Controller Design for Performance Improvement of Two-Mass Resonance Systems

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Abstract — Torsional vibration is an important matter in two-mass resonance systems, such as robots. If the rate of load inertia/motor inertia and the shaft stiffness are small mechanical vibrations will easily be done. A two-mass resonance system is being considered in this article. Controlling the rate of resonance is one of the effective methods in controlling the two-mass systems and reducing the rate of vibration. Two-mass systems structures, connected as two masses by a spring, could be used in robots or machine tools. Machines components, such as gears and shafts could limit the relevant cycle and stick-slip.

I. INTRODUCTION

Today, two-mass systems are used a lot in industry, e.g., rolling mill, robot arms and textile machine. Measuring systems variables is difficult in mechanical resonance systems. When there are no variables that could be measured, the designing of the system controllers could be difficult. Description of this method is observed in [1].

The simplest method for multi-mass resonance systems is the two-mass system. There have been different controllers for the two-mass systems created, among which the Haptic controller, optimal control, classical control, approximations and system identification, resonance ratio control could be indicated [4-8].

In this article, a suitable controller with disturbance observer is given for a two- mass system that is load vibration, friction and limit cycle due to nonlinear existing elements in the system. The load distortion is estimated by the disturbance observer and the load distortion observer is described in [9].

We are going to design the controlling parameters. The parameters proposed for the system should be appropriate where the load varies and where speed and accuracy are considerable and important. Finally, the result of simulating a robot arm submitted, to show the effectiveness of the proposed controller.

II. TWO-MASS MICHANICAL SYSTEM

Long shaft and axles are inclined for vibration that could cause problems in the control system and unstable the system.

The general structure of a two-mass system is shown in fig. 1. The two masses of J_1 and J_2 are connected to each other by a shaft and a gear. The input drive torque m_4 of the system is mechanical that is considered with regards to the defined input m_4. T_M is a mechanical torque. The difference of α_1 and α_2 angles m_w to be dependent on the shaft reaction. The torque m_R is created by the friction and the torque m_L could be formed by the additional load. $n_1$and n_2 indicate the drive and load speed.

In describing the two-mass mechanical system, the nonlinear state-space is given by:

$$\dot{X} = A(X) + B(X)U \quad (1)$$

$$Y = C(X) \quad (2)$$

where:

$$X = \begin{bmatrix} n_1 \\ a_1 \\ n_2 \\ a_2 \end{bmatrix}$$

$$A(X) = \begin{bmatrix} -\frac{c}{J_1} P(a_1 - a_2) - m_R(n_1) \\ n_1 \\ \frac{c}{J_1} P(a_1 - a_2) - m_R(n_2) \\ n_2 \end{bmatrix}$$

$$B(X) = \begin{bmatrix} \frac{1}{J_1} \\ 0 \\ 0 \\ 0 \end{bmatrix}$$

$$C(x) = a_2$$

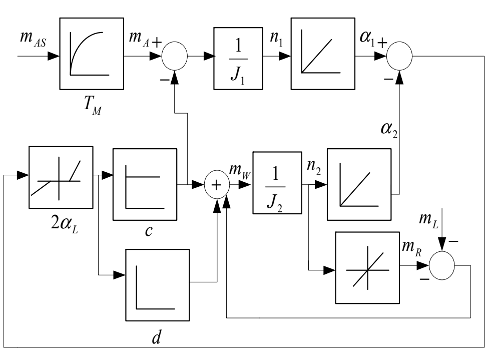

Fig. 1. Two-mass system general structure

III. LIMIT CYCLE BY THE NONLINEAR ELEMENT

In case a nonlinear element, such as friction or saturation, exists in the system, we will therefore sometimes have a constant vibration and stable frequency. This vibration is called "limit cycle". The full description of this method is in [10]. Hereby, we merely explain that if N(x) is the descriptive function of the nonlinear element and F(jω) is the descriptive function of the linear part of the system and if F(jω) and -1/N(x) have an intersection on a gain-phase graph, a limit cycle will be found by the frequency and amplitude a similar to the point of intersection. If there is no point of intersection, then the limit cycle does not occur.

The nonlinear function of N(x) and the linear element of F(jω) are described in fig. 2. When limit cycle occurs, the nonlinear element of e(t) will be in the form of e(t)=Asinωt.

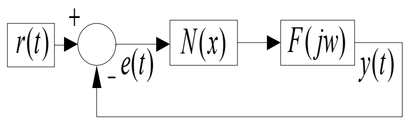

Fig. 2. Feedback control system with nonlinear element.

