ACTIVE CONTROL FOR MACHINERY EQUIPMENT INDUCED STRUCTURAL VIBRATION USING H∞ CRITERION AND PSO TECHNIQUE

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Abstract:

In this paper, machinery equipment induced structural vibration was investigated and a composite system for structure and equipment was proposed. Tuned mass damper (TMD) and active tuned mass damper (ATMD) were respectively performed for vibration control, in addition, particle swarm optimization (PSO) was utilized for pursuing an optimal active control. Numerical results confirmed that the presented active control strategy could achieve a better vibration suppression compared to TMD control. The PSO based active control also gave inspiration for improving the traditional vibration control.

1 Introduction

In modern industry, building structures are inseparable with machinery equipment, such as rotating equipment, reciprocating equipment, impacting equipment and other typical machines used in manufacturing. Machinery equipment induced vibrations are very harmful for industrial structures and employing isolation and suppression strategies is critical and necessary.

Yang Y B et al. [1] investigated the seismic vibration control for a frame structure and inner precision equipment. Igusa T et al. [2] treated the structure and equipment as a two degree of freedom (2dof) system, and a seismic vibration control was performed. Xu Y L et al. [3] carried out the seismic vibration control for installed precision equipment in an industrial structure. Ismail M et al. [4] proposed a practical device for isolating the vibration of structure and inner equipment under seismic excitation. Simplified 2dof system for vibrations of structure and inner presion equipment has been carried out recently, however, machinery equipment induced structural vibraions are rarely taken into account, which are aptly important vibration hazards in modern industry. In practice, optimization is critical for systematic parameters, configurations and properties etc., which also plays a key role in vibration control. Traditional gradient-based optimization requires computations of sensitivity factors and eigenvectors using an iteration process. This gives rise to heavy computational burden resulting in slow convergence. Moreover, there is no local criterion to decide whether a local solution is also the global solution. Thus, conventional methods that use derivatives and gradients typically cannot locate or identify global optimum. For real-world optimization applications, a good solution is often acceptable, even if it is not the best solution. Consequently, intelligent methods are widely used for global optimization problems. In 1995, Eberhart and Kenney first proposed a swarm intelligence algorithm referred to as particle swarm

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optimization (PSO) [5]. This method is a population based heuristic method, which utilizes swarm intelligence generated by cooperation and competition between particles in a swarm and has emerged as a useful tool for engineering optimization [6].

2 The PSO algorithm

The PSO algorithm is a random optimization method based on swarm intelligence. This algorithm is inspired by social behavior based on bird flocking. It employs a swarm of multiple particles, each with their own position and velocity. All particles share information obtained from other particles, and interaction among the particles makes the search efficient. Each potential solution is also assigned a randomized velocity and potential solutions are called particles. These particles are then "flown" through hyperspace. Each particle keeps track of its own coordinates in hyperspace, which is associated with the best solution (fitness) it has achieved so far. This solution is referred to as '*pbest*'. All values of *pbest* for each of the particle are tracked simultaneously. By keeping track of the overall best value, and its location, the globally optimized solution, *gbest* can be obtained [7-9].
Updating the equations of the velocities and positions of each particle are core parts of the PSO algorithm, and can be described as

$$v_{ij} = \omega v_{ij}(t) + c_1 r_1 (pbest_{ij}(t) - x_{ij}(t)) + c_2 r_2 (gbest_{ij}(t) - x_{ij}(t)) \tag{1}$$

$$x_{ij}(t+1) = x_{ij}(t) + v_{ij}(t+1) \tag{2}$$

where i represents the ith particle, j represents the jth dimension of particle. Inertia weight factor ω plays a key role in the global optimization. A linear changing strategy proposed by Shi and Eberhart [9] is often used for the global optimization. A simple and effective form of the inertia weight factor is used here, namely $\omega = 0.99^t$ [8]. The flowchart of PSO is shown in Fig. 1.

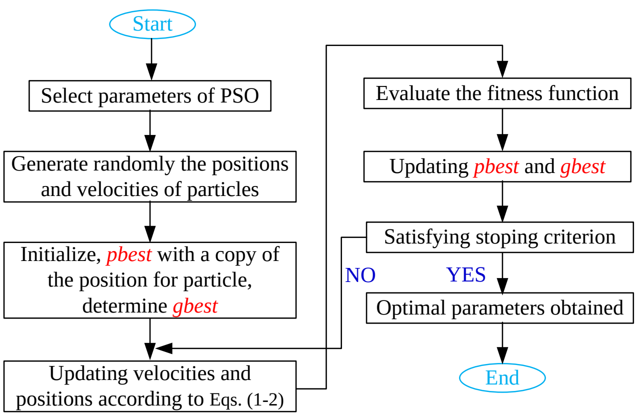


Figure 1. Flowchart of the PSO algorithm.

