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Vibration Suppression of a 2-Mass Drive System with Multiple Feedbacks

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Abstract- Torsional Vibration in industrial drive system can lead to a great economic loss due to loss in production when unexpected components failures result in unscheduled system shutdown. The control structures with proportional-integral controller supported by one additional feedback (from different state variables) effectively suppress the oscillation. However, this control structure can ensure only the free setting of damping coefficient but, not the resonance frequency at the same time. So, this control structure will not be proved efficient where the design specifications require the free setting of both. This paper presents the utilisation of two feedbacks control structures for the free setting of both damping coefficient and resonant frequency simultaneously. The structures are being simulated in MTALAB and the results are present here. Also, a comparison is done between the responses of single feedback and two feedbacks. The result shows that two feedback structures are more efficient than single feedbacks.

Index Terms- Two-mass drive system, Elastic shaft, Torsional vibration, free setting, and inertia ratio

I. Introduction

THE INDUSTRIAL DRIVES are the backbone of any manufacturing, material handling and industrial automation. The increasing demand for higher quality and better performance has lead to requirement of relatively high dynamics properties and to minimise transient. A drive system consists of a motor connected to a load machine through a shaft. Normally while designing the industrial drive, the shaft is considered to be stiff. This consideration is acceptable in case of low power servos and standard drives, but not in the cases where characteristics of mechanical parts are to be considered like deep-space antenna or high power servos. Neglecting shaft elasticity in crucial designs can lead to damaging oscillations. This Torsional Vibration can lead to poor product quality and system reliability: the systems can even loss stability or could destroy the mechanical coupling between the driven and loading machine.

In the field of industrial drives, the torsional vibration issue was first encountered in rolling mills application [1][2]. Long shafts and large inertia difference between the motor and load side results to an elastic system. The finite stiffness of the shaft and the large torsional torque (due to the speed difference of motor and load side) on the shaft causes the torsional vibration, which affect the drive system significantly.

In order to suppress the torsional oscillations, different control structures have been developed. In [3] Szabat and Orlowsaka and [4]T. Ananthapadmanabha analyzed to damp the torsional vibration effectively, the application of the feedback from one selected state variable is necessary[9]. The classical cascade control structure [5] with the parameters of proportional-integral aped controller adjusted according to the symmetrical criterion cannot damp the torsional vibration effectively. An additional feedback was required to support PI controller in order to suppress the torsional vibration effectively. In the literature, a large number of possible feedbacks have been reported. However, all these feedbacks can be grouped into three groups [3], according to their dynamic characteristics. The feedbacks were either fed to the electromagnetic torque control loop or the speed control loop. The group 1 consists the feedback from shaft torque '$\mathbf{K_1}$'[6], derivative of motor and load speed difference '$\mathbf{K_2}$' [7] and derivative of load speed '$\mathbf{K_3}$' [8], to the electromagnetic torque control loop. The group 2 consists of shaft torque's derivative '$\mathbf{K_4}$' [3], motor and load speed difference '$\mathbf{K_5}$' [3] and load speed '$\mathbf{K_6}$' [3], to electromagnetic torque control loop. In group 3, derivative of shaft torque '$\mathbf{K_7}$'[6], difference of motor and load speed '$\mathbf{K_8}$'[7] and load speed '$\mathbf{K_9}$' [3], to the speed control loop. The additional feedback allows setting the desired value of the damping coefficient. This control structures can damp the torsional vibrations effectively, but it cannot ensure the free setting of resonant frequency. If the systems design specifications require free setting of damping coefficient and resonant frequency simultaneously, the application of two feedbacks is necessary. In this paper, speed control structure of two –mass system using a conventional PI controller along with two feedbacks is proposed. As the PI controller with one feedback has three parameters and the system is of fourth order,[10] we can add one more feedback, then the closed-loop poles can be placed in every desired position. Thus, we can ensure the free setting of both damping coefficient and resonant frequency. This paper presents only the systematic analyse and analytic guidelines of the selected efficient pairs of two feedbacks control structures, because the number of combinations for are large and cannot be shown here all together.

II. Mathematical Model of the Electrical Drive and the Proposed Control Structure

Basically, the drive system can be modelled as a two mass system, where the first mass represents the moment of inertia of the driven motor and the other mass refers to the moment of inertia of the load side. The mechanical coupling is treated as inertia free. The internal damping of the shaft is sometimes also taken into consideration. The schematic diagram of that model is presented in Fig. 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.