At this time we consider the input as "zero". (r(t)=0), Hence:

$$N(x)F(j\omega)e(t) = -e(t) \quad (3)$$

$$-\frac{1}{N(x)} = F(jw) \quad (4)$$

A simple model of two-mass system is shown in fig. 3. ω, D and k are angular velocity, viscosity and stiffness, respectively.

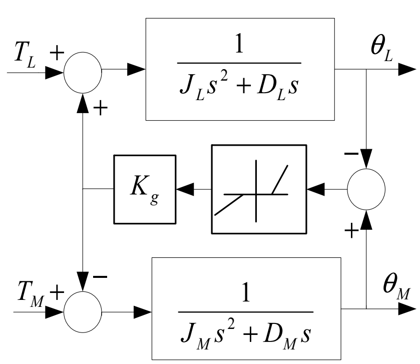

Fig. 3. A simple mode of two-mass system.

The frequency for resonance and anti-resonance are calculated as follows:

$$w_r = \sqrt{\frac{k_g}{J_L}\left(1 + \frac{J_L}{J_M}\right)} \quad (5)$$

$$w_a = \sqrt{\frac{k_g}{J_L}} \quad (6)$$

In resonance frequency w_r, we have:

$$R = \frac{J_L}{J_M} \quad (7)$$

The amount of resonance H is defined as follows:

$$H = \frac{w_r}{w_a} \quad (8)$$

The motor inertia is obtained as follows:

$$J_{MO} = \frac{J_M}{K} \quad (9)$$

Hence, by putting (9) in (5) and (7) the resonance frequency and the rate of inertia are changed as follows:

$$w_{ro} = \sqrt{\frac{K_g}{J_L}\left(1 + \frac{J_L}{J_{MO}}\right)} \quad (10)$$

$$Ro = \frac{J_L}{J_{MO}} = KR \quad (11)$$

The anti-resonance frequency does not change and hence resonance ratio (8) is changed, in the following from.

$$Ho = \sqrt{1 + R_o} = \sqrt{1 + \frac{J_L}{J_{MO}}} = \sqrt{1 + KR} \quad (12)$$

The amount of new resonance alters from 2 to $\sqrt{5}$. Hence the new resonance is hereto changed.

IV. CONTROLLER DESIGN

In this article, two controllers are designed for two systems. The specification of systems 1 and 2 are shown in table I.

Table I. The systems parameters

Parameters	System 1	System 2
D_M	0.0	8.3
D_L	0.0	0.0
J_M	0.87	0.4
J_L	0.12	1.4
K_g	1.12	22601
W_r	3.26	26905
W_a	3.1	12701

The speed controller for system 1 is designed according to resonance ratio control and the situation controller, for system 2 is designed according to the disturbance observer and the PD controller. Fig. 4 shows the controller situation with the disturbance observer the used functions in fig. 4 are calculated as follows:

$$C_b(s) = K_1 \frac{g_s}{s + g_s} + K_2 \quad (13)$$

$$C_f(s) = s^2 + k_1 s + K_2 \quad (14)$$

$$P_n(s) = \frac{1}{J_n s^2} = \frac{1}{(J_M + J_L)s^2} \quad (15)$$

For the step disturbance observer:

$$G(s) = \frac{g_2}{s^2 + g_1 s + g_2} \quad (16)$$

For the ramp disturbance observer:

$$G(s) = \frac{g_2 s + g_3}{s^3 + g_1 s^2 + g_2 s + g_3} \quad (17)$$

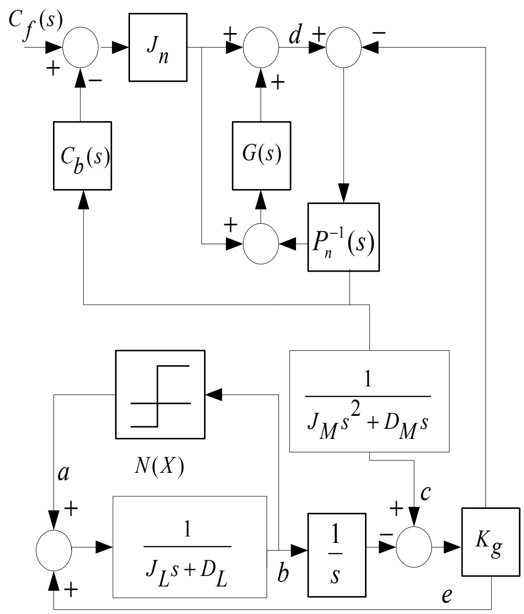

Fig. 4. Situation of the controller

The transmission function of F(s)=b/a in Fig. 4 is calculated as follows. Parameters "a", "b", "c", "d" and "e" are defined in Fig. 4.

$$c = \frac{1}{J_M s^2 + D_M s}(d - e) \tag{18}$$

In the formula 18, "d" and "e" are calculated as below.

