Improvement of Maneuvering Feeling of Human-Mechanical Cooperative System and Its Application to Electric Power Steering System

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Abstract: In human-mechanical cooperative systems, a significant issue is to improve the control performance and the maneuvering feeling of human operation. However, since it is not easy to evaluate the feeling of operators numerically, control engineers design controllers only through experience. Thus, in this paper, a new evaluation method for control performance of human-mechanical cooperative system is proposed based on the reserge waveform. Various distortions of waveform represent deteriorations of control performance and maneuvering feeling. In some cases, since there is a tradeoff between the control performance and the maneuvering feeling, it is difficult to compensate for both of them by usual feedback controllers. To overcome this situation, the two degrees of freedom control system is applied to human-mechanical cooperative system. Some numerical simulation results for an electric power steering system are shown to confirm the effectiveness of proposed control design method.

Keywords: human-mechanical cooperative system, maneuvering feeling, reserge waveform, two degrees of freedom control, electric power steering system

1. INTRODUCTION

As the mechatronis technology has recently developed, the robust control techniques have been widely applied to mechanical systems in industrial applications. The robust control strategy is one of the superior approaches to improve the stability and control performance by compensating for the robustness against uncertainties in actuators and mechanical load and nonlinearity such as friction and backlash.

However, in human-mechanical cooperative system, it is also significant to improve the maneuvering feeling of human operation. For example, a power steering system is one of the assisting systems for human steering operations. The hydraulic power steering system(HPS), has been main stream for the power steering system. However, since hydraulic pump is always driven by engine, fuel efficient is remarkably worse. Recently as the motor drive technique has been developed, the electric power steering system(EPS) is introduced to power steering system because of fuel economy and lightweight. However, the control theory has been mainly applied to compensate for only the control performance of mechanical flexibility and disturbances[1][2][3]. Therefore, it is desirable to apply the unified design method based on the robust control theory to the mechanical system in order to improve not only the control performance but also maneuvering feeling of human operation.

From this background, the control problem of human-mechanical cooperative system is discussed. The control purpose in this paper is not only to compensate for the stability for mechanical flexibility and disturbances but also to improve the maneuvering feeling of a human operation. To this end a new evaluation method for control performance is proposed to measure maneuvering feeling of human operation. Deteriorations of the maneuvering feeling in such a system are caused by mechanical mass, mechanical oscillation, the nonlinearity such as friction and backrush, and so on. In order to evaluate such performance degradation, the reserge waveform is proposed. On the basis of the qualitative and quantitative evaluation using the reserge waveform, the control specification can be determined. Then the two degrees of freedom control system is applied to achieve the control purpose.

In the following section, the mathematical model of a human-mechanical cooperative system is formulated. In Sec.3 the characteristics of the reserge waveform to evaluate the control performance is explained. In Sec. 4 the two degrees of freedom system is designed to achieve the control purpose. Finally, in Sec 5, the proposed design method is applied to the electric power steering system and some numerical simulation results are shown to confirm the effectiveness.

2. MATHEMATICAL MODEL OF HUMAN-MECHANICAL COOPERATIVE SYSTEM

The human-mechanical cooperative system treated in this paper is shown in Fig. 1.

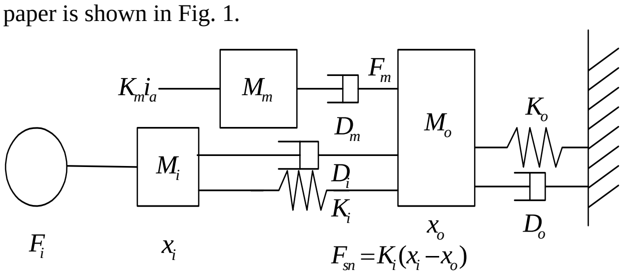


Fig.1 Human-mechanical cooperative system

The mechanical system considered in this paper is described by a three mass-damper-spring system. The demand force by the operator is inputted to the mechanical system and its actuation is assisted by the additional force generated by DC motor. DC motor is attached at the second mass-damper-spring system through a gear box. The total force from the operator and the motor is affected to environment through the second mass system. The parameters are denoted in Table 1.

