

Full Length Research Paper

Reducing torsional oscillation and performance improvement of industrial drives using PI along with additional feedbacks

T. Ananthapadmanabha^{1*}, A. D. Kulkarni¹, Benjamin A. Shimray¹, R. Radha¹
and Manoj Kumar Pujar²

Department of Electrical and Electronics Engineering, The National Institute of Engineering, Mysore-08, Karnataka, India.

²Manoj Kumar Pujar Bangalore Transmission Zone, KPTCL Karnataka India.

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Torsional oscillations in electrical drive systems with elastic shafts are a well known problem. In the industrial drive systems, a shaft torsional vibration or oscillation is often generated when a motor and a load are connected with a flexible shaft. Large inertias of the motor and load side and a long shaft create an elastic system. The motor speed is different from the load side and the shaft undergoes large torsional torque. In order to damp the torsional vibration effectively, the application of various feedbacks are necessary. The control structures of electrical drives working in the industry are usually based on linear PI controllers. This paper presents the design, analysis and comparison of the conventional PI-control to control structure using PI supported by various feedbacks for performance improvement of the Industrial drives system.

Key words: Torsional oscillation, 2- mass drive system, industrial drive system, transient response, mechanical elasticity, vibration suppression.

INTRODUCTION

Without Drive control systems there could be no manufacturing, no vehicles, and no material handling. Modern industrial drive systems require relatively high dynamic properties. When the industrial drives are designed, the elasticity of the shaft is neglected. In the case of standard drive and low power servos such an assumption is reasonable; however, there is a large group of high power drives and other application such as rolling-mill drives, robot arms, servo systems, textile drives, throttle systems, conveyor belts, and deep-space antenna drives where characteristics of the mechanical part have to be included in the analysis (Szabat and Orłowska, 2007; Szabat and Orłowska, 2008; Zhang and Furusho, 2000), and the shaft elasticity must be taken into consideration. This type of assumption (neglecting elasticity) can lead to damaging oscillations (Valenzuela et al., 2005; Sugiura and Hori, 1996; Ji and Sul, 1995). This oscillation or torsional vibration decreases the

product quality and system reliability; the system can even lose stability.

The control problem of the two-mass system is originally derives from rolling-mill drives (Zhang and Furusho, 2000; Zhang, 1999; Valenzuela et al., 2005). Large inertias of the motor and rolls and a long shaft makes the drive system an elastic system. The non ideal characteristics of the shaft worsen the performance in practical industrial drive system. An analogous problem also appears in the paper and textile industry (Valenzuela et al., 2005) and rolling mills (Zhang et al., 2007).

To suppress the torsional oscillations, different control structures have been developed (Rached et al., 1994). The simplest method to avoid the system state variable oscillations till now relies on decreasing the dynamics of the control structures. However, this approach neglects the performance of the drive and is hardly ever utilized. If desirable control system performances are required, the application of the additional feedbacks from a selected state variable is necessary ((Szabat and Orłowska, 2007; Szabat and Orłowska, 2008).

A technique that can improve system performance

*Corresponding author. E-mail: drapn2008@yahoo.in.

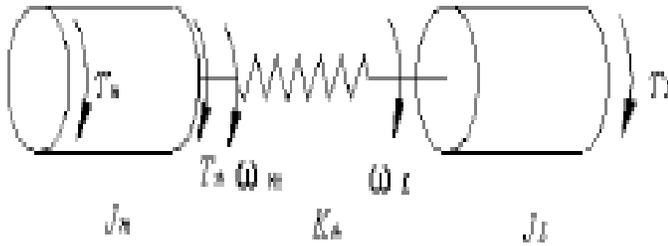


Figure 1. 2-mass system. Where: ω_m : motor speed, ω_l : load speed; T_m : motor torque; T_l : load torque; T_{sh} : shaft torque; J_m : motor inertia; J_l : load inertia; K_{sh} : spring coefficient (stiffness) of drive shaft.

exploit alternative tuning techniques for the classical cascade control structure (with a PI speed controller and the basic feedback from the motor speed), based on a suitable location of the close loop system poles. Three different pole locations with identical radius, damping coefficient, and real part were presented (Zhang and Furusho, 2000). Suggestions for the application of a proportional–integral–derivative controller are also made. But the derivative part D increased the inertia ratio of the system and virtually decreasing the moment of inertia of the motor. To improve the performances of industrial drive system, additional feedback loop from one selected state variable can be used. The additional feedbacks can be inserted to the electromagnetic-torque control loop or the speed control loop. Another modification of the control structure results from inserting the additional feedback from the shaft torque. This type of feedback was applied in (O’Sullivan et al., 2006). The damping of the torsional vibration is reported to be successful. This structure is less sensitive to measurement noises than the previous one since the derivative of the shaft torque does not exist. Use of additional feedback from the derivative of the load speed was proposed in (Zhang, 1999). This results in the same dynamical performance as for the previous control structure.

In recent development, nonlinear and soft computing control methods have attracted much attention. The application of the sliding or the fuzzy control will increase the robustness of the drive system to parameter variations. These techniques can allow obtaining better dynamical characteristics of the system, as compared to the classical ones, but they are not yet popular in industrial applications.

Control structures of electrical drives working in almost all the present industry are usually based on linear PI controllers. Thus, the main goal of this paper is to undergo a systematic analysis and present design guidelines for the speed control structures of the two-mass system using PI speed controller that is supported by different additional feedbacks. This work is similar to work carried out by Krzysztof Szabat and Teresa Orłowska-Kowalska (Szabat and Orłowska, 2007; Szabat and Orłowska, 2008) but the method of analysis and

results extracted are different.

MODELING OF DRIVE SYSTEM

Even though the main drive system used in industrial systems are complex multi-mass system, in many cases, it can be roughly modeled by a two-mass system considering only the first resonant mode as shown in Figure 1.

The damping of the 2-mass system due to the friction is very small, so that it can be neglected without affecting the analysis accuracy. The following Laplace transfer functions are derived from the dynamic mechanical principles:

$$\omega_m = \frac{1}{J_m s} (T_m - T_{sh}) \tag{1}$$

$$\omega_l = \frac{1}{J_l s} (T_{sh} - T_l) \tag{2}$$

$$T_{sh} = \frac{K_{sh}}{J_l s} (\omega_m - \omega_l) \tag{3}$$

The stated equation of the system is given by

$$\overline{X} = A X + B U + E T_L \tag{4}$$

Where:

$$X = [\omega_m \quad \omega_l \quad \omega_{sh}]^T$$

$$A = \begin{bmatrix} 0 & 0 & -\frac{1}{J_m} \\ 0 & 0 & \frac{1}{J_l} \\ K_{sh} & -K_{sh} & 0 \end{bmatrix}$$

$$B = \begin{bmatrix} \frac{1}{J_m} \\ 0 \\ 0 \end{bmatrix}$$

$$C = [1 \quad 0 \quad 0]$$

$$E = \begin{bmatrix} 0 \\ \frac{1}{J_l} \\ 0 \end{bmatrix}$$

$$U = T_m$$

To simplify the comparison of the dynamical performances of the drive systems of different powers, the mathematical model can be expressed in a per-unit system, using the following notation as new state variables:

$$\omega_1 = \frac{\Omega_1}{\Omega_N} \quad \omega_2 = \frac{\Omega_2}{\Omega_N}$$

$$m_e = \frac{M_e}{M_N} \quad m_s = \frac{M_s}{M_N} \quad m_L = \frac{M_L}{M_N}$$

