Design for Sensorless Force Control of Flexible Robot by Using Resonance Ratio Control Based on Coefficient Diagram Method

Generally, the flexible robot system can be modeled as the two-mass system which consists of a motor and load connected by a spring. Thus, its elasticity causes resonance in the system. By using the conventional PID controller, this method cannot perform well in this situation. Much research has proceeded with the aim of reducing vibration. A new effective control method, the resonance ratio control, has been introduced as a new way to guarantee the robustness and suppress the oscillation during task executions for a position and force control. In this paper, three techniques are proposed for improving the performance of resonance ratio control. Firstly, a new multi encoder based disturbance observer (MEDOB) is shown to estimate the disturbance force on the load side. The proposed observer is not necessary to identify the nominal spring coefficient. Secondly, coefficient diagram method (CDM) has been applied to calculate a new gain of the force controller. A new resonance ratio gain has been presented as $2.0$. Finally, the MEDOB and load side disturbance observer (LDOB) are employed to identify a spring coefficient of flexible robot system. By using the proposed identification method, it is simple to identify the spring coefficient and easy to implement in the real flexible robot system. The effectiveness of the proposed identification method is verified by simulation and experimental results.

**Key words:** Disturbance observer, Two-mass system, Coefficient diagram method, Resonance ratio control

1 INTRODUCTION

Recently, society demands new robot designed to support human. These robots are required to have an ability to accommodate the interaction potential with human operator. In the conventional force control system, several proposed techniques have paid attention to develop force control system and implemented force sensors to detect the external force. However, the force information detected by force sensor has narrow bandwidth due to the soft structure of the strain gauge [1]. To improve the control performance, the sensorless force control technique is proposed to estimate the external force without force sensor. Many research have involved this techniques in order to reduce
the complicity of the overall control system and increase their stability as shown in [2, 3].

In the industrial robot applications, the effects of structural flexibility can lead to limit the performance of a control system [4, 5]. The flexible robot system can be modeled as the two-mass system which consist of a motor and load connected by a spring. Since the actuator is connected to the load side with a transmission mechanism, its elasticity causes resonance in the system. Generally, the conventional PID controller is widely used in the industrial machines and robotics. Although this method offers a simple way to design the controller, it does not perform well when the plant system is unstable or has a resonance structural vibration. Therefore, it is necessary to design the controller that guarantees the robustness and suppresses vibration.

Controlling machine with the mechanical resonance has been receiving increased attention [6]-[7]. A new effective control method, the resonance ratio control [8], has been introduced by Yuki et al. as a new way to guarantee the robustness and suppress the oscillation during task executions for position control system. This method relies on the possibility to change the virtual inertia moment in the motor side by feeding back the estimated reaction torque to the motor in an acceleration dimension. This means that the resonance frequency and the resonance ratio of the system can be changed to an arbitrarily value. In the position control, it is found that the vibration is well compensated by using the resonance ratio gain as $\sqrt{5}$. In addition, the sensorless force servoing based on resonance ratio control was proposed by Katsura [9]. This method has proven itself as an excellent controller for vibration compensation by setting the resonance ratio gain as $\sqrt{6}$.

In this paper, three techniques are proposed for improving the performance of resonance ratio control:

1) A coefficient diagram method (CDM) [10] has been applied to calculate a new gain of the force controller. The CDM method is an algebraic approach combining classical and modern control theories and applied polynomials to represent the system. More reliable parameters selection can be designed based on the stability index and equivalent time constant. From the calculation, a new resonance ratio gain based on the CDM has been presented as 2.0.

2) A multi encoder based disturbance observer (MEDOB) is proposed and applied to estimate the external force on the load side, rather than using the load side disturbance observer (LDOB) [9] or force sensor [4]. The proposed observer is not necessary to estimate the nominal spring coefficient.