Table 1 System parameters

mass of input side	M_i
viscous damping coefficient of input side	D_i
stiffness coefficient of input side	K_i
mass of out side	M_o
viscous damping coefficient of output side	D_i
stiffness of input side	K_i
mass of motor shaft and gear box	M_m
viscous damping coefficient of motor shaft	D_m
transfer coefficient from motor current to linear force	K_m
demand force of operator	F_i
coulomb friction of input side	F_{cs}
coulomb friction of output side	F_{co}
coulomb friction of motor shaft and gear	F_{cm}
assistant force generated by motor	F_m

From the figure the motion equations of the system are obtained as follows:

Motion equation of the first mass system

$$F_i = M_i\ddot{x}_i + D_i\dot{x}_i + K_i(x_i - x_o) + F_{ci}\cdot\mathrm{sgn}(\dot{x}_i)\,. \tag{1}$$

Motion equation of the second mass system

$$K_i(x_i - x_o) + F_m = M_o\ddot{x}_o + D_o\dot{x}_o + K_o x_o + F_{co}\cdot\mathrm{sgn}(\dot{x}_o)\,. \tag{2}$$

Motion equation of the motor at the output side of a gear

$$F_m = K_m i_a - \{M_m\ddot{x}_o + D_m\dot{x}_o + F_{cm}\cdot\mathrm{sgn}(\dot{x}_o)\}\,. \tag{3}$$

Substituting Eq.(3) to Eq.(2) yields

$$K_i x_i + K_m i_a = M\ddot{x}_o + D\dot{x}_o + Kx_o + F_c\,, \tag{4}$$

where

$M = M_o + M_m$,

$D = D_o + D_m$,

$K = K_o + K_i$,

$F_c = (F_{co} + F_{cm})\cdot\mathrm{sgn}(\dot{x}_o)\,.$

From Eq.(1) and Eq.(4) the block diagram of the system is represented in Fig.2. It is assumed that a reaction force F_{sn} against the operator, which is generated from the difference between x_i and x_o, is measured by a torque sensor.

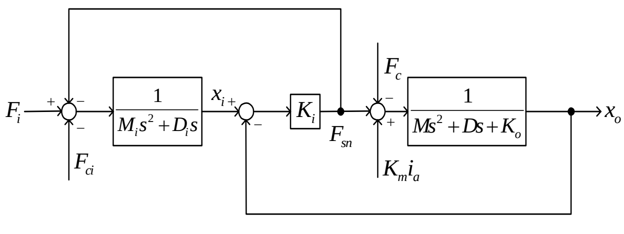
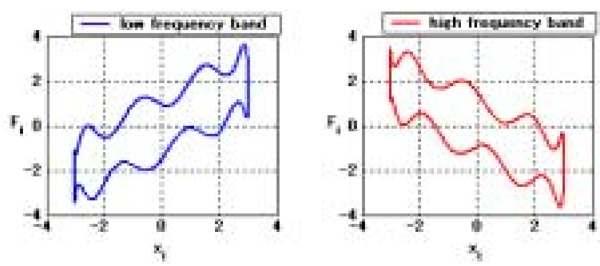


Fig.2 Block diagram of the system

3. EVALUATION OF MANEUVERING FEELING BY RESERGE WAVEFORM

When an operator maneuvers a mechanical system, he/she often feels several deteriorations of the maneuvering. There are two kinds of factors for unpleasant reaction force. One is externally caused by disturbances received from environment. The other is internally caused by mechanical mass, mechanical flexibility, and nonlinearity such as friction and backrush. These have been conventionally evaluated by operators' own feelings and experiences, and thus there have been no systematic method to evaluate the maneuvering feeling. Hence it is expected to evaluate the maneuvering feeling quantitatively and to give information for control design guideline. To solve this problem, in this paper, a new criterion on the basis of the reserge waveform is proposed.

In Fig. 3, examples of reserge waveforms of the considered model are shown. The reserge waveform is drawn by input-output signals where the input is the maneuvering position x_i (horizontal axis) and the output is the maneuvering force F_i (vertical axis). The sinusoidal signal is inputted. Figures (a) and (b) express the results in cases when input frequencies are different.



(a) (b)

Fig.3 Reserge waveform

As shown in the figure, there are some distortions of waveform. To investigate the causes of distortions, some numerical simulations were executed by changing parameters of mathematical model. As a result, the following results was found(see Fig.4).