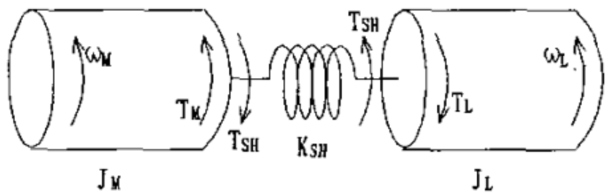

Figure 1. 2-mass system. Where: ωm: motor speed, ωl: load speed; Tm: motor torque; Tl: load torque; Tsh: shaft torque; Jm: motor inertia; Jl: load inertia; Ksh: spring coefficient (stiffness) of drive shaft.

The following Laplace transfer functions are derived from the dynamic mechanical principles:

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

$$\omega_l = \frac{1}{J_l s}(T_{sh} - T_1) \qquad (2)$$

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

The state equation of the system is given by

$$\bar{X} = AX + BU + E^{T_L} \qquad (4)$$

Where,

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

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

$$E = \begin{bmatrix} 0 \\ \frac{1}{J_l} \\ 0 \end{bmatrix}; \quad C = [\,1 \quad 0 \quad 0\,]$$

$$U = T_m;$$

For simplification the model is expressed in per-unit system [3], using the following notation as new state variables:

$$\omega_1 = \frac{\Omega_1}{\Omega_N}; \quad \omega_2 = \frac{\Omega_2}{\Omega_N}; \quad m_L = \frac{M_L}{M_N}$$

$$m_e = \frac{M_e}{M_N}; \quad m_s = \frac{M_s}{M_n}$$

Where $\mathbf{\Omega_N}$ is the nominal speed of the motor; $\mathbf{M_N}$ is the nominal torque of the motor; $\boldsymbol{\omega_1}$ and $\boldsymbol{\omega_2}$ are the motor and load speeds, respectively; $\mathbf{m_e}$, $\mathbf{m_s}$, and $\mathbf{m_L}$ are the electromagnetic, shaft, and load torques in the per-unit system, respectively. The mechanical time constant of the motor $\mathbf{T_1}$ and the load machine $\mathbf{T_2}$ **and** the stiffness time constant $\mathbf{T_c}$ and internal damping of the shaft $\boldsymbol{d}$ are thus, given as

$$T_1 = \frac{\Omega_N J_1}{M_N} \qquad T_2 = \frac{\Omega_N J_2}{M_N} \qquad T_c = \frac{M_N}{K_c \Omega_N} \qquad d = \frac{\Omega_N D}{M_N}$$

Where,
D is the internal damping of the shaft.

The analysed system is described by the following state equations (in p.u. system)

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

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

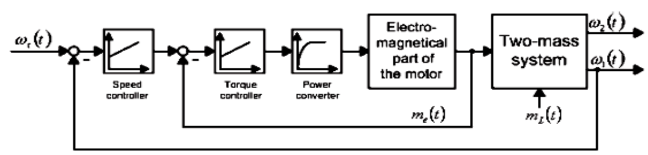

Figure 2. Classic Control Structure

A typical electrical drive system is composed of a power-converter-fed 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 motor torque regulation is performed by the inner loop, consisting 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.

In the literature nine additional feedbacks were proposed to support the PI controller for obtaining the desired output.[3] In Figure 3. But, it ensures only the free setting of damping coefficient.

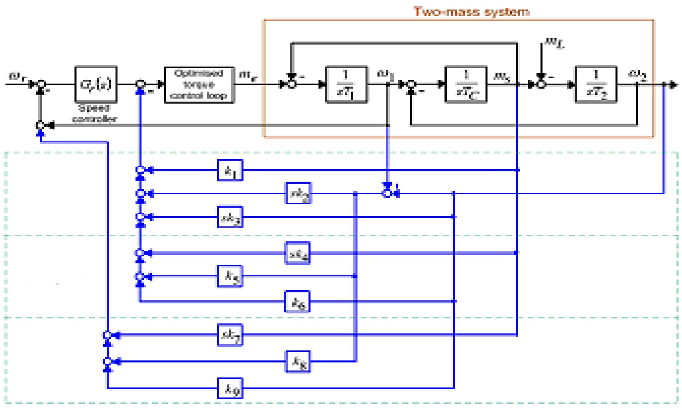

Figure 3. Control structures with addition feedbacks [3]