3) A new parameter identification is also proposed based upon the LDOB and MEDOB. The correct value of the parameter identification is needed for model based controlling of flexible robot and designation of structural observer. There are two parameters due to the error in the load side external force estimation of LDOB, which are because of the nominal load mass and the spring parameters. From the experimental setup, it is easy to find the nominal load mass of the system by measuring or finding this value from catalog. By the way, the correct spring coefficient is very difficult to obtain. Analysis of the calculation reveals that the correct region of spring coefficient is found if the response of the load side external force from both LDOB and MEDOB are equal.

From the simulation and experimental results, it is shown that this new identification method, which has the comparison of MEDOB and LDOB, is capable of accurately approximating the spring coefficient. It is also confirm that a new resonance ratio gain as 2.0 is also suitable for suppressing vibration of the mechanical resonance.

![Fig. 1. The model of flexible robot system](image-url)

### 2 TWO-MASS SYSTEM MODEL

In recent years, robot systems and intelligent machines have been used not only for industrial processes but also for a wide range of applications. Generally, the mechanics of machine and robot system often consist of a motor $M$ and a load $L$ connected by a flexible structure to transmit the actuator torque to a distant joint as shown in Fig. 1. Since the actuator is connected to the load side with a transmission mechanism, its elasticity causes resonance in the system. Therefore, the elasticity of flexible robot is realized under assumptions that it can be modeled by spring coefficient $K_s$ and mass system. A block diagram of the linear motor with two-mass system that controls under the ideal current source is shown in Fig. 2. The dynamic equation of the linear motor is described by:

\[
\begin{align*}
\dot{X}_m &= -\frac{K_s}{M} X_s + \frac{K_t}{M} I_m, \quad (1) \\
\dot{X}_l &= \frac{K_s}{L} X_s - \frac{1}{L} F_d, \quad (2) \\
\dot{X}_s &= \dot{X}_m - \dot{X}_l, \quad (3)
\end{align*}
\]

where subscripts $m$ and $l$ denote the motor side and load side, respectively. $F_s$ and $F_d$ are the spring force and the external force on the load side. $K_t$ is the force coefficient, $I_m$ is the current, $X_s$ is the torsional position from the position of the motor $X_m$ and the position of the load $X_l$, respectively.
M and L denote the equivalent mass of the linear motor and load mass, respectively. Naturally, this flexible transmission can negatively affect the overall performance in terms of increased vibrations.

From Fig. 2, the transfer function from current reference \( I_m \) to the velocity of motor side \( \dot{X}_m \) and the transfer function from current reference \( I_m \) to the velocity of load side \( \dot{X}_l \) can be calculated as follows

\[
\dot{X}_m = \frac{K_t}{s^3 + (\frac{K_t}{K_L} + \frac{K_t}{M})s},
\]

(4)

\[
\dot{X}_l = \frac{K_t}{s^3 + (\frac{K_t}{K_L} + \frac{K_t}{M})s}.
\]

(5)

Then, the anti-resonance \( \omega_{ar} \) and resonance frequencies \( \omega_r \) can be described as follows

\[
\omega_r = \sqrt{\frac{K_s}{L} + \frac{K_t}{M}},
\]

(6)

\[
\omega_{ar} = \sqrt{\frac{K_s}{L}}.
\]

(7)

3 ESTIMATION OF DISTURBANCE AND EXTERNAL FORCE

3.1 Disturbance Observer

Consequently, a disturbance observer has been used, not only for improving the robustness and bandwidth of the control system, but also for estimation the external force for examples, [11]– [12]. It has been confirmed that a robust force control can be obtained, when a disturbance observer is implemented as the feed forward control for a disturbance force compensation loop. Here, the disturbance force on the motor side \( F_{dm} \) is defined as

\[
F_{dm} = F_{fm} + D_m \ddot{X}_m + F_s + \Delta M \dddot{X}_m + \Delta K_t I_m,
\]

(8)

where, the resistive force which the direction is opposite to the motor force is the summation of friction force \( F_{fm} \), viscous friction \( D_m \ddot{X}_m \) and the spring force \( F_s \). \( \Delta M \dddot{X}_m \) is the variation torque due to changed mass, \( \Delta K_t I_m \) is the force ripple due to space harmonics.

