VIBRATION SUPPRESSION OF THE HORIZONTAL FLEXIBLE PLATE USING PID CONTROLLER TUNED BY PARTICLE SWARM OPTIMISATION

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Abstract

This paper presents the development of an active vibration control (AVC) for vibration suppression of the horizontal flexible plate structure using proportional-integral-derivative (PID) controller tuned by a conventional method via Ziegler-Nichols (ZN) and an intelligent method known as particle swarm optimization (PSO) algorithm. Initially, the experimental rig was designed and fabricated with all edges clamped at the horizontal position of the flexible plate. Data acquisition and instrumentation systems were designed and integrated into the experimental rig to collect input-output vibration data of the flexible plate. The vibration data obtained through experimental study was used to model the system using system identification technique based on auto-regressive with exogenous input structure. The plate system was modelled using PSO algorithm and validated using mean squared error, one step ahead prediction and correlation tests. The stability of the model was assessed using pole zero diagram stability. The fitness function of PSO algorithm is defined as the mean squared error between the measured and estimated output of the horizontal flexible plate system. Next, the developed model was used in the development of an AVC for vibration suppression on the horizontal flexible plate system using a PID controller. The PID gains are optimally determined using two different ways, the conventional method tuned by Ziegler-Nichols (ZN) tuning rules and the intelligent method tuned by PSO algorithm. The performances of developed controllers were assessed and validated. PID-PSO controller achieved the highest attenuation value for first mode of vibration by achieving 47.28 dB attenuation as compared to PID-ZN controller which only achieved 34.21 dB attenuation.

Keywords: *Flexible structure, system identification, particle swarm optimization, active vibration control, intelligent controller*

1. Introduction

The usage of flexible structure has been increasing rapidly in many engineering applications. The elements of flexible structure such as plate, beam, frames and shells are widely used in manufacturing applications such as automotive, aircraft, aerospace and submarine. Recently, the dynamic behavior of plates like thin, rectangular, flat and flexible have attracted plenty of considerations by researchers because of its technical importance. The different shapes and boundary conditions at edges of the plate can be frequently found in several engineering applications and used in the industrialized world. These includes the development of solar panel, naval structures, bridge decks and electronic circuit board design [1].

However, the flexible plate is easily influenced by external forces, which lead to the system experiencing high volume of vibration. The dynamic forces and random cyclic loads indirectly influence the stability of the system. A large number of discrete frequencies exist in the system

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can impart high amplitude of vibrations which brings about several negative implications, including noise, fatigue, wear, failure and deterioration in performance of flexible plate system. Thus, mitigating unwanted vibration on the plate structure is compulsory to ensure the effectiveness of the system remains constant [1]. Most researchers have introduced passive vibration control to remove undesired vibration. However, due to several limitations such as lead to an increase in the weight of overall structure and lack of versatility, it cannot withstand the low frequency vibrations of the flexible plate system. Additionally, this method is only applicable for high frequency vibration systems but can be very costly and bulky for low frequency systems. Hence, active vibration control (AVC) is proposed in this research study to overcome low frequency vibrations faced by flexible plate systems [2].

Lueg is a pioneer who introduced the fundamentals of AVC technique for noise cancellation in the early 1930s [3]. He proposed the superposition theory where unwanted sound signal may be cancelled by introducing a secondary sound signal 180 degree out of the phase. Since then, plenty of interest has been shown by other researchers to utilize this technique for various applications [4-8]. AVC can be defined as cancelling the vibration at a desired location using superposition of waves by generating a secondary vibration to destruct interferes of unwanted vibration source [9].

In the past decades, many researchers implemented AVC in various applications for industrial processes. Tokhi and Hossain (1996) presented a framework of adaptive control to suppress vibration of a flexible beam using active control mechanism. The control mechanism was designed for on-line implementation so that the developed controller can work as a self-tuning controller [10]. Caruso et al. (2003) proposed to apply AVC system for an elastic plate using basic control law. The AVC mechanism was then implemented into finite element method (FEM) simulation algorithm [11]. Mat Darus (2004) proposed an intelligent control strategy to suppress the vibration of a square thin plate structure. The flexible plate was modeled using finite difference method (FDM) formulation and a feedforward self-tuning controller was tested to the system in order to reduce unwanted vibration [12].