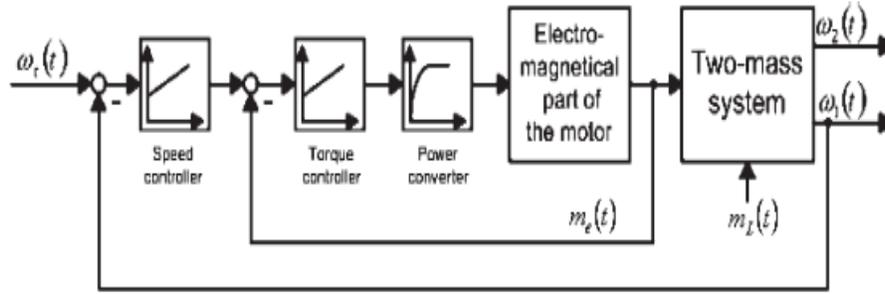


Figure 2. Classical control structure.

Where Ω_N is the nominal speed of the motor; M_N is the nominal torque of the motor; ω_1 and ω_2 are the motor and load speeds, respectively; m_e , m_s , and m_L are the electromagnetic, shaft, and load torques in the per-unit system, respectively. The mechanical time constant of the motor T_1 and the load machine T_2 are, thus, given as

$$T_1 = \frac{\Omega_N J_1}{M_N} \quad T_2 = \frac{\Omega_N J_2}{M_N}$$

The stiffness time constant T_c and internal damping of the shaft d can be calculated as follows:

$$T_c = \frac{M_N}{K_c \Omega_N} \quad d = \frac{\Omega_N D}{M_N}$$

Where,

D is the internal damping of the shaft.

The analyzed system is described by the following state equation (in per unit system):

$$\frac{d}{dt} \begin{bmatrix} a_1(t) \\ \omega_1(t) \\ \omega_2(t) \end{bmatrix} = \begin{bmatrix} 0 & 0 & \frac{1}{T_1} \\ 0 & 0 & \frac{1}{T_2} \\ \frac{1}{T_c} & \frac{1}{T_c} & 0 \end{bmatrix} \begin{bmatrix} a_1 \\ \omega_1 \\ \omega_2 \end{bmatrix} + \begin{bmatrix} \frac{1}{T_1} \\ 0 \\ 0 \end{bmatrix} [T_m] + \begin{bmatrix} 0 \\ \frac{1}{T_2} \\ 0 \end{bmatrix} [T_L] \quad (5)$$

Where:

- ω_1 : motor speed;
- ω_2 : load speed;
- T_m : motor torque;
- T_{sh} : shaft (torsional) torque;
- T_1 : disturbance torque;
- T_1 : mechanical time constant of the motor;
- T_2 : mechanical time constant of the load machine;
- T_c : stiffness time constant.

PROPOSED CONTROL STRUCTURE

General description

A typical electrical drive system is composed of a power-converted motor coupled to a mechanical system; microprocessor-based speed and torque controllers, and current, speed, and/or position sensors used for feedback signals. The diagram of such system is presented in Figure 2.

The inner control loop performs motor torque regulation and consists of a power converter, the electromagnetic part of the motor and current sensor and respective current or torque controller. This control loop is designed to provide sufficiently fast torque control, so it can be approximated by an equivalent first-order term. The outer control loop consists of the mechanical part of the drive, speed sensor, and speed controller, and is cascaded to the inner torque control loop. It provides speed control according to its reference value.

Suitable oscillation damping of the two-mass system can be obtained using different additional feedbacks. The block diagram of the drive system with a simplified inner loop and additional feedbacks is presented in Figure 3. In a typical industrial drive internal damping coefficient d of the shaft has a very small value and, therefore, will be neglected in the further analysis.

Three additional feedbacks k_2 , k_6 , and k_7 , which were not mentioned in the literature, were introduced, that is, the feedback from the derivative of the speed difference ($\omega_1 - \omega_2$) in group A, the feedback from the load speed in group B, and the feedback from the derivative of the shaft torque in group C. The control structures were divided into three different groups according to their dynamical characteristic (Szabat and Orłowska, 2006). The link between different feedbacks (in every group) can be found out from Figure 3. The relationship can be directly seen between feedbacks k_4 and k_5 in group B: The derivative of the shaft torque is simply the difference between the motor and load speeds multiplied by the stiffness coefficient. The same relationship exists between the feedbacks k_7 and k_8 in group C. The last feedback k_9 is based on the motor and load speeds. The link between feedbacks k_1 and k_2 is not so clearly seen in group A. But, if the electromagnetic and load torques are neglected, the derivative of the difference speeds is the shaft torque multiplied by the following coefficient: $d(\omega_1 - \omega_2)/dt = -ms(1/T_1 + 1/T_2)$ (Szabat and Orłowska, 2007).

The closed-loop transfer functions from the reference speed to the motor and load speeds, respectively, for the control structure demonstrated in Figure 3, are given by the equations (6) and (7) (Szabat and Orłowska, 2007), with the assumption that the optimized transfer function of the electromagnetic-torque control loop is equal to 1. The close loop transfer can be obtained by using signal flow graph (SFG) and Mason's gain formula.