In this research, it is assumed that the influence of the friction force is very small by using a small-size of linear motor; thus, the spring force has almost the same value of the disturbance force. Disturbance observer is an external disturbance estimation algorithm that is based upon motor current and velocity information as shown in

\[
\dot{F}_{dm} = \frac{G_{dis}}{s + G_{dis}} \left( K_{tn} I_m + G_{dis} M_n \dddot{X}_m \right) - G_{dis} M_n \dddot{X}_m,
\]

(9)

\[
\dddot{X}_m = \frac{sG_{vm}}{s + G_{vm}} \dddot{X}_m,
\]

(10)

where \( K_{tn} \) and \( M_n \) are refer to the nominal force coefficient and the nominal motor mass, respectively. \( G_{dis} \) is the cut-off frequency of the disturbance observer, \( G_{vm} \) is the cut-off frequency of the motor velocity estimation. To apply in the real controller, accurate measured data of velocity is required to provide high accurate force sensing of disturbance observer.

3.2 Load Disturbance Observer

Because of the advantages and unique features of disturbance observer, this approach has received much attention in sensorless force control system. However, instability and vibration of system can occur when using disturbance observer for the two-mass system because of the presence of spring that degrade the overall performance.

To address these problems, a load disturbance observer (LDOB) [9], which is composed of the nominal spring coefficient \( K_m \), the nominal load mass \( L_n \) and the load-side encoder, has been proposed as follows
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3.3 Multi-Encoder Based Disturbance Observer

In this paper, a novel multi-encoder-based disturbance observer (MEDOB) is proposed for estimating the external force at the load side of flexible robot as shown in Fig. 5. The MEDOB is proposed to estimate the external force on the load side. However, the exact parameters of the spring coefficient is difficult to obtain. Moreover, it can be assumed by a non-linear relationship.

\[
F'_{dl} = K_s X_s - L_n \dddot{X}_l, \quad \hbox{(11)}
\]

\[
\ddot{F}'_{dl} = \left( \frac{G_{mob}}{s + G_{mob}} \left( K_{tn} \dddot{X}_l - \dddot{X}_m \right) + G_{mob} L_n \dddot{X}_l \right), \quad \hbox{(12)}
\]

\[
\dddot{X}_l = \frac{s G_{vel}}{s + G_{vel}} \dddot{X}_l, \quad \hbox{(13)}
\]

where \( G_{vel} \) and \( G_{mob} \) are the cut-off frequency of the load velocity estimation and the LDOB, respectively. The structure of LDOB as described in this section can be applied to estimate the external force at the load side as shown in Fig. 4. The LDOB is proposed to estimate the external force on the load side. However, the exact parameters of the spring coefficient is difficult to obtain. Moreover, it can be assumed by a non-linear relationship.

4 CONTROLLER AND IDENTIFICATION

4.1 Resonance Ratio Control Based on Coefficient Diagram Method

A new effective control method, the resonance ratio control, has been introduced by Yuki et al. as a new way to guarantee the robustness and suppress the oscillation during task executions for a torsional vibration in the position control system [8]. Briefly, this approach relies on the possibility to change the virtual inertia moment in the motor side by feeding back the estimated reaction torque to the motor in an acceleration dimension. This means that the resonance frequency and the resonance ratio of the system can be changed to an arbitrarily value. Also, in [9], the resonance control is used in order to achieve the excellent performance that compensates vibration for high-bandwidth force servoing. Due to the compensation of disturbance observer on the motor side, the simplified block diagram of this reaction force feedback is shown in Fig. 6. The controller of robot system consist of a force gain \( K_f \), a velocity gain \( K_v \) and a reaction force gain \( K_r \). The feedback force will depend on the force contacted on the robot by a known environment stiffness \( K_e \).
By setting these designed gains, the resonance ratio is chosen to be $\sqrt{6}$. Therefore, the force response exhibits no overshoot and vibration. In this paper, a new resonance ratio parameter is calculated by using the coefficient diagram method (CDM). The CDM design method is used to design the characteristic polynomial of the closed loop system by achieving a good balance of stability and good robust performance. As it is seen from equation (20), the coefficients of characteristic polynomial $a_i$ are found as