Gardonio and his teams (2004a; 2004b; 2004c) developed a velocity feedback controller for a smart plate in order to reduce sound radiation or transmission. The prototype smart panel was developed with 16 decentralized control units where each control unit possessed a collocated accelerometer sensor and a piezoceramic patch actuator transducer [13-15]. Tavakolpour (2010) presented an intelligent active vibration control system for flexible structures. The flexible structure was initially modeled using finite differences method and validated through experimental study. Three intelligent control schemes were developed using AVC mechanisms for vibration suppression purpose. However, the best controller can be developed, if only, an accurate model of the flexible plate system is found. Finding the appropriate model of the dynamic system will lead to effective vibration control [16].

Recently, system identification techniques has been widely recognized by researchers as means to find an accurate model for dynamic systems based on observing input and outputs for various control application [12]. In this research, a model of a horizontal flexible plate system was developed using system identification via PSO algorithm based on input-output vibration data obtained through experimental study. Details of the experimental setup for collecting vibration data purpose was explained in detail in the following section.

PID controller has been widely used to control systems in many applications. In fact, more than 90% of industrial controller uses a PID controller for their applications because of its simplicity and robustness [17, 18]. In spite of that, finding an optimal gain for PID controllers is crucial and challenging to achieve the best performance in control system. Many methods have been proposed previously, which includes conventional methods such as ZN. The simplicity of this method makes it a favorite among researchers. Even so, the ZN formula still faces persistent issues such as finding the parameter gains of PID controller when applied to industrial plants. Frequently, this ZN method fails to yield an acceptance performance since the value of PID parameters are often subsequently refined. This leads to a large overshoot and settling time [19].

In recent times, evolutionary algorithms such as genetic algorithm (GA), artificial bee colony (ABC), firefly algorithm and PSO has been commonly employed by researchers to tune the parameter gains value of the PID controller in various plants. However, PSO algorithm has become a favorite choice among industries for application because evolutionary algorithms consist of less parameters required to be tuning coupled with very low computational cost while exhibiting high performance [20]. Various attempts have been made to find the optimal gains for PID controller by using PSO algorithm. Zamani *et al*. (2009) developed a fractional order PID (FOPID) controller for automatic voltage regulator (AVR) tuned by PSO algorithm [21]. PSO algorithm was used to carry out the aforementioned design procedure. Alfi and Modares (2010) developed a novel adaptive particle swarm optimization (APSO) to find the optimal system parameters. The aims of the proposed algorithms is to achieve faster convergence speed and better solution accuracy [22].

Hongqing *et al.* (2011) has proposed an improved PSO algorithm to optimally tune PID gains for water turbine governor. The performance of IPSO was compared to conventional PSO for five well-known benchmark functions [23]. Moharama *et al.* (2015) presented a new algorithm to achieve optimal parameters of PID controller. The algorithm was developed based on hybridizing both differential evolution (DE) and PSO with an aging leader and challengers (ACL-PSO) algorithm [24]. Sayed *et al*. (2015) proposed a new hybrid jump PSO (HJPSO) to tune PI controller for boiler turbine unit. HJPSO was developed based on Gaussian and Cauchy mutation in order to improve standard PSO algorithm. The strategy of HJPSO is observing the local and global best particles and bring that particle into its new best position [25]. Xufei *et al*. (2015) proposed an improved PSO algorithm using adaptive inertia weight based on chaotic map and applied to tune the PID controller. The modified algorithm was validated using three benchmark functions and it proved its effectiveness [26].

Inspired by the thoughts of previous research, PSO algorithm was proposed in this paper to find the optimal parameter gains for the PID controller. The performance of the proposed tuning method is compared with the ZN method. The input-output vibration data was collected experimentally. The experimental setup for collecting the vibration data was discussed in the section 2.0 whereas the development of system identification using PSO was described in section 3.0. The development of the proposed controller and the performance of the developed controller in this research was discussed in section 4.0 and 5.0, respectively.