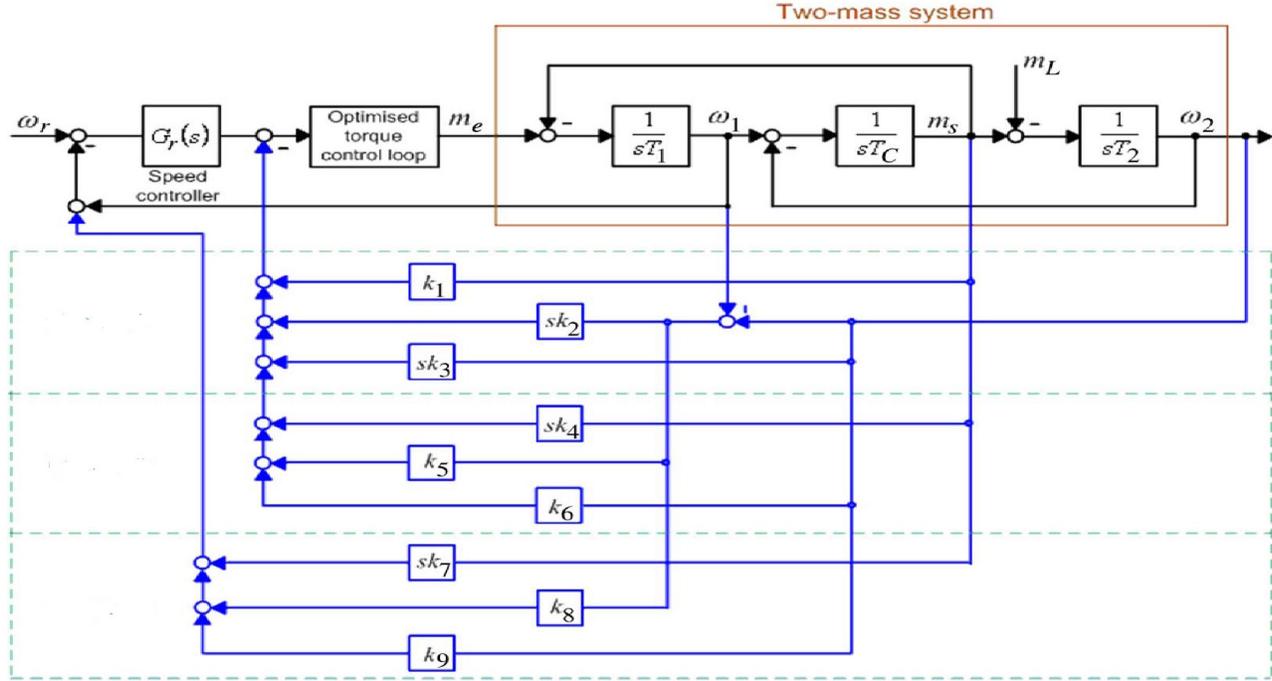


Figure 3. Control structure with different additional feedbacks.

$$G_{a_1}(s) = \frac{a_1(s)}{a_r(s)} = \frac{G_r(s)(s^2 T_2 T_c + 1)}{s^3 T_2 T_c (T_1 + K_2) + s^2 T_2 (G_r(s) T_c + G_r(s) K_7 + G_r(s) T_c K_8) + s(T_1 + T_2(1 + K_1 + s K_4 + s T_c K_5) + K_3) + G_r(s)(1 + K_9) + K_6} \quad (6) \text{ and } (7)$$

$$G_{\omega_2}(s) = \frac{\omega_2(s)}{\omega_r(s)} = \frac{G_r(s)}{s^3 T_2 T_c (T_1 + K_2) + s^2 T_2 (G_r(s) T_c + G_r(s) K_7 + G_r(s) T_c K_8) + s(T_1 + T_2(1 + K_1 + s K_4 + s T_c K_5) + K_3) + G_r(s)(1 + K_9) + K_6}$$

Where:

$$G_r(s) = K_p + K_I \frac{1}{s} \quad (8) \quad s^4 + s^3(4\xi\omega_o) + s^2(2\omega_o^2 + 4\xi^2\omega_o^2) + s(4\xi\omega_o^3) + \omega_o^4 = 0 \quad (11)$$

is the transfer function of the PI controller.

Through the comparison of relationships (9) and (11), the set of four equations is created, that is

Cascade control structure without additional feedbacks

At first, the control structure without additional feedback as in Figure 4 was considered. The characteristic equation of the analyzed system is given by

$$s^4 + s^3 + s^2 \left(\frac{K_I}{T_1} + \frac{1}{T_1 T_c} + \frac{1}{T_2 T_c} \right) + s \left(\frac{K_p}{T_1 T_2 T_c} \right) + \frac{K_I}{T_1 T_2 T_c} = 0 \quad (9)$$

The desired polynomial of the system has the following form:

$$\left(s^2 + 2\xi\omega_o + \omega_o^2 \right) \left(s^2 + 2\xi\omega_o s + \omega_o^2 \right) = 0 \quad (10)$$

Where ξ is the damping coefficient and ω_o is the resonant frequency of the closed-loop system.

$$4\xi\omega_o = \frac{K_p}{T_1}$$

$$2\omega_o^2 + 4\xi^2\omega_o^2 = \frac{K_I}{T_1} + \frac{1}{T_1 T_c} + \frac{1}{T_2 T_c}$$

$$4\xi\omega_o^3 = \frac{K_p}{T_1 T_2 T_c}$$

$$\omega_o^4 = \frac{K_I}{T_1 T_2 T_c}$$

(12)

Solving the equation set (12), the parameters of the system, that is, damping coefficient ξ and resonant frequency ω_o , as well as the controller parameters, that is, K_p and K_I are obtained as follows Equation 10 can be rewritten as follows:

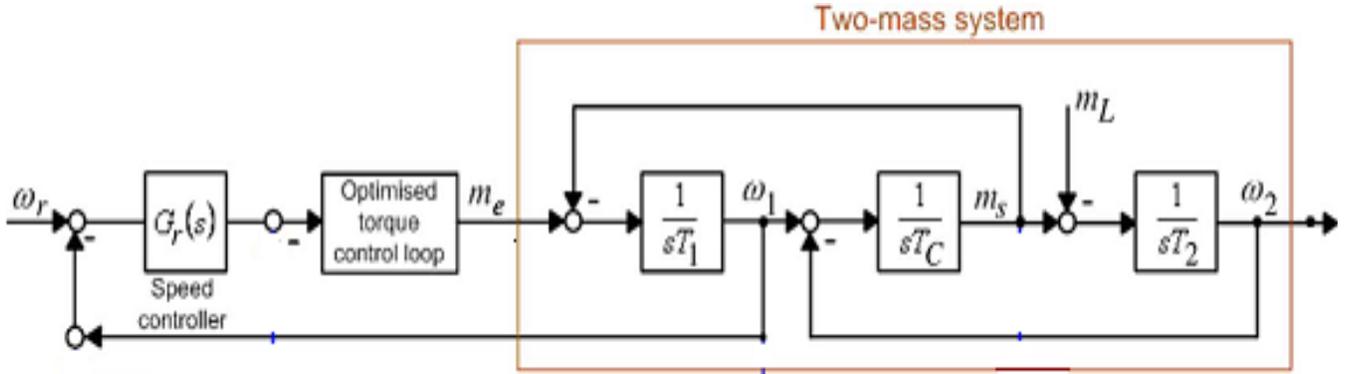


Figure 4. Control structure without additional feedback.

$$\xi = \frac{1}{2} \sqrt{\frac{T_1}{T_2}} \quad \omega_o = \sqrt{\frac{T_1}{T_2 T_c}} \quad (13)$$

$$K_P = 2 \sqrt{\frac{T_1}{T_c}} \quad K_I = \frac{T_1}{T_2 T_c}$$

Control structures with additional feedbacks K₁, K₂ and K₃

This group includes the modified control structures with additional feedbacks from the shaft torque k₁, from the derivative of the difference between the motor and load speeds k₂, or from the derivative of the load speed k₃.