\begin{align*}
a_0 &= K_pK_v\omega_{ar}^2, \\
a_1 &= K_v\omega_{ar}^2, \\
a_2 &= \omega_v^2, \\
a_3 &= K_v, \\
a_4 &= 1.0.
\end{align*}

The standard stability indices $\gamma_i$ for the Manabe form are chosen as $\gamma_1=2.5$, $\gamma_2=2.0$, and $\gamma_3=2.0$.

\begin{align*}
\gamma_1 &= \frac{a_1}{a_0a_2} = \frac{K_v^2\omega_{ar}^2}{K_pK_v\omega_v^2} = 2.5, \\
\gamma_2 &= \frac{a_2}{a_1a_3} = \frac{\omega_v^2}{K_v^2\omega_{ar}^2} = 2.0, \\
\gamma_3 &= \frac{a_3^2}{a_2a_4} = \frac{K_v^2}{\omega_v^2} = 2.0.
\end{align*}

Thus, the controller parameters calculated by the design of CDM are given as follows

\begin{align*}
K_v &= K_{vc} = \sqrt{2}\omega_v, \\
K_p &= K_{pc} = \frac{4\omega_{ar}^2}{5K_v}, \\
K &= K_v = \frac{\omega_v}{\omega_{ar}} = 2.0, \\
K_r &= K_{rc} = \frac{3}{L_1}.
\end{align*}

By setting these controller gain, a new resonance ratio parameter is chosen as 2.0.

### 4.2 Feedforward Load Disturbance Compensation

The block diagram of the proposed control structure is shown in Fig. 7. A robust motion control of each motor has been achieved by feedback the inner disturbance observer loop. By using the divider’s gain $1/K_{m}$, the disturbance force is transformed to the compensation current $i_{db}$, and it is utilized to cancel out the disturbance on the motor side and the uncertainty of a system due to the parameter variation.

The load disturbance compensation is introduced to compute and feedback the estimated load disturbance force.
through the inverse system of the motor side. The inverse system can be represented by the transfer function from \( F_{\text{cmd}} \) to \( F_{\text{res}} \) as shown in (20) and from \( F_{\text{dl}} \) to \( F_{\text{res}} \) as follows

\[
\frac{F_{\text{res}}}{F_{\text{dl}}} = \frac{\frac{1}{T} K_x (s^2 + K_c s + K_1 K_2)}{\frac{s^4}{s^4 + K_v s^3 + \omega_n^2 s^2 + K_v \omega_n^2 s + K_p K_v \omega_n^2}}.
\]  

(38)

Then, the inverse system can be calculated by the following equation:

\[
\frac{F_{\text{cmd}}}{F_{\text{dl}}} = \frac{s^2 + K_v s + K_1 K_2}{K_p K_v}.
\]  

(39)

In the next section, the experiment and simulation examples illustrate that the CDM design method gives significantly improved performance both for a step input force command and for vibration suppression.