3 Active control for machinery equipment induced structural vibration

Recently, tuned mass damper (TMD) technique has been broadly applied into the seismic vibration control and wind-induced vibration control [10-12], etc. There is no active control energy in TMD,therefore the suppression effect is often undesirable. To further improve the control performance of TMD, active tuned mass damper (ATMD) was developed [13-15]. The ATMD mainly consists of mass and active controller, in addition, spring and damper can be provided.
Figure 2 Presents a TMD/ATMD control for structure and inner machinery equipment, m_p is the mass of machinery equipment; , c_p are the stiffness and damping of isolation system; $x_p, \dot{x}_p, \ddot{x}_p$ are respectively the displacement, velocity and acceleration of equipment; m_b is the mass of foundation or supported structure; k_b, c_b are stiffness and damping of foundation or supported structure; $x_b, \dot{x}_b, \ddot{x}_b$ are respectively the displacement, velocity and acceleration of foundation or supported structure; m_1, m_2, m_3 and k_1, k_2, k_3 and c_1, c_2, c_3 are respectively the mass, stiffness and damping of structure from floor1 to floor 3; x_1, x_2, x_3 and $\dot{x}_1, \dot{x}_2, \dot{x}_3$ and $\ddot{x}_1, \ddot{x}_2, \ddot{x}_3$ are respectively the displacements, velocities and accelerations of structure from floor 1 to floor 3; m_d is the mass of TMD/ATMD; k_d, c_d are the stiffness and damping of TMD/ATMD; $F_a(t)$ is the active control force supported by actuator in the ATMD; $F_d(t)$ is the generated disturbance by the machinery equipment.

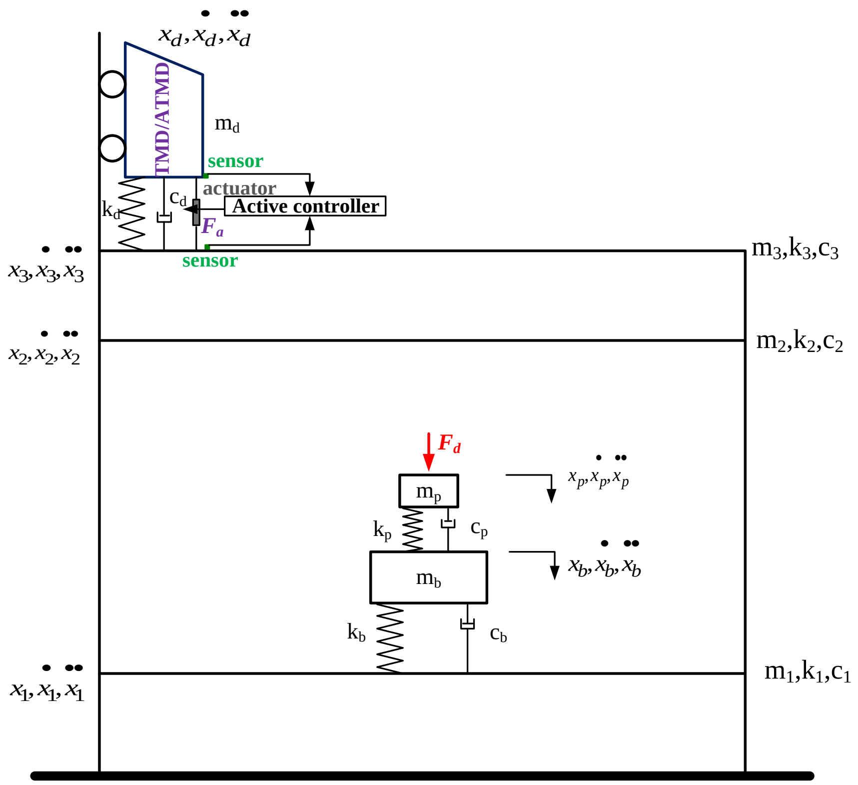


Figure 2. TMD/ATMD control for machinery equipment induced structural vibration.