1. According to the increase of masses, the inclination of the waveform rotates clockwise when the frequency of input signal increases. (Part A in Fig.4)
2. According to the increase of the coulomb friction F_{ci} at the input side, the width of the part B becomes larger.(Part B in Fig.4)
3. According to the increase of viscous damping coefficients of the mechanical system, the oscillation at the part C decays and the shape of the reserge waveform approximates a circle. (Part C in Fig.4) The magnitude of this oscillation is different in accordance with frequency of input signal, and it becomes larger in high frequency band.
4. According to the increase of the coulomb friction F_c, the distortion at the part D becomes larger and at the same time the width of the part B also becomes larger.
5. According to the decrease of stiffness coefficients, the inclination of the waveform becomes steeper.

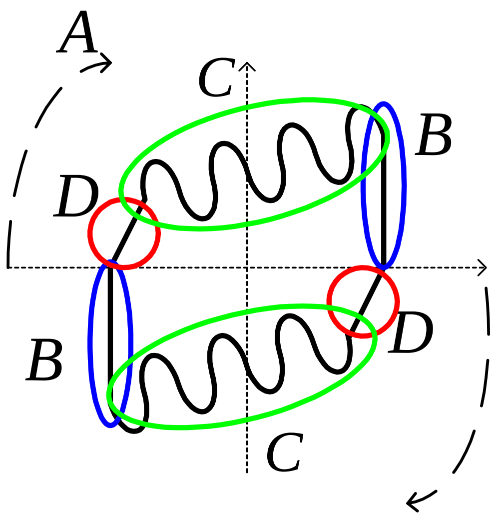


Fig.4 Analysis using the reserge waveform

From the above numerical results, their causes are guessed as follows: The first result shows that the phase difference between the input and the output signals becomes wider according to the increase of frequency of an input signal. In this case an operator may feel distasteful response delay between a demand displacement and a maneuvering force. The second and the fourth results show that coulomb frictions make the maneuvering feeling heavier, in particular, when an operator changes the direction of his/her maneuver. The third result suggests that the oscillation distortion is caused by vibrations of mechanical system, because the increase of viscous damping coefficients leads the rise of stability margin. Therefore this oscillation distortion implies the lack of stability margin in the corresponding frequency band.

From these quantitative evaluations of shape of the reserge waveform, the control design policy can be planed along this scenario.

4. APPLICATION OF TWO DEGREES OF FREEDOM CONTROL TO HUMAN-MECHNICAL COOPERATIVE SYSTEM

4.1 Two degrees of freedom control system[4]

It is well known that the two degrees of freedom control is effective technique satisfying both the transient performance and the stability. The block diagram of a typical two degrees of freedom control system is shown in Fig. 5.

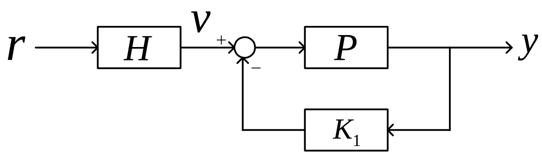


Fig.5 Two degrees of freedom control system

K_1 is a feedback controller and H is a feed-forward controller, respectively. In general, the feedback characteristic such as stability and disturbance attenuation is mainly improved by a feedback controller. However, at the same time the transient performance from a reference signal r to an output y may not be necessarily improved by only a feedback control, because these two different characteristics are the relation of tradeoff. For this reason, in the two degrees of freedom control, a feed-forward controller is inserted to improve the transient performance.

The basic concept of this control system is as follows: Set the transfer function G_{yv} from v to y as

$$G_{yv} = \frac{P}{I + PK_1}. \tag{5}$$

Then by setting the feed-forward controller H as

$$H = G_{yv}^{-1} G_{yr}^{ref}, \tag{6}$$

the transfer function from r to y becomes the desired one G_{yr}^{ref}.

The main advantage of two degrees of freedom control system is to realize both the transient performance and the feedback characteristic by combining feedback and feed-forward controllers. However, since a inverse system of a plant is used in Eq.(6), H becomes unstable when an unstable zero exists in G_{yv}. Moreover when H is not proper, its realization also becomes difficult.