$$d = -C_b J_n c + G(d - P_n^{-1} c) \tag{19}$$

$$e = Kg\left(-\frac{b}{s} + c\right)$$

b is also calculated as follows:

$$b = \frac{1}{J_L s + D_L}(a + e) \tag{20}$$

By replacing the formula 20 in 19 and its results in 18, "c' is calculated as follows:

$$c = \frac{\frac{Kg}{s}}{Kg - F_1 + J_m s^2 + D_m s} b(s) \tag{21}$$

$$(J_L s + D_L + \frac{kg}{s}(1 - \frac{kg}{F_2}))b = a \Rightarrow F = \frac{b}{a}$$

$$F_2 = kg - F_1 + J_m s^2 + D_M s$$

$$F_1 = \frac{C_b J_n + G p_n^{-1}}{G - 1}$$

Fig. 5 shows the gain-phase graph for two different disturbance observers (step and ramp). Hence, the lack of limit cycle means that the angel F(s) is less then 300° degrees for all the frequencies.

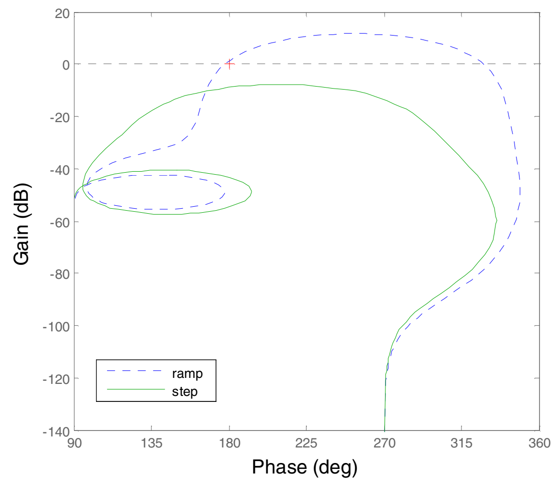

Fig. 5. Gain-phase plot for disturbance observer.

V. Simulation Results

Tables II and III show the controller parameters for system 1 and 2.

Table II. Controller parameters of system 1

Parameters	Proposed
K_p	2.92
K_i	4.61
K	6.53
g_1	4.58

Table III. Controller parameters of system 2

Disturbance Observer	Step	Ramp
K_1	46	46
K_2	1342	409
J_n	1025	1043
g_1	72	86
g_2	1289	2857
g_3	-	12346
g_s	86	11

K_p in table II Is the proportional controller gain and K_i is the integral controller gain. In table 3, k_2 is the element of phase regulation for the regulator speed. k_1 is the conforming element that is used by the disturbance observer in order to indicate the effect of lowering the disturbance. J_n is a sum of motor inertia (J_M) and other inertias.

Fig. 6 is the simulation of the responding results of speed for system 1, with the parameters shown in table II. The reference speed is 1.1 rad/sec in 6 sec and the load disturbance is T_L=-0.52 Nm in 26 sec.

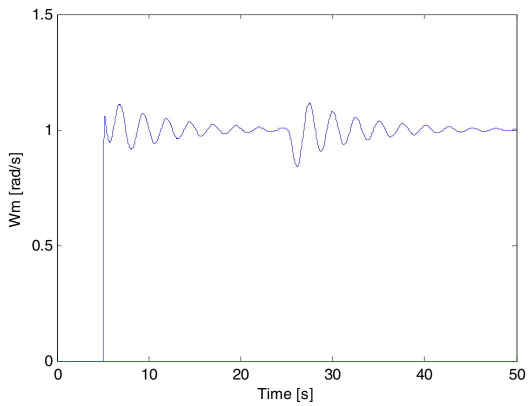

Fig. 6-a. Speed of system 1 with the controller.

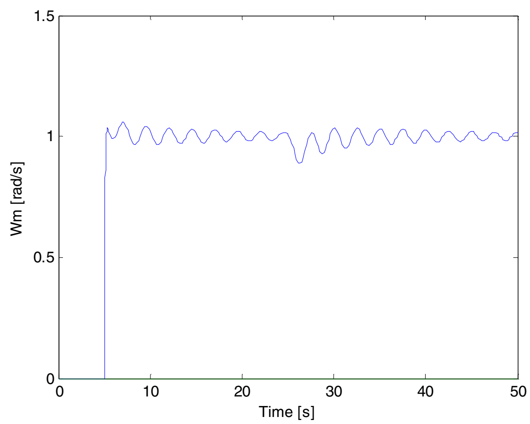

Fig. 6-b. Speed of system 1 without the controller.

As observed in figs 6-a and 6-b, the system with the designed controller in this article will converge to zero faster, in comparison with the system without the controller.