First, the control structure with additional feedback K₁ was investigated. The damping coefficient and resonant frequency of this structure with the PI speed controller are the following:

$$\xi^{K_1} = \frac{1}{2} \sqrt{\frac{T_2(1+K_1)}{T_1}} \quad \omega_o^{K_1} = \sqrt{\frac{1}{T_2 T_c}} \quad (14)$$

Similarly the damping coefficient and resonant frequency of the second System, with additional feedback from the derivative of the difference between two speeds k₂, are

$$\xi^{K_2} = \frac{1}{2} \sqrt{\frac{T_2 - K_2}{T_1 + K_2}} \quad \omega_o^{K_2} = \sqrt{\frac{1}{T_2 T_c}} \quad (15)$$

For the next control structure with additional feedback from the derivative of the load speed k₃, the damping coefficient and resonant frequency are

$$\xi^{K_3} = \frac{1}{2} \sqrt{\frac{T_2 + K_3}{T_1}} \quad \omega_o^{K_3} = \sqrt{\frac{1}{T_2 T_c}} \quad (16)$$

The following equations allow setting the parameters of the feedback loop and the speed controller (Szabat and Orlowska, 2007).

$$K_1 = \frac{4\xi_r^2 T_1}{T_2} - 1 \quad K_P^{K_1} = 2 \sqrt{\frac{T_1(1+K_1)}{T_c}}$$

$$K_2 = \frac{T_2 - 4\xi_r^2 T_1}{4\xi_r^2 T_1 + 1} \quad K_I^{K_1} = \frac{T_1}{T_2 T_2}$$

$$K_P^{K_2} = 2 \sqrt{\frac{(T_1 + K_2)(T_2 - K_2)}{T_2 T_c}} \quad K_P^{K_3} = 2 \sqrt{\frac{T_1(T_2 - K_3)}{T_c}} \quad (17)$$

$$K_P^{K_3} = \frac{T_1 + K_2}{T_2 T_c} \quad K_I^{K_3} = \frac{T_1}{T_2 T_2}$$

$$K_3 = 4\xi_r^2 T_1 - T_2$$

In the three mentioned structures, the application of additional feedback (k₁, k₂, or k₃) increases the damping coefficient of the drive system, yet the resonant frequency remains unchanged [see, e.g., (14), (15), and (16)].

Control structures with additional feedbacks K₄, K₅ and K₆

Next, the control structures with additional feedbacks from the derivative of torsional torque k₄, the difference between motor and load speeds k₅, or the load speed k₆, inserted to the torque node, were tested one after another.

As in previous case, the damping coefficient and the resonant frequency of the system with additional feedback from the derivative of the shaft torque k₄ are:

$$\xi^{k_4} = \sqrt{\frac{T_c + x}{4T_1 T_c} (T_1 + T_2) + \frac{T_c}{4(T_c + x)}} \frac{1}{2} \quad \omega_o^{k_4} = \sqrt{\frac{1}{T_2(T_c + x)}}$$

Where:

$$x_{1,2} = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}$$

$$a = T_2^2 (T_2 + T_2)$$

$$b = 2T_2^3 T_c - 4\xi_r^2 T_1 T_2^2 T_c$$

$$c = T_2^3 T_c - 4\xi_r^2 T_1 T_2^2 T_c$$

Then, the control structure with additional feedback from the difference between the motor and load speeds k₅ was investigated. The damping coefficient and the resonant frequency of the analyzed system are:

$$\xi^{k5} = \sqrt{\frac{(T_c+x)(T_1+T_2)}{4T_1} + \frac{1}{4(T_c+x)^2}} \frac{1}{2} \quad \omega_o^{k5} = \sqrt{\frac{1}{T_2 T_c (1+x)}}$$

The following equations allow setting the parameters of the feedback loop and the speed controller (Szabat and Orlowska, 2007).

$$\begin{aligned} K_4 &= xK_P^{K4} & K_P^{K4} &= \frac{4\xi_r \omega_0^{K4} T_1 T_c}{T_c + x} & K_I^{K4} &= (\omega_o^{K4})^4 T_1 T_2 T_c \\ K_5 &= xK_P^{K5} & K_P^{K5} &= \frac{4\xi_r \omega_0^{K5} T_1}{1+x} & K_I^{K5} &= (\omega_o^{K5})^4 T_1 T_2 T_c \\ K_6 &= xK_P^{K6} & K_P^{K6} &= \frac{4\xi_r \omega_0^{K6} T_c}{4\xi_r \omega_0^{K6} T_c} & K_I^{K6} &= (\omega_o^{K6})^4 T_1 T_2 T_c \end{aligned} \quad (18)$$

Control structures with additional feedbacks: K_7 , K_8 and K_9

In this case, control structure with additional feedbacks from the derivative of shaft torque k_7 , the difference between the load and motor speeds k_8 , or the load speed k_9 . Unlike the previous two groups, the additional feedbacks are inserted to the speed node were investigated.

First, the control structure with additional feedback from the derivative of the torsional torque k_7 was examined. The system damping coefficient and resonant frequency are

$$\xi^{k7} = \frac{1}{2} \sqrt{\frac{(T_c+x)(T_c+K_7)}{T_1 T_c}} \quad \omega_o^{k7} = \sqrt{\frac{1}{T_2 (T_c+K_7)}}$$

Next, the control structure with additional feedback from the difference between motor and load speed k_8 was tested. The damping coefficient and resonant frequency are:

$$\xi^{k8} = \frac{1}{2} \sqrt{\frac{T_1 K_8 + T_2 (1+K_8)}{T_1}} \quad \omega_o^{k8} = \sqrt{\frac{1}{(1+K_8) T_2 T_c}}$$

Finally, the system with additional feedback from the load speed k_9 was considered. The damping coefficient and resonant frequency of this system are defined as

$$\xi^{k9} = \frac{1}{2} \sqrt{\frac{T_1 + T_2}{T_1 (1+K_9)}} \quad \omega_o^{k9} = \sqrt{\frac{1+K_9}{T_2 T_c}}$$

The following equations allow setting the parameters of the feedback loop and the speed controller (Szabat and Orlowska, 2007).

$$\begin{aligned} K_7 &= \frac{(4\xi_r^2 + 1) T_1 T_c}{T_1 + T_c} - T_c & K_8 &= \frac{(\xi_r^2 4T_1) - T_2}{T_1 + T_c} & K_9 &= \frac{(T_1 + T_2)}{T_1 (4\xi_r^2 + 1)} - 1 \\ K_P^{K7} &= 4\xi_r (\omega_0^{k7})^3 T_1 T_2 T_c & K_P^{K8} &= \frac{4\xi_r \omega_0^{K8} T_1}{1+K_8} & K_P^{K9} &= 4\xi_r \omega_0^{K9} T_1 \\ K_I^{K7} &= (\omega_0^{k7})^4 T_1 T_2 T_c & K_I^{K8} &= \frac{T_1}{(1+K_8) T_2 T_c} & K_I^{K9} &= \frac{T_1 (1+K_9)}{T_2 T_c} \end{aligned} \quad (19)$$

SIMULATION RESULTS AND DISCUSSION

The performance parameters or time domain transient response specifications are presented in details in Tables

1 and 2. For simulation purpose, the required damping co-efficient is taken as 0.7 that is $\xi_r = 0.7$. A 5.5 KW Induction motor connected through a long shaft to a 6.4 KW servo induction motor which can induce disturbance torque given in Table 3 is taken for simulation purpose. The control structure is simulated in MATLAB/SIMULINK. Various control structures simulation results are presented in Figures 5 a - j.