A. Proposed Control Structures using Two Feedbacks

For the free setting of damping coefficient and resonant frequency, simultaneously, utilisation of two feedbacks is presented in this paper. Total 36 pairs of two feedback combination were formed, where one feedback is fed to the torque control loop while the other is fed to the speed control loop. Here it is not possible to present each of the combination. So, among these 72 combinations only 6 are presented which provides the promising results then the rest. The block diagram of the drive system with simplified inner loop and two additional feedbacks is presented in Figure 4. In a typical industrial drive internal damping coefficient *d* of the shaft has a very small value and, therefore, will be neglected in further analysis.

Figure 4.(a) shows the combination of shaft torque '$\mathbf{K_1}$' and load speed '$\mathbf{K_9}$' , to the electromagnetic torque loop and speed control loop, respectively.

Figure 4.(b) shows the combination of differential of the motor and load speed difference '$\mathbf{K_2}$' and only the difference of motor and load speed '$\mathbf{K_8}$' , to the electromagnetic torque loop and speed control loop, respectively.

Figure 4.(c) presents the combination of '$\mathbf{K_2}$' and '$\mathbf{K_9}$', to the to the electromagnetic torque loop and speed control loop, respectively.

Figure 4.(d) presents the combination of derivative of load speed '$\mathbf{K_3}$' and derivative of shaft torque '$\mathbf{K_7}$' to the to the electromagnetic torque loop and speed control loop, respectively.

Figure 4.(e) presents the combination of '$\mathbf{K_3}$' and '$\mathbf{K_8}$' to the to the electromagnetic torque loop and speed control loop, respectively.

Figure 4.(f) presents the combination of '$\mathbf{K_3}$' and '$\mathbf{K_9}$' to the to the electromagnetic torque loop and speed control loop, respectively.