4.2 Two degrees of freedom control system using coprime factorization

As mentioned in the pervious section, there are some disadvantages to realize when H is unstable and/or is not proper. Moreover, H is designed by trial and error, because H depends on the feedback controller K_1. In order to avoid these disadvantages, the two degrees of freedom control system is constructed by using the coprime factorization of the transfer function P. The control configuration using this representation is shown in Fig. 6. Here N and D denote right coprime factorizations of P defined by

$$P = ND^{-1},\ N \in RH_\infty,\ D \in RH_\infty. \tag{7}$$

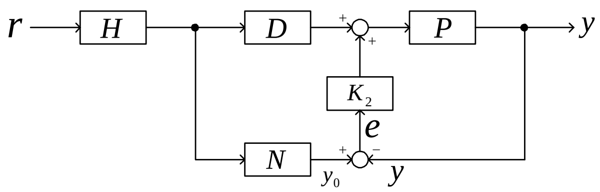


Fig.6 Two degrees of freedom control system using coprime factorization

K_2 in Fig.6 denotes a feedback controller. From this figure the transfer function from r to y becomes

$$y = NHr. \tag{8}$$

Thus by setting H to satisfy

$$G_{yr}^{ref} = PDH = NH, \tag{9}$$

the desired transient performance can be realized independently from the selection of an feedback controller K_2.

4.3 Application to the human-mechanical cooperative system

Under consideration in Sec.3 the control purposes of the human-mechanical cooperative system are explained as follows:

1. To compensate for stability margin at the frequency band where the mechanical resonance occurs.
2. To compensate for frictions of both mechanical system and motor shaft.
3. To compensate for the degradation of maneuvering feeling caused by the delay between a demand displacement and a maneuvering force.

The last purpose can be achieved by compensate the influence of masses of mechanical system. In order to these requirements the two degrees of freedom control is applied.

Set the reference input and the controlled output as

$$r = x_i,\ y = F_{sn}. \tag{11}$$

Then the proposed control system is represented in Fig. 7.

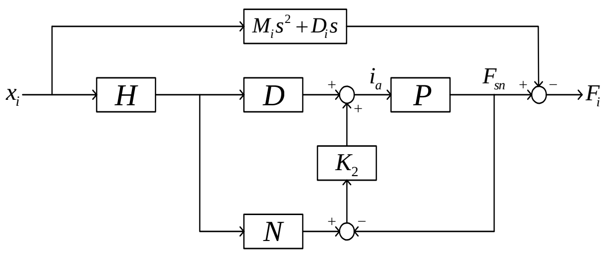


Fig.7 Two degrees of freedom control system for human-mechanical cooperative system

In Fig.7,

$$P = \frac{K_m K_i}{Ms^2 + Ds + K_o} \tag{12}$$

By comparing Fig.7 with Fig.2, the block diagram of the human-mechanical cooperative system is also represented in Fig.8.

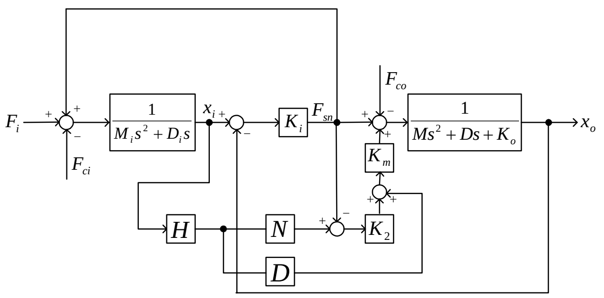


Fig.8 Block diagram of human-mechanical cooperative system with two degrees of freedom control

The desired transfer function G_{ref} is design considering the improvement of mass, viscous damping coefficient, and stiffness coefficient as follows:

$$G_{ref} = -\frac{K^{ref}{}_i}{M_{ref}s^2 + D_{ref}s + K_{ref}}. \tag{13}$$

where M_{ref}, D_{ref}, K_{ref} and K_i^{ref} are designed to improve control performances explained before. Then the feed-forward controller H is determined by

$$H = G_{ref} N^{-1}. \tag{14}$$

Moreover the feedback controller is designed to improve the stability in the frequency band of the mechanical resonance and to compensate for coulomb frictions.

5. APPLICATION TO ELECTRIC POWER STEERING SYSTEM

In order to verify the effectiveness of the proposed control strategy, it is applied to an electric power steering system and some numerical simulations are shown. As mentioned in introduction, an electric power system is an typical example of the human-mechanical cooperative system, although there is an difference between a linear motion and a rotational motion.