Firstly, the control structure of a PI controller without any additional feedback was investigated. The load speed transient has a large overshoot and quite a long settling time. Next, the control structure using PI with various additional feedbacks k_1 to k_9 considering one feedback at a time was simulated. The detail effect of using various feedbacks can be seen from the simulated results presented in Tables 1 and 2. With additional feedbacks k_1 , tremendous improvement in settling time is achieved on the load side transient. Overshoot is also reduced to 54.2 from 99.9%. Using feedback K_2 similar improvement is achieved; settling time is reduced to 4.7 s while overshoot is reduced to 27.9%.

In case where PI is supported by feedback K_4 , we observed that both the load speed and motor speed transient exhibit critically damped response with very small overshoot of 0.22% while the settling time remains around 53 s. This is useful in application where overshoot is the main criteria for consideration like in dryer section of paper mills [Valenzuela et al., 2005] and robotic application. With additional feedback K_5 , good performance on the motor is achieved but the load sides overshoot remains around 71.11%. When using K_6 , both motor side and load exhibit very good response. The detail performance parameters are given in Table 2. With K_7 load side transient is improved, but the motor side performance remains almost same as without feedback. But this can be further improved by re-tuning or changing ξ_r . With feedback K_8 improvement on the load side transient is observed but the motor side remains unbounded similar to control structure using K_2 .

Lastly, the control structure with K_9 was simulated. Both motor side and load side exhibit good transient response. The response time or the rise time of this control structure on both sides is very fast. It has rise time 0.4 s on the load side and 0.12 s on the motor side.

We observe that based on our requirement, we can choose different feedback. But the process operator will have to often do final tuning of the controller iteratively on the actual process to yield more satisfactory control.

Graphical output

The following Figures 5a to j are obtained from the simulation. Blue line plot represent transient response on the load side while the red line plot represent transient response on the motor side. For simulation purpose the required damping co-efficient is taken as 0.7 that is, $\xi_r = 0.7$.

Table 1. Load side transient response parameters.

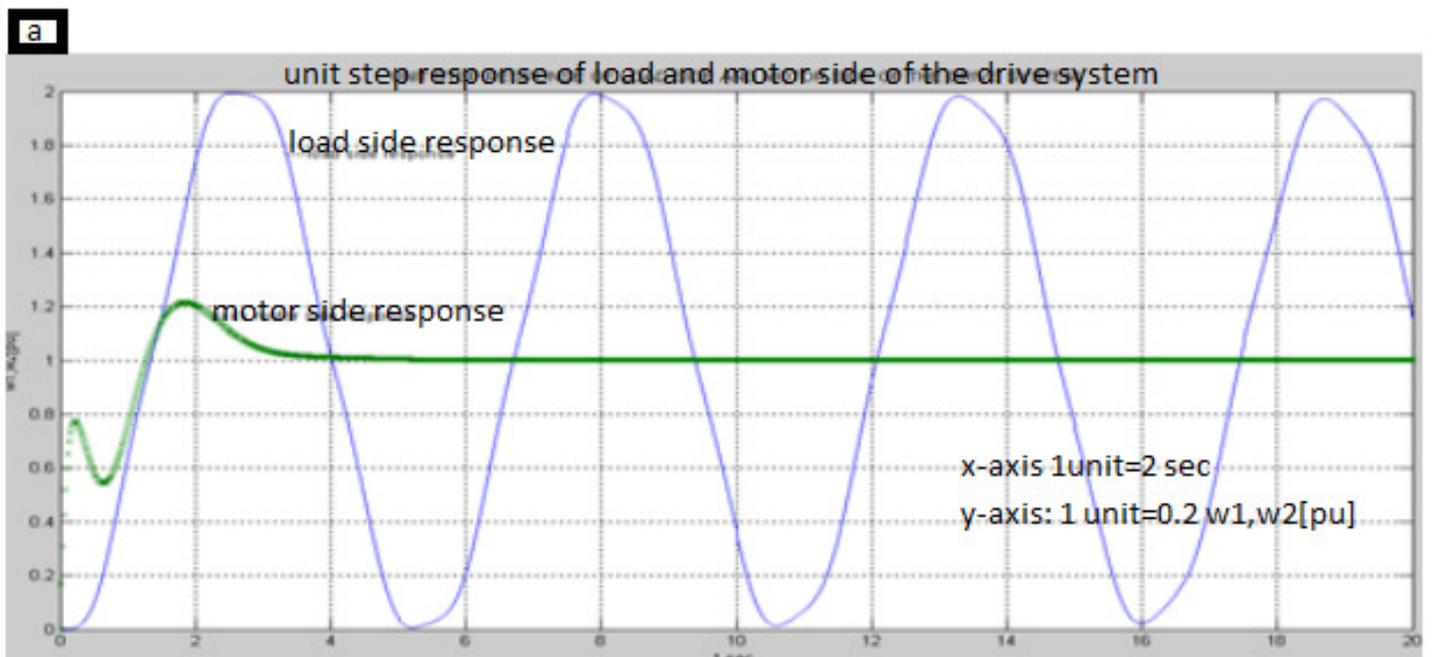
	PI	PI + K ₁	PI + K ₂	PI + K ₃	PI + K ₄	PI + K ₅	PI + K ₆	PI + K ₇	PI + K ₈	PI + K ₉
Tp(s)	2.6	1.41	2.03	1.33	141.08	1.44	1.66	1.99	1.19	1.19
Tr (s)	0.78	0.47	0.67	0.45	30.61	0.41	0.54	0.47	0.40	0.40
Ts (s)	1007.8	3.8	4.71	5.54	53.61	5.55	5.43	247.55	3.18	3.19
% O.S	99.52	54.23	27.99	60.78	0.22	71.11	39.08	109.69	54.32	54.31