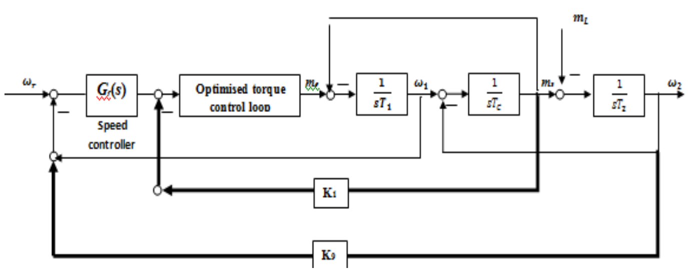

Figure 4.a

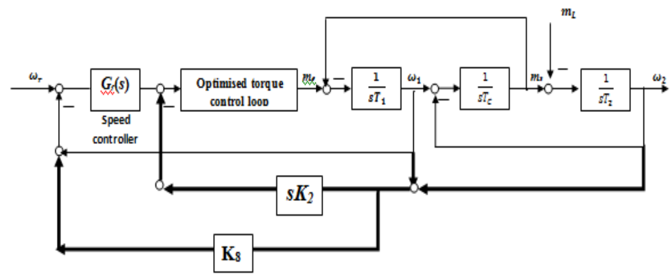

Figure 4.b

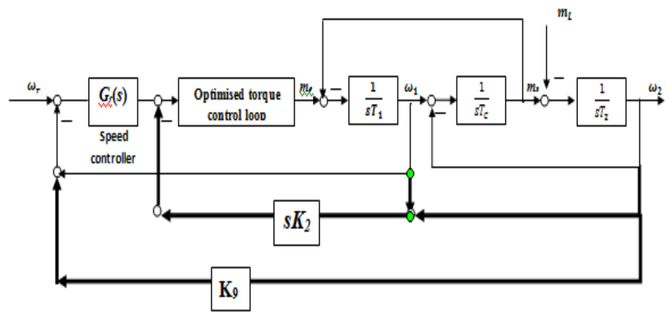

Figure 4.c

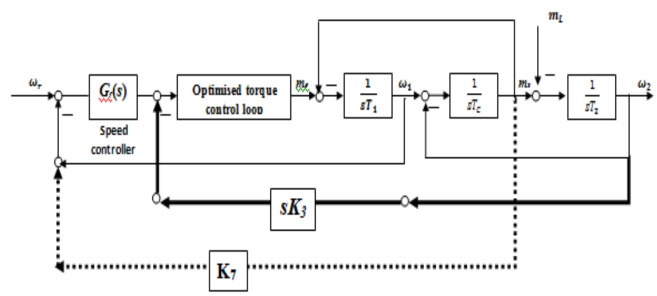

Figure 4.d

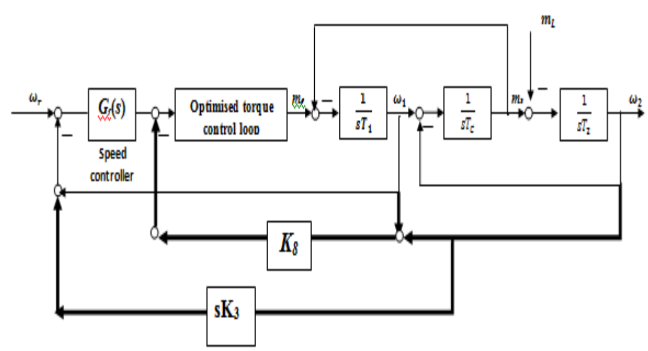

Figure 4.e

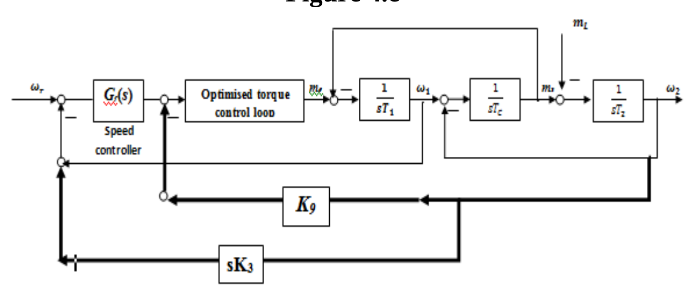

Figure 4.f

The general, close loop transfer functions from the reference speed to the motor and load speeds, respectively for all the control structures demonstrated in **FIGURE 4.(a),(b),(c),(d),(e)&(f)** are given by the equations **(5)** and **(6) []**, motor side $(G_{\omega_1}(s))$ and load side $(G_{\omega_2}(s))$, respectively**,** with the assumption that the optimised transfer functions of the electromagnetic control loop is equal to 1. Using, signal flow graph (SFG) and Mason's Gain formula, the close loop transfer function is obtained.

$$G_{\omega_1}(s) = \frac{\omega_1(s)}{\omega_r(s)} = \frac{G_r(s)(s^2T_2T_c+1)}{s^3T_2T_c(T_1+K_2)+S^2T_2G_r(s)(T_c+K_7+T_cK_8)+s(T_1+T_2(1+K_1)+K_3)+G_r(s)(1+K_9)}$$

$$G_{\omega_2}(s) = \frac{\omega_2(s)}{\omega_r(s)} = \frac{G_r(s)}{s^3T_2T_c(T_1+K_2)+S^2T_2G_r(s)(T_c+K_7+T_cK_8)+s(T_1+T_2(1+K_1)+K_3)+G_r(s)(1+K_9)}$$

Where:

$$G_r(s) = K_P + K_I\frac{1}{s} \qquad \textbf{(7)}$$

is the PI controller transfer function.

B. Structure Analysis of Classic Control

First, analyzing the classic control structure [], without any feedback support. The characteristic equation of the analyzed system is given by

$$s^4 + s^3\left(\frac{K_P}{T_1}\right) + s^2\left(\frac{K_I}{T_1} + \frac{1}{T_1T_c} + \frac{1}{T_2T_c}\right) + s\left(\frac{K_P}{T_1T_2T_c}\right) + \frac{K_I}{T_1T_2T_c} = 0 \qquad \textbf{(8)}$$

The desired polynomial of the system has the following form:

$$(s^2 + 2\xi\omega_0 + {\omega_0}^2)(s^2 + 2\xi\omega_0^2) = 0 \qquad \textbf{(9)}$$

Where ξ**,** is the **damping coefficient** and ω_0**,** is the **resonant frequency** of the close loop system.

$$s^4 + s^3(4\xi\omega_0) + s^2(2{\omega_0}^2 + 4\xi^2{\omega_0}^2) + s(4\xi{\omega_0}^3) + {\omega_0}^4 = 0 \qquad \textbf{(10)}$$