2. Experimental setup

Initially, a horizontal flexible plate was designed and fabricated for this work. The flexible structures used for the experiment are square, thin and flat aluminum plate with dimensions of $0.7m \times 0.7m \times 0.001m$. The specification of the flexible plate used in this research was listed in Table 1. The purpose of this experimental setup is to collect the input-output vibration data of the flexible plate experimentally. The flexible plate was placed horizontally to allow vertical vibration and all the edges were clamped.

An actuation force was applied on the experimental rig at the excitation point of flexible plate using a magnetic shaker and a circular shaped permanent magnet. The permanent magnet was attached at the excitation point while the magnetic shaker was located at a distance of $1\,\mathrm{cm}$ parallel to that magnet. Then, the magnetic shaker was connected to the power amplifier (type 2706) and function generator (GFG-8250A) as shown in Figure 1. The function generator was employed to create a sinusoidal actuation force to excite the plate.

Table 1: The specification of flexible plate used

Parameter	**Value**
Length, L	$0.7m$
Width, w	$0.7m$
Number of sections	20×20
Thickness, T	$0.001m$
Moment of inertia , I	$5.1924\times10^{-11}kgm^2$
Mass density per area, ρ	$2.71\times10^3 kg/m^3$
Young's modulus, E	$7.11\times10^{10}N/m^2$
Poison ratio, v	0.3

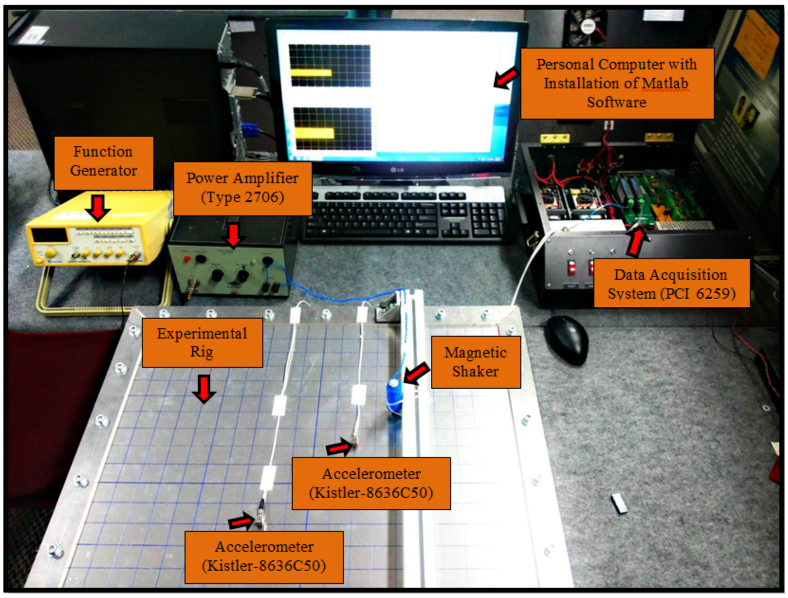


Figure 1: The experimental setup for collecting input-output vibration data.

Two pieces of piezo-beam type accelerometer (Kistler-863650C) were used to sense the acceleration signal which represents the structural vibration. The accelerometers were attached at two different positions, at the observation and detection points as shown in Figure 2. The piezo-beam type accelerometers were directly connected to the data acquisition system (PCI 6259) which was mounted inside a computer on a PCI-bus connected with SCC-68 through shielded cable. A personal computer equipped with high end processor, 4.00 GB RAM and MATLAB software was used to analyze the required signals obtained in the experiment.

3. SYSTEM IDENTIFICATION

In this paper, the system identification technique was used to model the dynamic system of the horizontal flexible plate structure. The evolutionary algorithm known asPSO was selected as the modeling the system. Details regarding the PSO was explained in detail in section 4.2. The system identification was devised using ARX model structure in MATLAB software. There were 5000 input-output vibration datasets acquired through the experimental study. The datasets were divided into two parts, where the first 2500 dataset was used to train the model, while the another 2500 dataset was used to test the performance of the developed model. The performance of the developed system was validated using mean squared error (MSE), one-step ahead predication (OSA), correlation tests and pole zero diagram stability. The structure realization was performed using heuristic method since there was no prior knowledge regarding on how to select the best model for a horizontal flexible structure system.