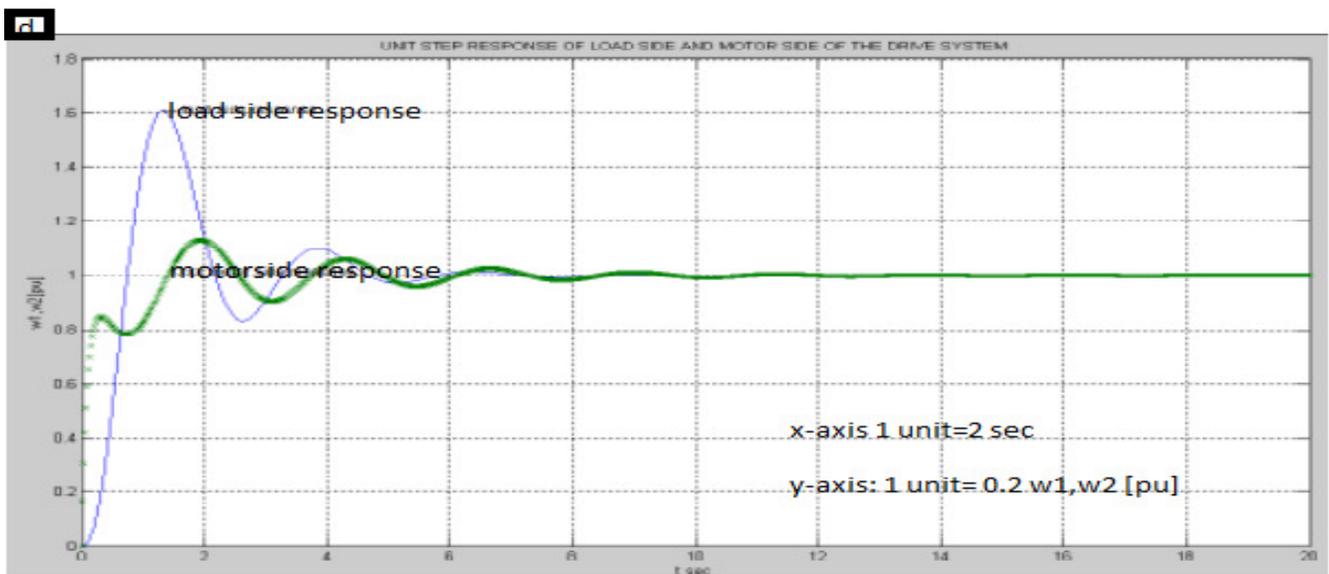
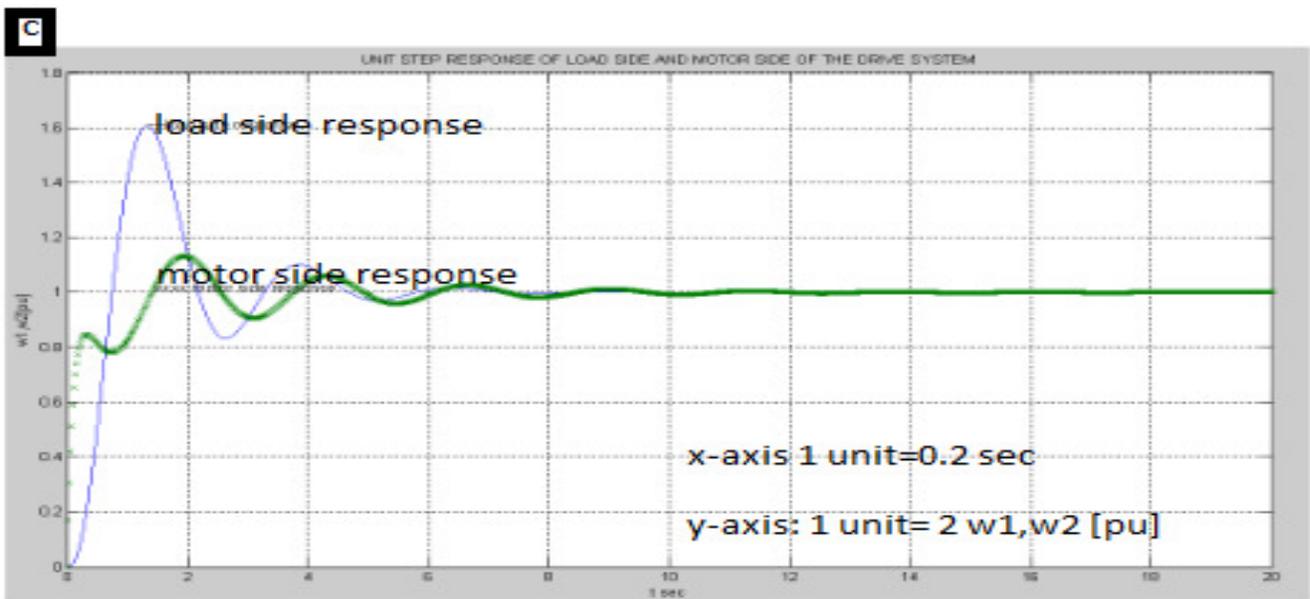
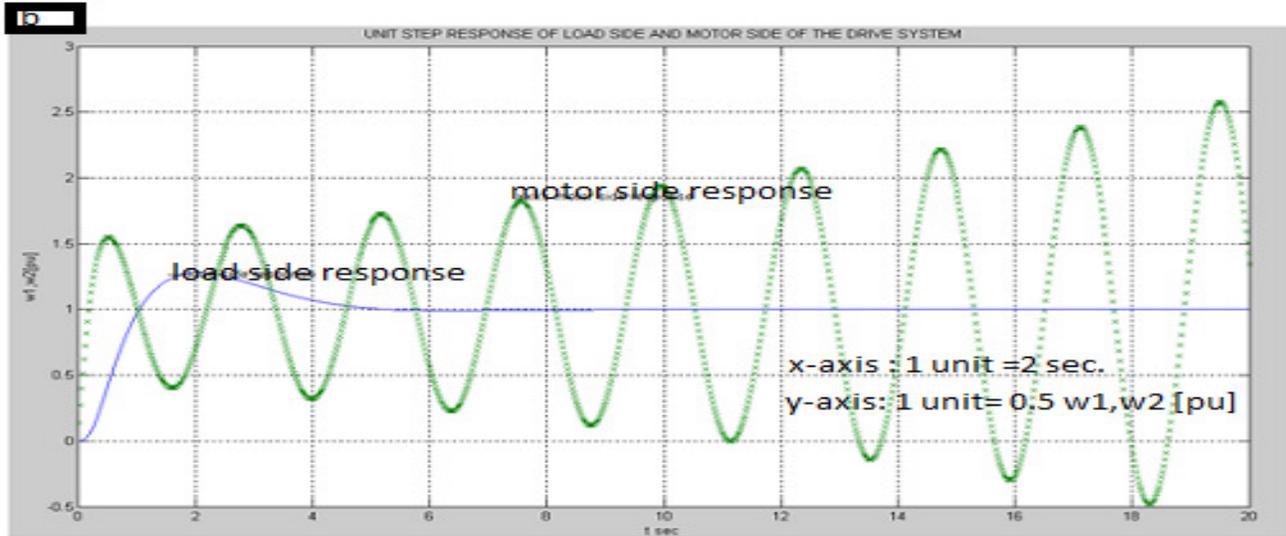
Table 2. Motor side transient response parameters.