Figure 2 indicates the TMD/ATMD vibration control for composite system with machinery equipment and structure, and the dynamic equations can be written as

$$\begin{cases} m_1 \ddot{x}_1 + (c_1 + c_2 + c_b)\dot{x}_1 - c_2 \dot{x}_2 - \\ c_b \dot{x}_b + (k_1 + k_2 + k_b)x_1 - k_2 x_2 - k_b x_b = 0 \\ m_2 \ddot{x}_2 - c_2 \dot{x}_1 + (c_2 + c_3)\dot{x}_2 - \\ c_3 \dot{x}_3 - k_2 x_1 + (k_2 + k_3)x_2 - k_3 x_3 = 0 \\ m_3 \ddot{x}_3 - c_3 \dot{x}_2 + c_3 \dot{x}_3 - k_3 x_2 + k_3 x_3 + c_d \dot{x}_3 + k_d x_3 - c_d \dot{x}_d - k_d x_d = -F_a(t) \\ m_b \ddot{x}_b - c_b \dot{x}_1 + (c_b + c_p)\dot{x}_b - c_p \dot{x}_p - k_b x_1 + (k_b + k_p)x_b - k_p x_p = 0 \\ m_p \ddot{x}_p - c_p \dot{x}_b + c_p \dot{x}_p - k_p x_b + k_p x_p = F_d(t) \\ m_d \ddot{x}_d + c_d \dot{x}_d + k_d x_d - c_d \dot{x}_3 - k_d x_3 = F_a(t) \end{cases} \quad (3)$$

With regard to Eq. (3), the control system is referred to as an ATMD strategy when the active control force $F_a(t)k_p$ exists , otherwise it is a TMD. In the ATMD system, a couple of state variables are provided $z_1 = x_1$, $z_2 = x_2$, $z_3 = x_3$, $z_4 = x_b$, $z_5 = x_p$, $z_6 = x_d$, $z_7 = \dot{x}_1$,

$z_8 = \dot{x}_2$, $z_9 = \dot{x}_3$, $z_{10} = \dot{x}_b$, $z_{11} = \dot{x}_p$, $z_{12} = \dot{x}_d$, and a state vector is formulated as:

$$z = \{z_1, z_2, z_3, z_4, z_5, z_6, z_7, z_8, z_9, z_{10}, z_{11}, z_{12}\}^T \quad (4)$$

And the Eq. (4) can be rewritten as the state spaceform,

$$\dot{z} = \mathrm{A}z + \mathrm{B}_1 F_\mathrm{d}(t) + \mathrm{B}_2 F_a(t) \quad (5)$$

$$\mathrm{A} = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ -\frac{k_1+k_2+k_b}{m_1} & \frac{k_2}{m_1} & 0 & \frac{k_b}{m_1} & 0 & 0 & -\frac{c_1+c_2+c_b}{m_1} & \frac{c_2}{m_1} & 0 & \frac{c_b}{m_1} & 0 & 0 \\ \frac{k_2}{m_2} & -\frac{k_2+k_3}{m_2} & \frac{k_3}{m_2} & 0 & 0 & 0 & \frac{c_2}{m_2} & -\frac{c_2+c_3}{m_2} & \frac{c_3}{m_2} & 0 & 0 & 0 \\ 0 & \frac{k_3}{m_3} & -\frac{k_3+k_d}{m_3} & 0 & 0 & \frac{k_d}{m_3} & 0 & \frac{c_3}{m_3} & -\frac{c_3+c_d}{m_3} & 0 & 0 & \frac{c_d}{m_3} \\ \frac{k_b}{m_b} & 0 & 0 & -\frac{k_b+k_p}{m_b} & \frac{k_p}{m_b} & 0 & \frac{c_b}{m_b} & 0 & 0 & -\frac{c_b+c_p}{m_b} & \frac{c_p}{m_b} & 0 \\ 0 & 0 & 0 & \frac{k_p}{m_p} & -\frac{k_p}{m_p} & 0 & 0 & 0 & 0 & \frac{c_p}{m_p} & -\frac{c_p}{m_p} & 0 \\ 0 & 0 & \frac{k_d}{m_d} & 0 & 0 & -\frac{k_d}{m_d} & 0 & 0 & \frac{c_d}{m_d} & 0 & 0 & -\frac{c_d}{m_d} \end{bmatrix};$$

$$\mathrm{B}_1 = [0,0,0,0,0,0,0,0,0,0,1/\mathrm{m}_p,0]^T ; \mathrm{B}_2 = [0,0,0,0,0,0,0,0,-1/\mathrm{m}_3,0,0,1/\mathrm{m}_d]^T .$$