Fig. 7 and fig. 8 are for parameters for system 2, by using table III. Fig. 7 shows the control results for system 2 by considering the designed controller and without considering the controller, where the limit cycle occurs for the ramping disturbance observer in fig. 5. Fig. 8 is also the results of the control situation for system 2 for the stepping disturbance observer. As it can be seen in fig. 7 and Fig. 8 the error angle is improved by the designed controller, as compared to the situation without the controller.

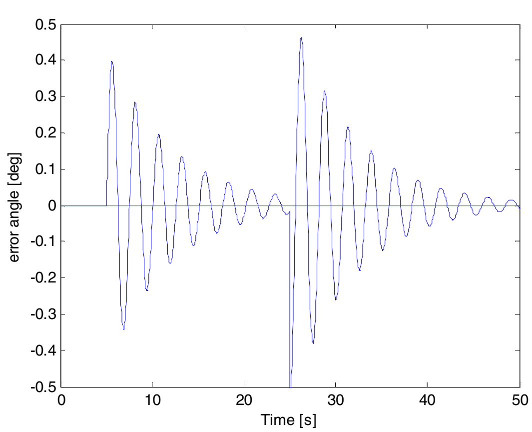

Fig. 7-a. The results of ramping disturbance observer of system 2 with the controller

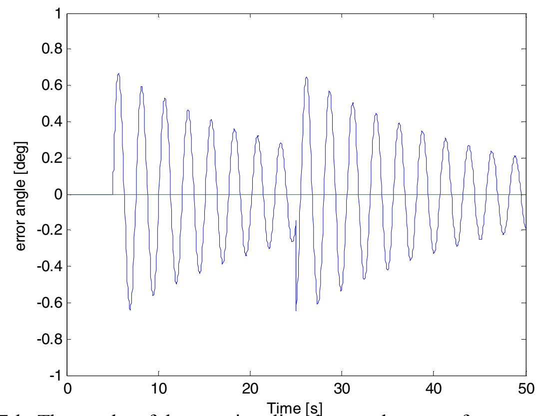

Fig. 7-b. The results of the ramping disturbance observer of system 2 without the controller.

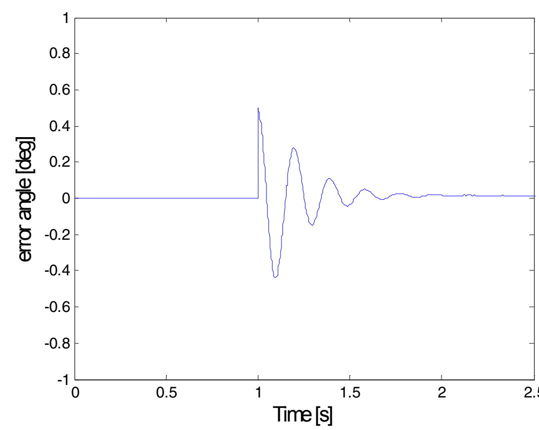

Fig. 8-a. The results of the stepping disturbance observer of system 2 whit the controller

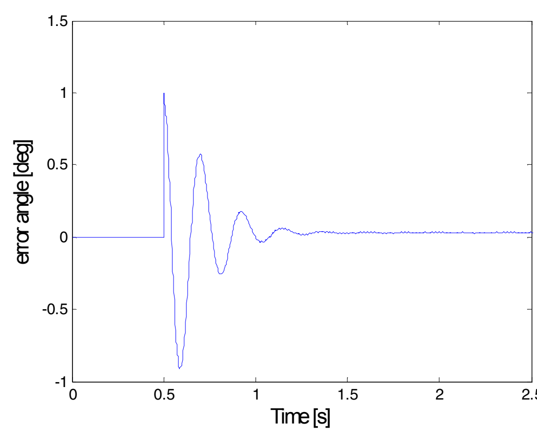

Fig. 8-b. The results of the stepping disturbance observer of system 2 without the controller.

VI. Conclusion

The example of two-mass systems are the robot arm that have been, in recent years, replaced the human beings hands in industry and the space station. Hence, it is necessary that they act similar to human beings hands. The problem is the vibration of the robot hand and for this case and this problem, the two-mass resonance system vibration control is studied. Among the problems of two-mass system one nonlinear friction and limit cycle. In this article, the controllers designed for two systems have resulted in improving the above problems. The shafts and axles in two-mass resonance system tend to vibrate and this problem could make the system unstable.

By controlling the resonance rate, we could reduce the vibrations of the two-mass system. The controlling parameters are designed for two-mass systems to improve the efficiency of the controller for changing of the load and preventing the limit cycle due to nonlinearity.

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