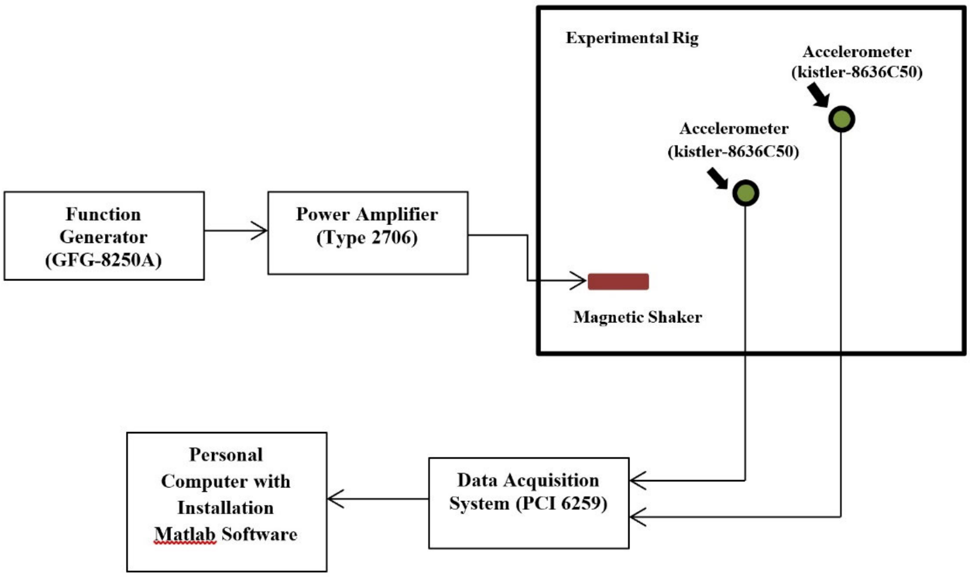


Figure 2: The layout of experimental setup

For identification using PSO, the best model was obtained by tuning different model orders, number of particle and number of iteration. The best model order was the 2nd order, which was obtained with 500 number of particles in the 400th iteration. The best and lowest MSE for PSO identification were $2.3307 \times 10^{-5}$and $4.3947 \times 10^{-6}$for training and testing data, respectively. The MSE versus number of generations has been plotted in Figure 3. Both actual and predicted output of the flexible plate system in time and frequency domains were plotted in Figures 4 and 5, respectively. Based on Figure 4 and 5, it was proven that the model developed is able to imitate the measured output very well. Figure 6 shows the error between actual and estimated output achieved using PSO modeling.

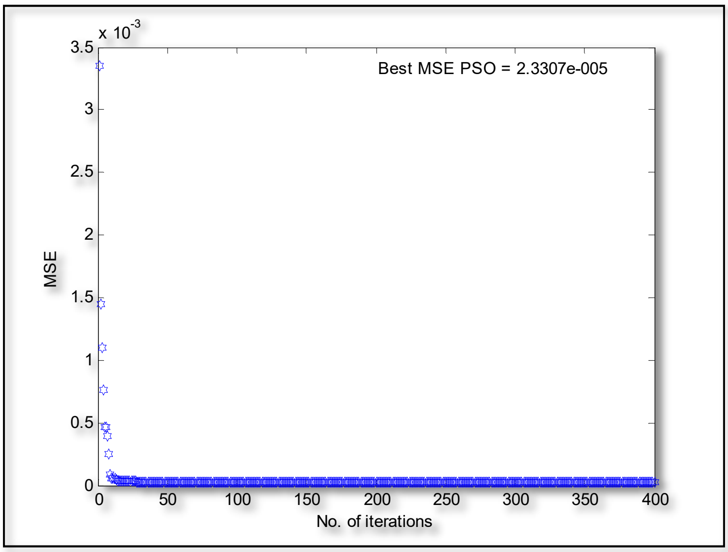
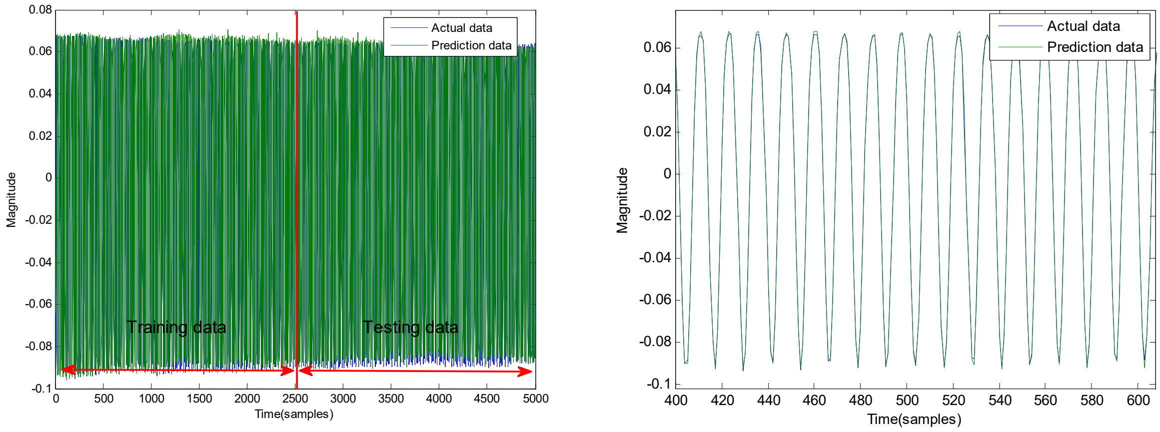