In the TMD/ATMD control, the control output of structure is as

$$Y(t) = [x_1, x_2, x_3]^T \quad (6)$$

and the Eq. (6) can be rewritten as the state space form,

$$Y(t) = \mathrm{C}_1 z + \mathrm{D}_{11} F_\mathrm{d}(t) + \mathrm{D}_{12} F_a(t) \quad (7)$$

where:

$$\mathrm{C}_1 = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix},$$

$\mathrm{D}_{11} = [0,0,0]^T$, $\mathrm{D}_{12} = [0,0,0]^T$.

The observation output of proposed ATMD control is assumed as

$$\mathrm{y}(t) = \begin{bmatrix} x_1 & x_2 & x_3 & x_4 & x_5 & x_6 & \dot{x}_1 & \dot{x}_2 & \dot{x}_3 & \dot{x}_4 & \dot{x}_5 & \dot{x}_6 \end{bmatrix}^T \quad (8)$$

The state space form of Eq. (8) is derived as following,

$$y(t) = \mathrm{C}_2 z + \mathrm{D}_{21} F_d(t) + \mathrm{D}_{22} F_a(t) \quad (9)$$

where:

$$C_2 = \begin{bmatrix} 1&0&0&0&0&0&0&0&0&0&0&0\\ 0&1&0&0&0&0&0&0&0&0&0&0\\ 0&0&1&0&0&0&0&0&0&0&0&0\\ 0&0&0&1&0&0&0&0&0&0&0&0\\ 0&0&0&0&1&0&0&0&0&0&0&0\\ 0&0&0&0&0&1&0&0&0&0&0&0\\ 0&0&0&0&0&0&1&0&0&0&0&0\\ 0&0&0&0&0&0&0&1&0&0&0&0\\ 0&0&0&0&0&0&0&0&1&0&0&0\\ 0&0&0&0&0&0&0&0&0&1&0&0\\ 0&0&0&0&0&0&0&0&0&0&1&0\\ 0&0&0&0&0&0&0&0&0&0&0&1 \end{bmatrix},$$

$$D_{21} = [0,0,0,0,0,0,0,0,0,0,0,0]^T,$$

$$D_{22} = [0,0,0,0,0,0,0,0,0,0,0,0]^T.$$

In this study, active control in the ATMD with H_∞ criterion is performed, by which a group of combined state-space equations can be formulated as,

$$\begin{cases} \dot{z}(t) = Az(t) + B_1 F(t) + B_2 F_a(t) \\ Y(t) = C_1 z(t) + D_{11} F(t) + D_{12} F_a(t) \\ y(t) = C_2 z(t) + D_{21} F(t) + D_{22} F_a(t) \end{cases} \quad (10)$$

In the Eq. (10), active control force $F_a(t)$ in the ATMD can be written as,

$$F_a(t) = Ky(t) \quad (11)$$

Where K is the feedback gain matrix of active controller in the ATMD. By substituting the Eq. (11) into Eq. (10), the Eq. (10) can be rewritten as,

$$\begin{bmatrix} \dot{z}(t) \\ Y(t) \\ y(t) \end{bmatrix} = \begin{bmatrix} A & B_1 & B_2 \\ C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{bmatrix} \begin{bmatrix} z(t) \\ F(t) \\ F_a(t) \end{bmatrix} \quad (12)$$

The active control force $F_a(t)$ can be rewritten as,

$$F_a(t) = K(I_1 - D_{22}K)^{-1}\left[C_2 z(t) + D_{21} F(t)\right] \quad (13)$$

Where I_1 is the identity diagonal matrix.