### 4.3 Parameter Identification Based on LDOB and MEDOB

The correct value of the parameter identification of spring coefficient is needed for model based controlling of resonance ratio and designation of structural observer as well. The main contribution of this paper is that, using the proposed parameter identification, the correct spring coefficient is obtained, improving for the performance of resonant ratio control. Figure 8 shows the proposed identification method, where the sinusoidal signal is fed as the force reference and the ramp signal is used to vary the spring coefficient of the LDOB. The sinusoidal signal used for the parameter identification is designed by the following equation:

\[
u(kT_s) = A_0 + w(k) \sin(w_k k T_s), \quad 0 \leq k \leq N - 1,
\]  

(40)

\[
w(k) = \text{Asat}(k/0.1N) \text{sat}((N - k)/0.1N),
\]  

(41)

\[
w_k = w_{\text{start}} + k/N (w_{\text{end}} - w_{\text{start}}),
\]  

(42)

where, \( w(k) \) is a weighting or window function that causes the input to start at zero, ramp up to amplitude \( A \), and ramp down to zero at the end. The “sat” is the saturation function, which is linear from -1 to +1 and saturates at -1 for arguments less than -1 and at +1 for arguments greater than +1. The constant \( A_0 \) is an offset value added to make the input have zero average value. Sampling time for command generation and total points of command signal are represented by \( T_s \) and \( N \), respectively.

The comparison of load side external force is used to identify a correct region of the spring coefficient. According to the structural of the observers, if the load side external force error between LDOB and MEDOB is nearly zero, then the correct region of spring coefficient is found.

### 5 SIMULATION RESULTS

#### 5.1 Simulation Results of the Proposed Identification Method

Numerical simulations are given to confirm the validity of the proposed parameter identification. The correct stiffness coefficient \( K_v \) was set as 1000 \( N/m \). The comparison result of the load side external force of LDOB and
MEDOB is shown in Fig. 9. The variation of the nominal stiffness coefficient $K_{sn}$ was bounded from 950 $N/m$ to 1050 $N/m$. It can be seen from 0 s to 4 s that the amplitude of the load side external force of LDOB is lower than MEDOB in the case of $K_{sn} < K_s$. By reaching the correct region of the spring coefficient at 5 s, the load side external force from LDOB can be tracked almost identically over the load side external force from MEDOB as shown in Fig. 9 (b) and (c). It is seen from results that adding too much stiffness parameter $K_{sn} > K_s$ give a poor response with very large amplitude of the load side external force estimation. Thus, at 5 s, the correct stiffness coefficient can be found as 1000 $N/m$.

5.2 Simulation Results of the Proposed Resonance Ratio Control

Numerical simulations are given to confirm the validity of the proposed approach. The results obtained from the proposed approach is compared with the results obtained from the conventional resonance ratio control and without the reaction force feedback. The comparison of the bode diagram with resonance ratio control is shown Fig. 10. From the result, the lowest order with the higher bandwidth and no-overshoot are obtained. In the simulation results, the force command input is a step function with the magnitude of 4.0 N. It is seen from Fig. 11 that without reaction force feedback give a poor response with a very large vibration. On the other hand, the vibration are well compensated in the both case of the resonance ratio control. It is clear from these results that the resonance ratio control based on CDM design approach gives a good response with no overshoot and short settling time compared to the conventional method.

6 EXPERIMENTAL RESULTS

The experimental setup of the flexible robot consists of the linear motors, linear guide devices, spring, two encoders at the motor and load side as shown in Fig. 12. The robot system is controlled by a PC using RT-Linux with a sampling time of 100 $\mu s$. The computer is equipped

![Diagram of the experimental setup](image-url)
with analog output board, analog input board, and encoder-pulse counter board. All of the data from sensors as well as the controllers are written in the C language. The specifications of the linear motor and the parameters used in the experiments are shown in Table 1.

### 6.1 Experimental Results of the Proposed Identification Method

Three classes of experiment was performed to identify the region of correct value of stiffness coefficient. Expere...
iments of contact motion are investigated. In this experiment, a hard environment, which is an aluminum block, was used.