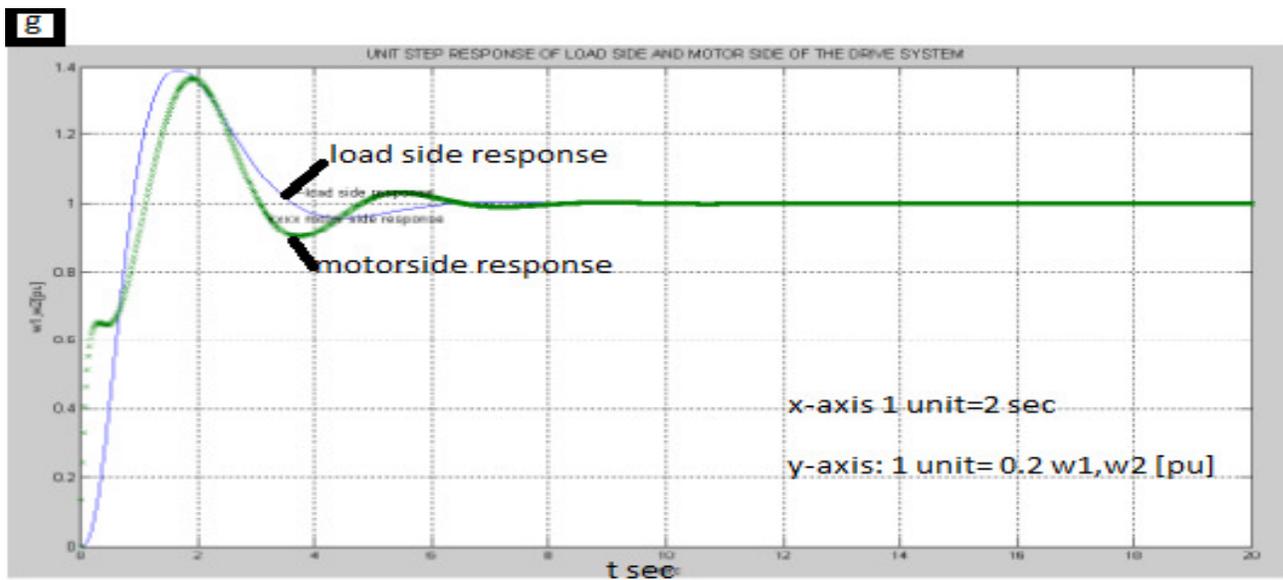
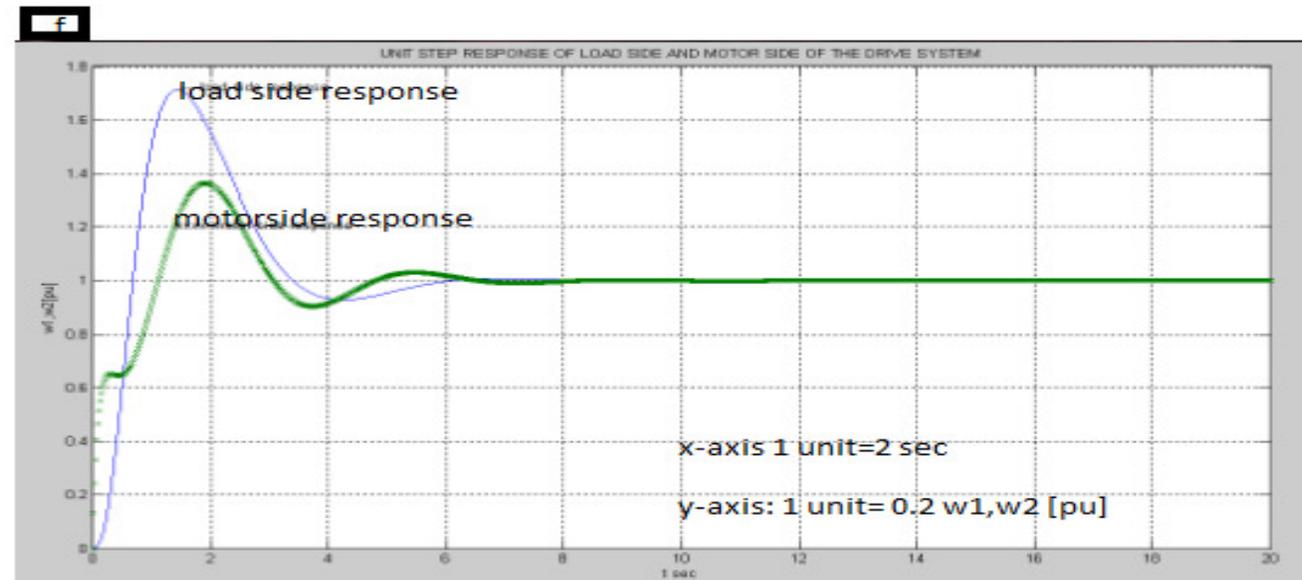
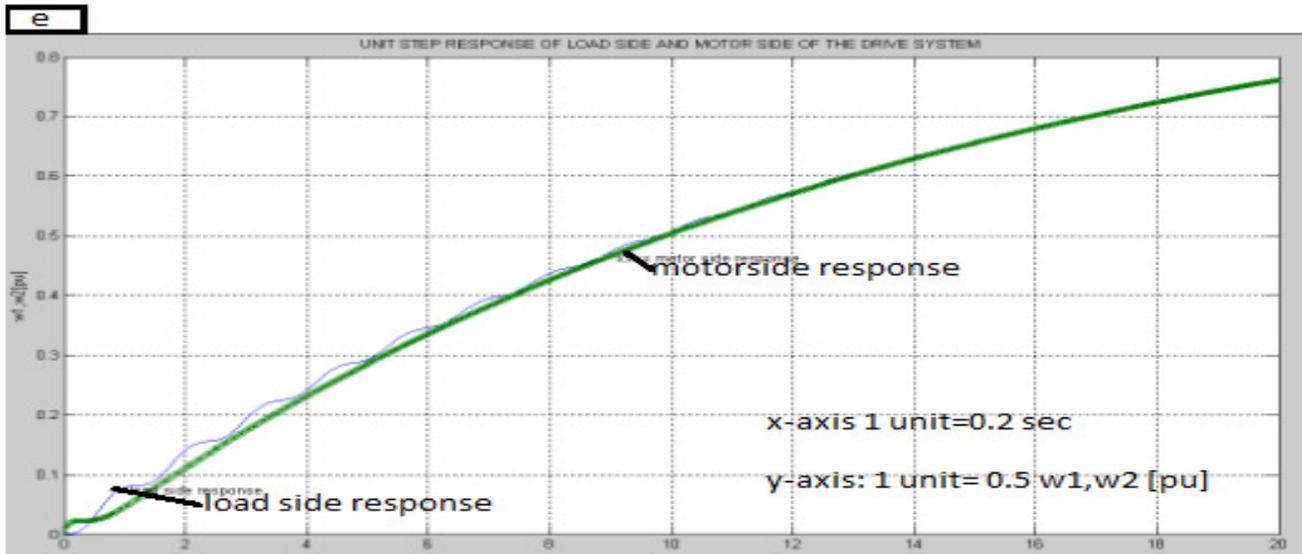
	PI	PI + K ₁	PI + K ₂	PI + K ₃	PI + K ₄	PI + K ₅	PI + K ₆	PI + K ₇	PI + K ₈	PI + K ₉
Tp(s)	1.83	1.83	Unbound response	1.93	137.57	1.90	1.90	5.8	Unbound response	1.83
Tr (s)	1.13	1.05	Unbound response	1.17	29.99	0.98	0.98	0.87	Unbound response	0.12
Ts (s)	3.42	3.97	Unbound response	6.92	52.69	5.96	5.95	219.83	Unbound response	6.70
% O.S	21.54	32.71	Unbound response	13.13	0.22	36.36	36.37	83.17	Unbound response	24.41

Table 3. System parameter.