Comparison of equation (10) and equation (8), gives this four set of equations:

$$4\xi\omega_0 = \frac{K_P}{T_1}$$

$$2{\omega_0}^2 + 4\xi^2{\omega_0}^2 = \frac{K_I}{T_1} + \frac{1}{T_1T_c} + \frac{1}{T_2T_c}$$

$$4\xi{\omega_0}^3 = \frac{K_P}{T_1T_2T_c}$$

$${\omega_0}^4 = \frac{K_I}{T_1T_2T_c} \qquad \textbf{(11)}$$

Solving the equation set (11), the parameters of the system, that is, ξ and ω_0**,** as well as the control parameters, $\mathbf{K_p}$ and $\mathbf{K_i}$ are obtained as follows:

$$\xi = \frac{1}{2}\sqrt{\frac{T_1}{T_2}} \qquad \omega_0 = \sqrt{\frac{T_1}{T_2T_c}}$$

$$K_p = 2\sqrt{\frac{T_1}{T_2}} \qquad K_I = \frac{T_1}{T_2T_c} \qquad \textbf{(13)}$$

C. Structure analysis of Proposed Control Structures

These structures use the multiple feedbacks from the shaft torque and the load speed to support the PI controller. Using, two feedbacks allow the free setting of damping coefficient 'ξ' and resonant frequency 'ω' simultaneously. The parameters are calculated using the same method-Pole Placement Method as for classic structure. The parameters of the feedback loop and the speed controllers for combinations are listed in Table 1:

Table 1. Parameters of all Combination

K_1 & K_9	$K_1 = \frac{1}{T_2}\left((4\xi_r^2+1)T_1(1+K_9) - T_1 - T_2\right)$ $K_9 = \omega_r^2T_2T_c - 1$ $K_P = 4\xi_r\omega_rT_1 \qquad K_I = \frac{\omega_r^4T_1T_2T_c}{1+K_9}$
K_2 & K_8	$K_2 = \frac{(T_1+T_2)(1+K_8)}{4\xi^2+1}$ $K_8 = \frac{1}{\omega_r^2T_2T_c} - 1$ $K_P = \frac{4\xi_r\omega_r(T_1+K_2)}{1+K_8} \qquad K_I = \omega^4T_2T_c(T_1+K_2)$
K_2 & K_9	$K_2 = \frac{(T_1+T_2)}{(4\xi^2+1)(1+K_9)} - T_1$ $K_9 = \omega_r^2T_2T_c - 1$ $K_P = 4\xi_r\omega_r(T_1+K_2) \qquad K_I = \frac{\omega^4T_2T_c(T_1+K_2)}{1+K_9}$
K_3 & K_7	$K_3 = \frac{(4\xi^2+1)T_1T_c}{(T_c+K_7)} - T_1 - T_2$ $K_7 = \frac{1}{\omega_r^2T_2} - T_c$ $K_P = \frac{4\xi_r\omega_rT_1T_c}{T_c+K_7} \qquad K_I = \omega_r^4T_1T_2T_c$
K_3 & K_8	$K_3 = \frac{(4\xi^2+1)T_1}{(1+K_8)} - T_1 - T_2$ $K_8 = \frac{1}{\omega_r^2T_2T_c} - 1$ $K_P = \frac{4\xi_r\omega_rT_1}{1+K_8} \qquad K_I = \omega_r^4T_1T_2T_c$

K_3 & K_9	$K_3 = \frac{(4\xi^2+1)T_1}{(1+K_9)} - T_1 - T_2$ $K_9 = \omega_r^2 T_2 T_c - 1$ $K_P = 4\xi_r \omega_r T_1$ $\quad K_I = \frac{\omega_r^4 T_1 T_2 T_c}{1+K_9}$

Where,
ξ_r = desired damping coefficient
ω_r = desired resonant frequency

III. ANALYSIS OF SIMULATION RESULTS

Every control structure is simulated in Matlab 2015a using the calculated parameters in equation 13 for classic control structure and Table 1 for the proposed two feedback structures. For the simulation purpose the inertia ratio '*R*' is set to 1. As the damping coefficient of the system depends on the inertia ratio. In this paper, the transient response of the proposed structures for system parameters T_1 =0.203s, T_c = 0.0026 and T_2= 0.203s are shown.