Followed by, by substituting Eq. (13) into Eq. (10), new state space equations can be formulated as,

$$\begin{bmatrix} \dot{z}(t) \\ Y(t) \end{bmatrix} = \begin{bmatrix} A_{cl} & B_{cl} \\ C_{cl} & D_{cl} \end{bmatrix} \begin{bmatrix} z(t) \\ U(t) \end{bmatrix} \quad (14)$$

where

$$A_{cl} = A + B_2 K(I_1 - D_{22}K)^{-1} C_2,$$

$$B_{cl} = B_1 + B_2 K(I_1 - D_{22}K)^{-1} D_{21},$$

$$C_{cl} = C_1 + D_{12} K(I_1 - D_{22}K)^{-1} C_2,$$

$$D_{cl} = D_{11} + D_{12} K(I_1 - D_{22}K)^{-1} D_{21}.$$

Transfer function by deriving Eq. (14) is obtained as,

$$T_{YF}(s) = C_{cl}\left(sI_2 - A_{cl}\right)^{-1} B_{cl} + D_{cl} \quad (15)$$

In H_∞ control, the following criteria must be satisfied,
(1) the constructed closed-loop control system must be stable,
(2) the infinity norm of transfer function $T_{YF}(s)$ must satisfy $\|T_{YF}(s,K)\|_\infty < \gamma$, where the γ is a small positive number.
In this paper, the PSO technique is utilized to pursue an optimal H_∞ control, and the fitness function can be defined as

$$\|T_{YF}(s,K)\|_\infty \quad (16)$$

where $s = j\omega$ is the complex frequency, and ω is the disturbed circular frequency of machinery equipment.
According to the research presented in Ref. [16], a proposed PSO based H_∞ ATMD control is developed here and depicted in Fig. 3.

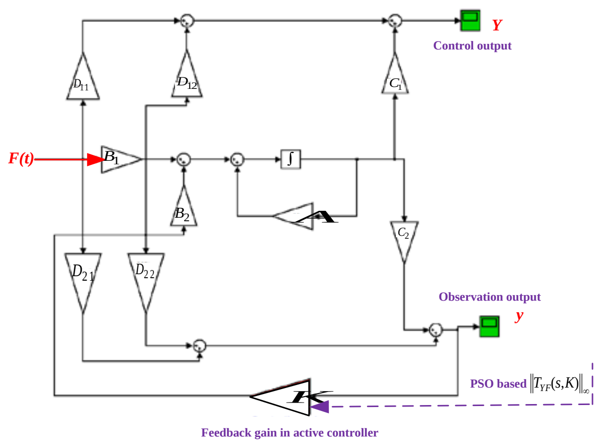


Figure 3. The PSO based H∞ ATMD.

4 Case studies

The parameters configuration of composite system with structure and equipment shown in Fig. 1 is set as

$m_1 = m_2 = m_3 = 4\times10^5\,\mathrm{kg}$,

$k_1 = k_2 = k_3 = 2\times10^8\,\mathrm{N/m}$,

$c_1 = c_2 = c_3 = 1\times10^6\,\mathrm{N\cdot s/m}$, $m_p = 600\mathrm{kg}$,

$m_b = 1200\mathrm{kg}$,

$k_p = 1.5\times10^4\,\mathrm{N/m}$, $k_b = 1.0\times10^6\,\mathrm{N/m}$,

$c_p = 1.0\times10^3\,\mathrm{N\cdot s/m}$, $c_b = 1.6\times10^4\,\mathrm{N\cdot s/m}$.

$F_d(t)$ is sinusoidal disturbance, and the amplitude is $2\times10^3\,\mathrm{N}$, and the frequency is $1.0\mathrm{Hz}$. The natural frequencies of this composite system are 0.7901, 1.5835, 4.4311, 4.6336 and 6.4148 (Hz). Tune the natural frequency of TMD same like the first order of natural frequency of composite system, i.e. 0.7901 Hz. The mass ratio of TMD/ATMD and composite system is assumed as 0.05, and the damping ratio of TMD/ATMD system is assumed as 0.07. In the ATMD control, the optimal configuration of feedback matrix *K* will be carried out by the PSO technique, which is defined as K= [K_1, K_2, K_3, K_4, K_5, K_6, K_7, K_8, K_9, K_{10}, K_{11}, K_{12}], by which an optimal H∞ active controller is obtained. In the PSO, the parameters configuration is set as population size=100, total iteration=500, $c_1 = 2$, $c_2 = 1$, $\omega = 0.99^t$ (t is the iteration number), and the searching range of *K* is defined as

$$\left[-1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3, -1\times10^3\right] \sim \left[1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3, 1\times10^3\right].$$

In the searching range, the -1×10^3 and 1×10^3 indicates respectively the lower and upper limit.