By transforming notations in Sec.2 as

$$F_i \to T_i,\ x_i \to \theta_i,\ F_{sn} \to T_{sn},\ x_o \to \theta_o,$$
$$M_i \to J_i,\ M_o \to J_o,\ M \to J,\ M_m \to J_m, \tag{15}$$

the previous discussions are able to be applied to an electric power steering system straightforward.

First of all, the reserge waveforms of electric power steering are shown in Fig.9. The input signal is a steering angle and the output signal is a steering torque. The magnitude and frequencies of input signal are

$$\theta_i = 40\sin(2\pi f)\ [\mathrm{deg}],\ f = 0.2, 0.5, 1.0, 1.8[Hz], \tag{16}$$

where the frequency band is selected considering usual human operations for steering. In the case of without control, no assisting control is used for steering. On the other hand, in the case of conventional control, the assisting control is applied. In this case, a phase lead-lag controller is inserted in a current feedback loop in order to compensate for the stability margin.

As shown in Fig.9, the oscillation distortions of the reserge waveform are improved owing to the compensation for stability margin and coulomb frictions by a phase lead-lag controller. However, the inclination of the waveform rotates clockwise when the frequency of the input signal increases. This phenomenon is due to the phase delay of input-output signals in a frequency band of human operations.

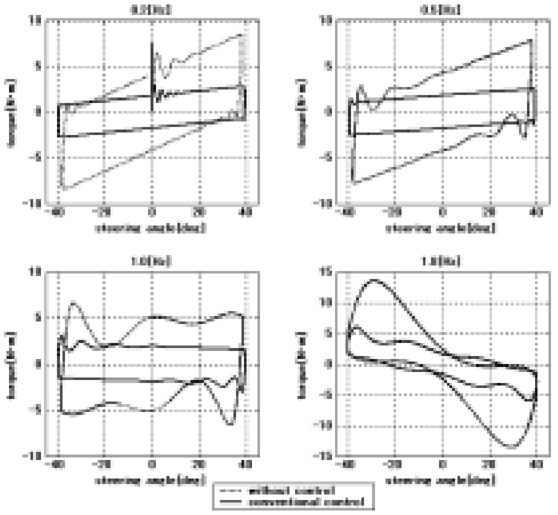


Fig. 9 Reserge waveforms for electric power steering system

To make a breakthrough in this situation, the two degrees of freedom control is applied to this system. The control design purposes are to suppress the rotation of inclination of the reserge waveform, to compensate for stability margin of mechanical resonance vibration, and to reduce influence of coulomb frictions. To achieve these control purposes, controllers of two degrees of freedom control system are designed as follows:

1. The feedback controller K_2 is designed to improve stability margin for the mechanical resonance vibration mode and to reduce influence of coulomb frictions. The phase lead-lag compensator is used similarly to the conventional controller. However, the parameters of the controller is retuned.
2. The feed-forward controller H is designed to suppress the rotation of the inclination of the reserge waveform in a frequency band of human operations. As mentioned in Sec.3, this is mainly caused by the curse of masses of mechanical and motor systems. Hence the feed-forward controller H is designed to remove superfluous masses and to let the transient response of the closed loop system follow the desired one. In fact for the plant transfer function

$$P = \frac{K_m K_i}{Js^2 + Ds + K_o}, \tag{17}$$

the desired transfer function is modified to

$$G_{ref} = \frac{K^{ref}{}_i}{J_{ref}s^2 + D_{ref}s + K_{ref}}. \tag{18}$$

where J_{ref}, D_{ref}, K_{ref}, K_i^{ref} are appropriately selected to suppress the rotation of the inclination of the reserge waveform.