Motor	Load	Mechanics
Power 5.5 KW	Power 6.4 KW	Inertia of motor 0.037 Kgm ²
Torque 36 Nm	Torque 39 Nm	Inertia of load side 0.125 Kgm ²
Speed 1455 min ⁻¹	Speed 2490 min ⁻¹	Shaft stiffness 2070 Nm/rad







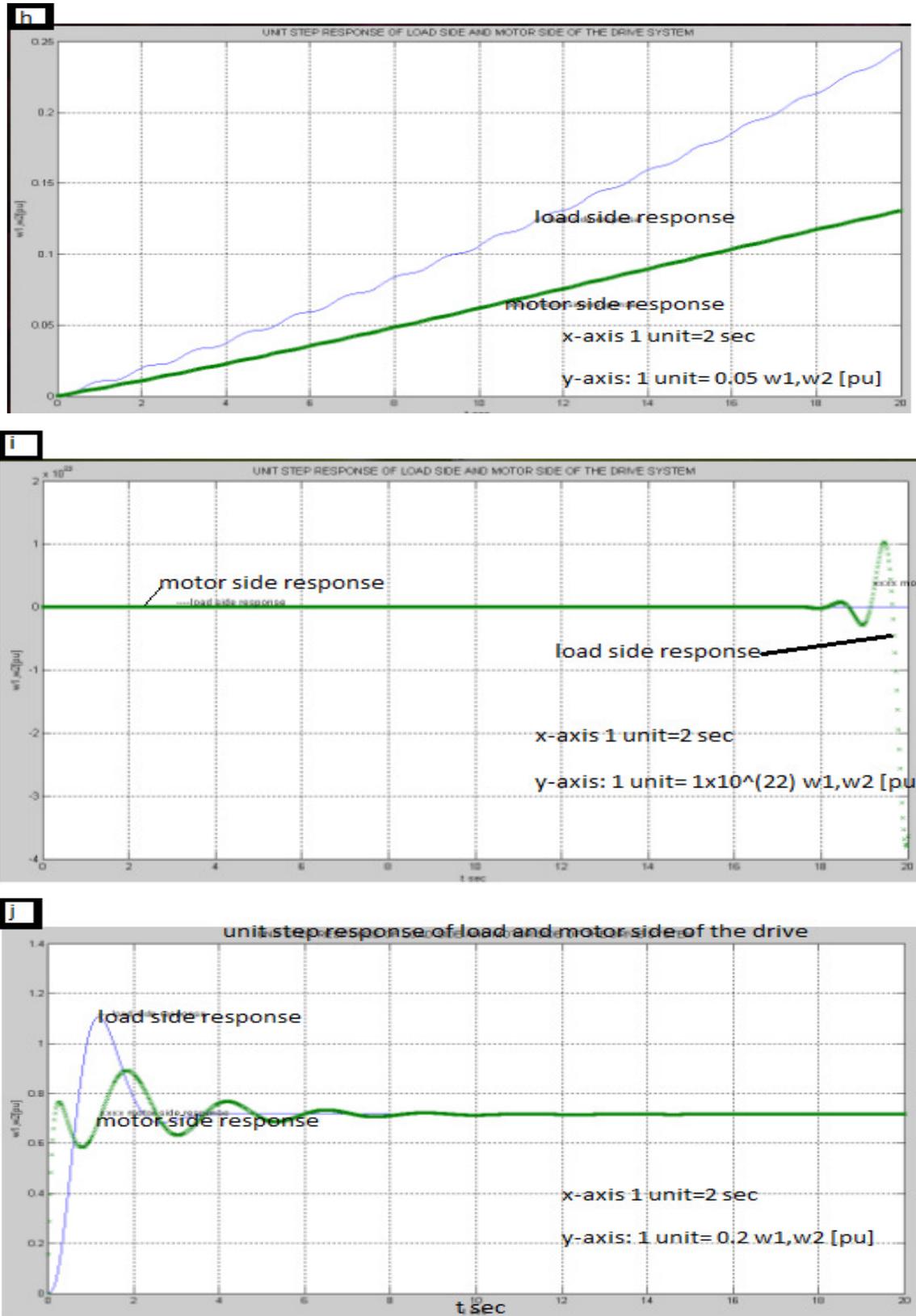


Figure 5. a.transient response when only PI present; b. transient response (PI + KI); c. transient response (PI + K₂); d. transient response (PI + K₃); e. transient response (PI + K₄); f. transient response (PI + K₅); g. transient response (PI + K₆); h. transient response (PI + K₇); i. transient response (PI + K₈), and j. transient response (PI + K₉).

Performance specification (transient response parameter) results

Tables 1 – 3 give details of the transient response parameter obtained from the simulation. The parameters obtained are similar to the response curve obtained. Table 1 gives the load side transient response parameters while Table 2 gives the motor side transient response parameters.

Conclusion

Different cascade control structure using PI supported by various additional feedbacks for industrial drive systems with elasticity were investigated. Classical pole placement method was implemented to calculate the control system controller parameters. The performance of the control structure without additional feedbacks depends only on the mechanical parameters of the drive system and is rather poor. The fact is that the system is of fourth order and only two parameters K_p and K_i available which makes it difficult to achieve desired performance. We observed that in order to damp torsional vibration effectively, application of additional feedbacks is necessary. Resulting from the review of the literature, the application of different feedbacks is possible. The structures with one additional feedback ensure setting the desired value of the damping coefficient, yet the required value of the resonant frequency cannot be adjusted at the same time.

If the design specifications require the free setting of the damping coefficient and resonant frequency simultaneously, then the application of two additional feedbacks is necessary. This work could be carried out in future analysis because with this structure the closed-loop poles can be placed in every desired position. Particular feedbacks can be selected according to our requirement. Conventional control structure using PI alone are rather poor, but with various feedbacks torsional oscillation are being damped. Thus, this methodology acts as sophisticated methods available to

develop a controller that will meet transient response specification and improves performance of various industrial drives. The controller parameters setting according to this methodology will provide acceptable response for many drive systems. But the process operator will have to often do final tuning of the controller iteratively on the actual process to yield more satisfactory control.

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