After computation, the fitness convergence is shown in Fig. 4, and the obtained *gbest* solution is

$$K = [\;-0.953 \quad 0.696 \quad -0.895 \quad -0.730 \quad 1.000 \quad 1.000 \quad -0.108 \quad -0.706 \quad -0.518 \quad -0.879 \quad 0.073 \quad 0.661]\times10^3$$

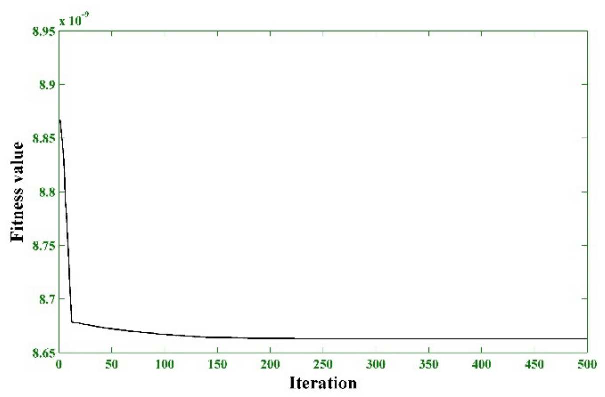
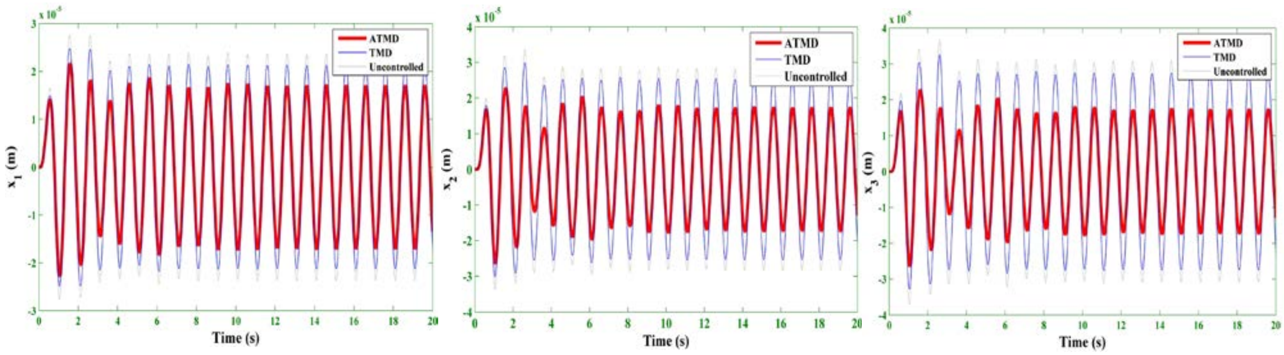


Figure 4. Fitness convergence.

Based on the obtained optimal H∞ active controller, the structural vibration responses with TMD/ATMD control can be computed and shown in Fig. 5, and seen from which, TMD control for machinery equipment induced structural vibration has suppression effect, but not significant. In contrast, the ATMD control is much more effective than TMD The suppression capability of TMD is improved and enhanced, therefore applying ATMD into the machinery induced structural vibrations is entirely feasible. Active control force in ATMD is shown in Fig. 6.



(a) Displacement of floor 1 (b) Displacement of floor 2 (c) Displacement of floor 3

Figure 5. Responses of ATMD and TMD for comparison.

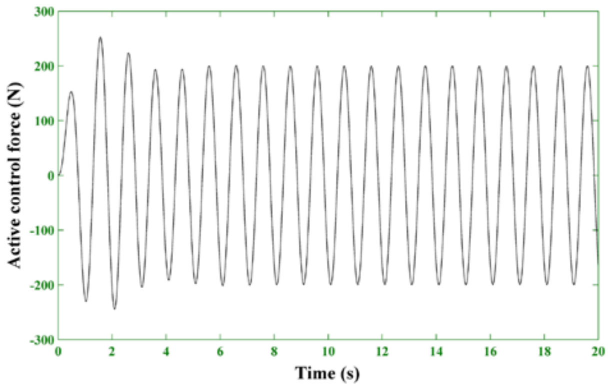


Figure 6. Active control force of the ATMD.

5 Conclusion

In this study, machinery equipment induced structural vibration is investigated, as well as a composite system with 3-floor structure and inner machinery equipment. TMD and H_∞ active control based ATMD are respectively carried out. In the ATMD, the PSO technique is introduced to optimize the active controller. Numerical results show that the proposed active control strategy is attractive for the machinery equipment induced structural vibration, and the developed composite system is novel and effective.

Artificial intelligence applied here can give inspiration for improving the active control. The empirical parameters configuration can be avoided while an optimal design is well assisted.

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