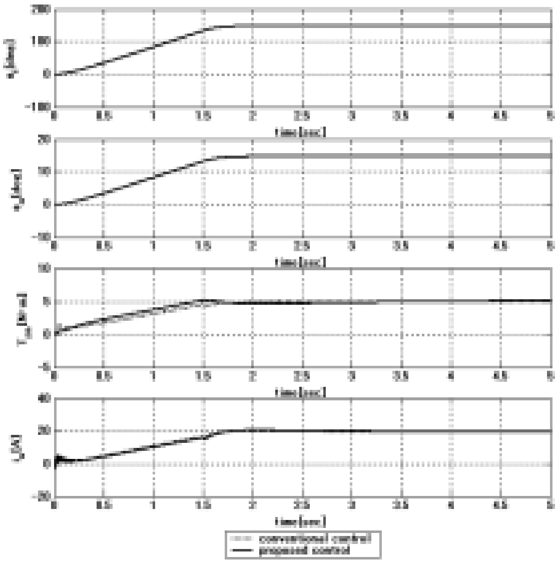


Fig.10 Transient responses in slow steering case

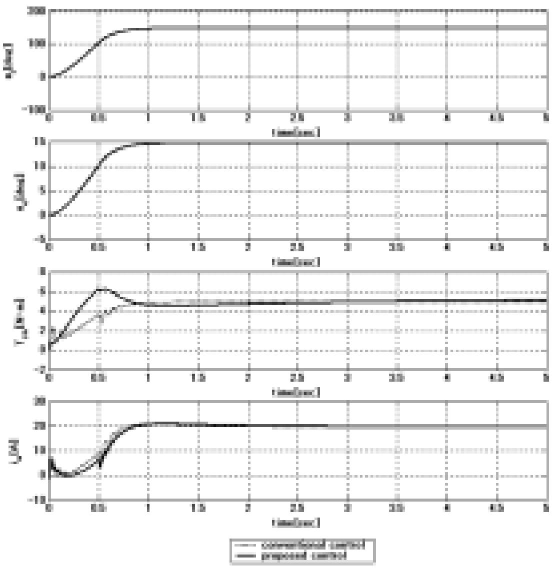


Fig.11 Transient responses in quick steering case

Simulation results of transient responses in cases changing steering speed are shown in Fig.10 and Fig.11. Fig.10 shows a slow steering case, and Fig.11 a quick steering case. In each figure θ_i, θ_o, T_{sn}, i_a are represented, where they denote input steering angle, output wheel angle, torque sensor output, and motor current, respectively. It is found that the transient response of torque sensor output follows the desired one by proposed controller in each case. At the same time the stability is somewhat improved.

On the other hand, the reserge waveforms in each case are shown in Fig. 12. It is clear that rotation of the inclination of the reserge waveform is suppressed in case of the proposed controller compared with other cases. Moreover it is

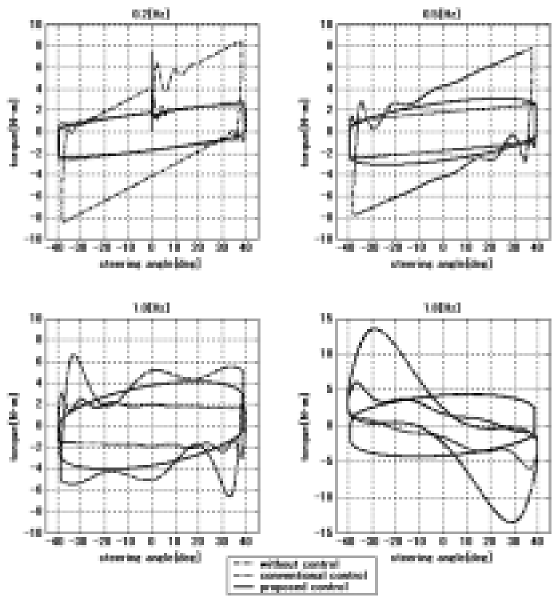


Fig. 12 Improved reserge waveforms

found that stability margin and influence of coulomb frictions are also improved.

6. CONCLUSIONS

In this paper a new method to evaluate the control performance for human-mechanical cooperative system is presented. This enables to evaluate the maneuvering feeling of human operations. In this paper, while the qualitative evaluation is only discussed, it is possible to evaluate the control specification quantitatively if the experimental data are accumulated.

Under consideration of distortions of the reserge waveform, the deterioration for maneuvering feeling is discussed. As a result, it is necessary to compensate not only for the usual lack of stability margin due to mechanical flexibility but also for the lag of maneuver caused by superfluous mechanical and motor masses. However, it is not easy to satisfy these requirements, because of a relation of tradeoff between the control performance and the maneuvering feeling. Thus the two degrees of freedom control system is proposed to compensate for both of them. It enables to achieve different two purposes independently. Some numerical simulations applied to an electric power system were represented. The result showed the effectiveness of proposed evaluation method and control system.

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