First, we performed experiments with varying of the wide range of the nominal stiffness coefficient; the nominal stiffness coefficient is set to increase from 900 N/m to 1400 N/m. Figure 13 (a) show the comparison of the load side external force between LDOB and MEDOB. From the experimental results, the improvement in the load side external force estimation of LDOB are illustrated at 3.0-5.0s. Therefore, the region of the correct value of $K_{sn}$ are bounded from 1050 N/m to 1150 N/m, while providing only a rough estimate of parameters. In the second experiment, the nominal stiffness coefficient is set to increase from 1050 N/m to 1150 N/m. The estimation performance of LDOB is more improving than that of the previous experiment since the correct value of spring coefficient is applied in the observer. From the experimental result in Fig. 13 (b), the force peak of LDOB is over than the MEDOB in the interval time at 5.2 s. An increase of the nominal stiffness coefficient produced the amplitude error enlargement of the load side external force estimation. Therefore, we acquired the correct region of the nominal stiffness coefficient as 1100 N/m. When the nominal stiffness coefficient at 1100 N/m is utilized in the LDOB as shown in Fig. 13 (c), we found that the external force response of the LDOB is almost equal to the MEDOB. By using the proposed parameter identification, it is possible to identify parameters with a very few experiment.

6.2 Experimental Results of the Proposed Resonance Ratio Control

The experiment was performed to evaluate the performance of the proposed approach with the input step function. All of the identified parameters as shown in the previous section was used to calculate the controller gain. A flexible robot system is moved to contact the aluminum block.

The controller without the reaction force feedback compensation was chosen to illustrate the worst-case vibration. Figures 14 (a) and (b) show the force response at the load side and motor side, respectively. Without the vibration compensation, it is found that the response of force and position are oscillated as shown in Fig. 14 (b) and (d). The influence of the reaction force feedback can be observed to significantly reduced the effective vibration by using the resonance ratio control. The experimented response also shows a good correlation with the simulation results. The improvement in the position and velocity responses at the motor side and load side are illustrated in Fig. 15. By using the proposed resonance ratio control based on the CDM, the best results for the faster response without overshoot is obtained. The vibration on the system can be rejected very successfully. Moreover, the time
The conventional method
The proposed method
Without compensation

Fig. 14. The force response of the flexible robot system

(a) Motor force with resonance ratio control
(b) Motor force without compensation
(c) Load force with resonance ratio control
(d) Load force without compensation

(a) Motor position
(b) Motor velocity
(c) Load position
(d) Load velocity

Fig. 15. The motion movements of the flexible robot system
response of the controlled closed loop system has a small settling time and the system is high robustness against the parameter variations and disturbance effects.

7 CONCLUSION

In this paper, a new resonance ratio gain has been presented as 2.0 for the rejection of vibration in the flexible robot system. The proposed gain controller based on the CDM is simple to design and easily implemented into the resonant ratio control. By using the proposed controller design based on CDM method, the best results for the fast response and vibration suppression in the flexible robot system are also achieved. Current work also includes the design of the MEDOB to estimate the external force on the load side. No feedback force sensors are required in the system. By using the proposed MEDOB, it is not necessary to estimate the nominal spring coefficient.

A new parameter identification has been presented for the consideration of the correct value of stiffness coefficient in the resonant ratio control. The proposed identification based on the LDOB and MEODB is simple to design and easily implemented into the system. Analysis of the calculation reveals that the correct region of spring coefficient is bounded if the differentiation of load side external force between both LDOB and MEDOB is zero. From the experimental and simulation results, the correct value of stiffness coefficient is obtained and the load side external force from LDOB can be tracked almost identically over the load side external force from MEDOB. These results serve as guidelines to identify the stiffness of the unknown environment parameters. For future works, it might be useful to apply and develop a model of environment for haptic technology, advanced medical device and telerobotic research.

ACKNOWLEDGMENT

This research was supported by a grant-in-aids from Japan Society for the Promotion of Science (JSPS) under the JSPS Postdoctoral Fellowship.

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Received: 2012-06-30
Accepted: 2012-10-15