The classic control structure has a fourth order close-loop system, but due to availability of only two parameters from PI self it is not possible to locate all the poles. The transient speed response of motor and load side is shown in Figure 5. We can observe that there is an unwanted large overshoot at the load side speed transient. In order to improve the system dynamics, we need to place close-loop pole in every desired location which is possible by introducing two feedbacks from the system state space variables.

The utilisation of two feedbacks ensures the free setting of both damping coefficient and resonant frequency simultaneously, which was missing in the use of single feedback. For simulation purpose the desired damping coefficient and the desired resonant frequency is set to $\xi_r = 0.7$ and $\omega_r = 40s^{-1}$.

In **Figure 6.a**, we can notice that there is a significant drop in the load side speed response by using $\mathbf{K_1}$ and $\mathbf{K_9}$ multiple feedbacks. Also, the resting time of the system is also improved. We can see similar results for the other combinations too. The rise times, settling times and the overshoots of all combinations are almost equal for all combinations shown here. **Figure 6.b, c, d, e & f** shows the transient response of motor and load side speed of $\mathbf{K_2}$ and $\mathbf{K_8}$, $\mathbf{K_2}$ and $\mathbf{K_9}$, $\mathbf{K_3}$ and $\mathbf{K_7}$, $\mathbf{K_3}$ and $\mathbf{K_8}$ and $\mathbf{K_3}$ and $\mathbf{K_9}$, respectively. In Table 2, the response specifications of all analysed structures are present. While for comparison between the multiple feedback response and single feedback response the system with single feedback is simulated using the same parameter data and the parameter specifications are shown in Table 3. The torsional vibration is effectively suppressed using the purposed control structures also the dynamics of the system are improved

