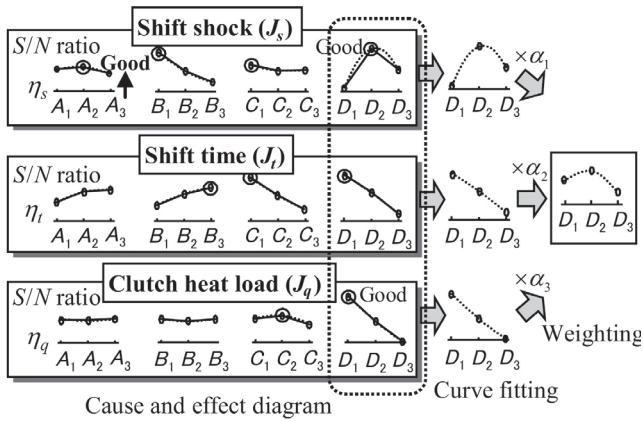

Fig. 14 2 to 3 up-shift simulation model

Fig. 15 Determining of robust target output torque.

Fig. 16 Robust target output torque patterns.

Fig. 17 Flow of the inverse directional design.

weighted the clutch heat load requirement. Simulations were performed where ±15 [kPa] hydraulic variations were added to the designed oil pressure patterns for verification. The results are shown in **Figs. 18**(a) and (b). The changes in the objective function values after
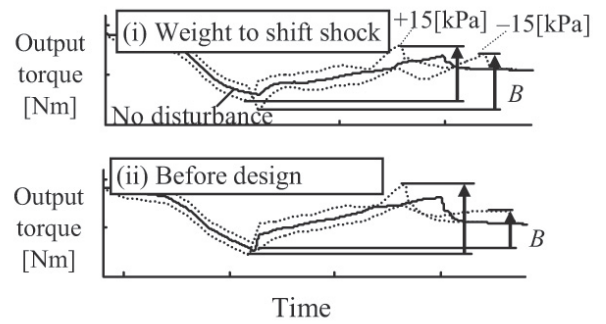
(a) Robustness of shift shock
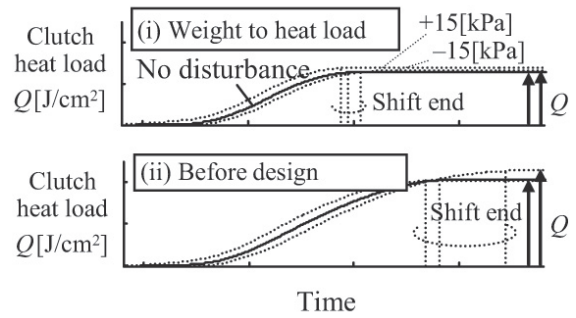
(b) Robustness of heat load

Fig. 18 Simulation results (2 to 3 up shift of 8 speeds AT).

the design are smaller than those before the design, even though hydraulic variations exist. Thus, the validity of the proposed method is demonstrated.

5. Conclusion

In the present study, robust design methods were proposed for clutch control, which include feedback and feedforward methods that are generally applied to the clutch control systems of automatic transmissions. A torque converter lock-up clutch slip system and a shift control system were used as typical examples of feedback and feedforward controls, and the validity of the proposed methods was verified.

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