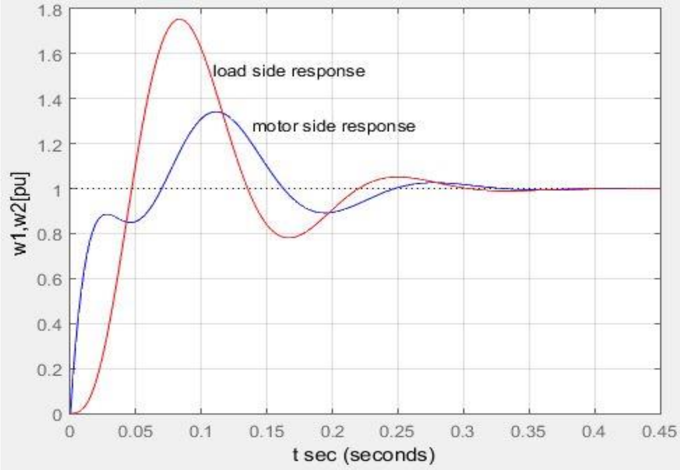

Figure 5 without feedback

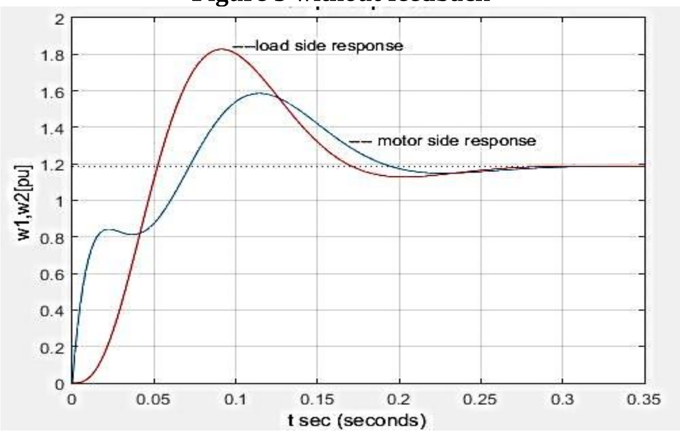

Figure 6.a K_1 and K_9

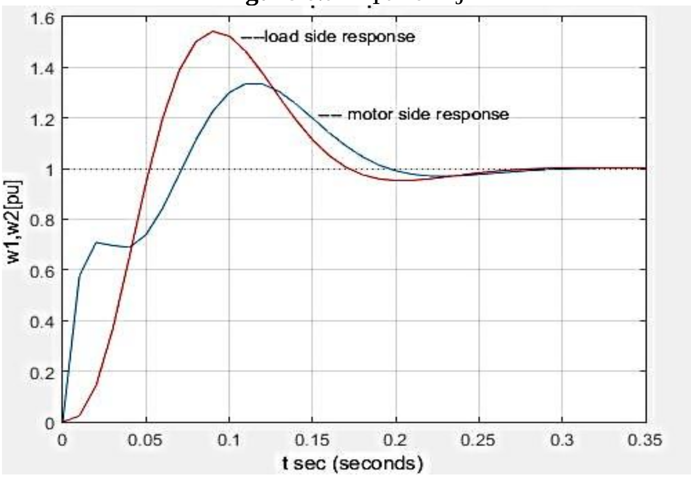

Figure 6.b K_2 and K_8

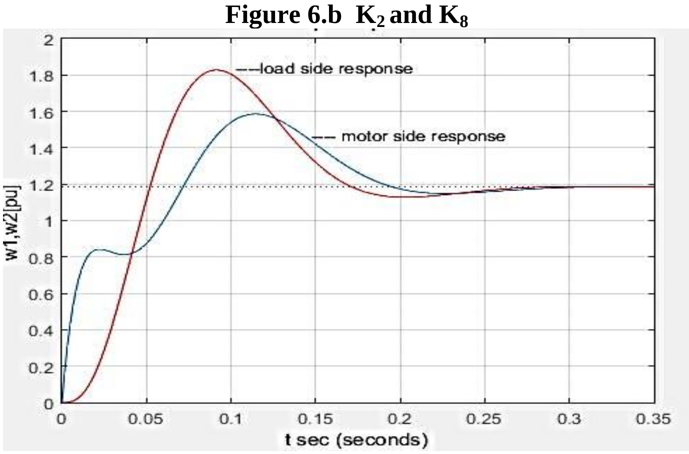

Figure 6.c K_2 and K_9

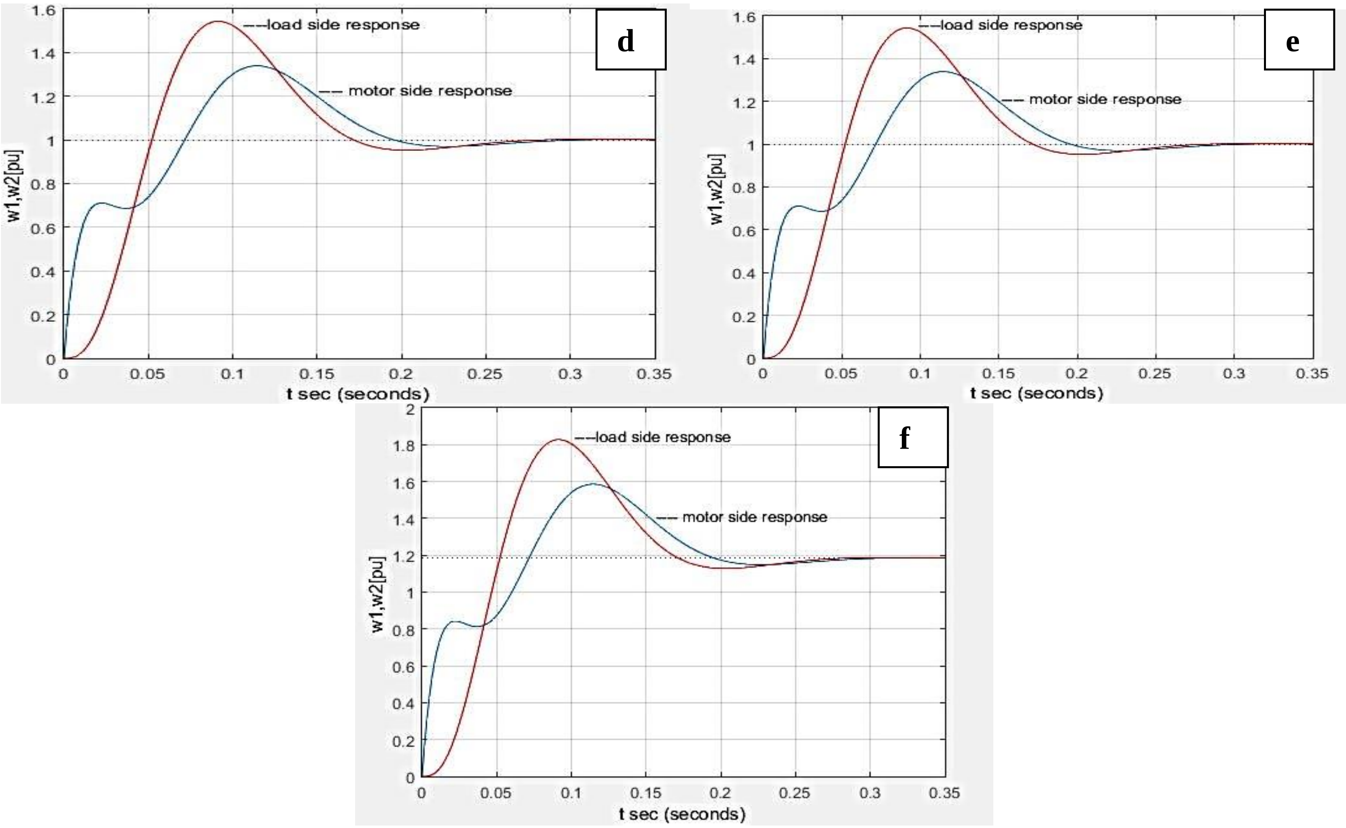

Figure 6. d, e and f presents the K_3 and K_7, K_3 and K_8 and K_3 and K_9, respectively

TABLE 2. <u>Performance Specification of Motor and Load Side Transient Response with Multiple Feedbacks</u>

		T_p (s)	T_R (s)	T_S (s)	% O.S.
Classic PI	motor	0.11	0.583	0.29	34.172
	load	0.831	0.266	0.284	75.443
PI+ K_1+K_9	motor	0.11	0.632	0.255	33.924
	load	0.914	0.310	0.245	54.324
PI+ K_2+K_8	motor	0.11	0.632	0.255	33.324
	load	0.914	0.310	0.245	54.324
PI+ K_2+K_9	motor	0.11	0.632	0.255	33.924
	load	0.91	0.310	0.245	54.324
PI+ K_3+K_7	motor	0.11	0.632	0.255	33.924
	load	0.91	0.310	0.245	54.324
PI+ K_3+K_8	motor	0.11	0.632	0.255	33.924
	load	0.91	0.310	0.245	54.324
PI+ K_3+K_9	motor	0.11	0.632	0.255	33.924
	load	0.91	0.310	0.245	54.324

TABLE 3. Performance Specification of Motor and Load Side Transient Response with Single Feedback

		T_p (s)	T_R (s)	T_S (s)	% O.S.
Classic PI	motor	0.11	0.0582	0.297	34.1
	load	0.831	0.266	0.284	75.443
PI+ K_1	motor	0.10	0.0224	0.297	34.1
	load	0.08	0.264	0.481	105
PI+ K_2	motor	0.10	0.0299	0.0266	32.6
	load	0.08	0.310	0.245	54.324
PI+ K_3	motor	1.8	0.0665	1.96	64.8
	load	1.3	0.666	2.09	87.2
PI+ K_7	motor	Unbound	Unbound	Unbound	Unbound
	load	Unbound	Unbound	Unbound	Unbound
PI+ K_8	motor	0.12	0.76	0.443	34.8
	load	0.10	0.0449	0.443	40.3
PI+ K_9	motor	0.11	0.0165	0.353	26.5
	load	0.91	0.259	0.464	77.8

IV. CONCLUSION

In this paper, new proposed control structures with multiple feedbacks for vibration suppression in industrial drives with long shaft are investigated. The issue of Torsional vibration is effectively suppressed using the multiple feedbacks from system state variables. This paper shows the two feedback method is more promising than using single feedback. As the utilisation of two feedbacks allow the free setting of damping coefficient and resonant frequency simultaneously, this was not possible in the single feedback structure. Pole-Placement method is used to determine the system parameters. Earlier structures with additional feedback surely damped the torsional oscillation but it allows only the free setting of desired damping coefficient for desired output. Systems with the requirement of free setting of both damping coefficient and resonant frequency; there one feedback structure proves not to be efficient. Hence, utilisation of two feedbacks is necessary. As the system is of fourth order and with two feedbacks with PI, we can place the close-loop poles to every desired place for the desired output. All the structures are simulated using the general equations which govern the drive system response.

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