Figure 3: Mean square error versus number of generation using particle swarm optimization modelling



(a) Full data (b) 400-600 samples

Figure 4: System output in time domain using PSO modeling

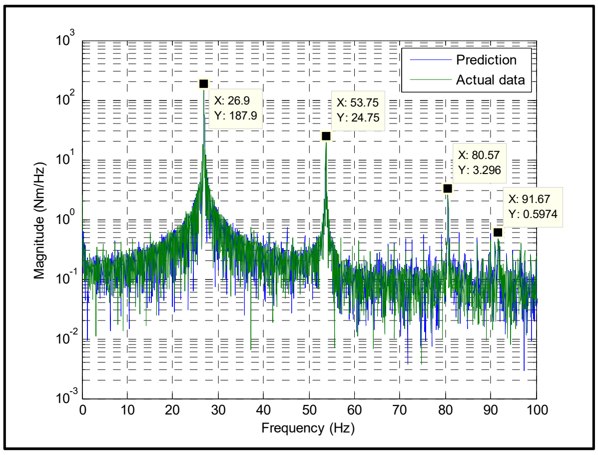


Figure 5: Output and estimated output of the system in Frequency domain using PSO modeling

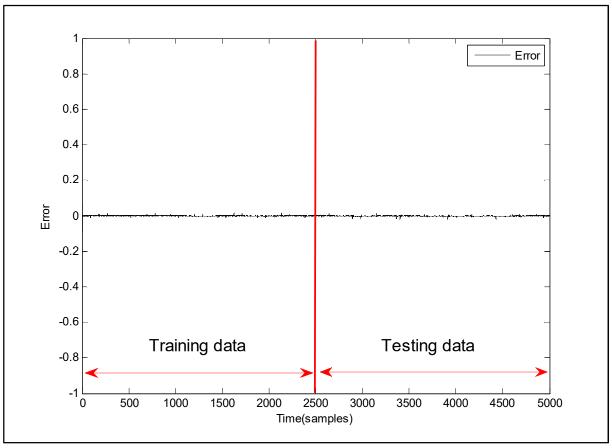


Figure 6: Error between actual and estimated output of the system using PSO modeling

The pole-zero diagram and correlation tests results are depicted in Figure 7 and 8, respectively. According to Figure 7, it was noticed that the model was stable where all the poles of the transfer function in an unit circle. The correlation tests were carried out to determine the effectiveness of the developed system. PSO modeling indicated that the model is unbiased as the results were found to be within a confidence level of 95%. The transfer function obtained using PSO modeling were described as Equation 1. This transfer function will be used later on to develop a controller for vibration supresssion of horizontal flexible plate.

$$\frac{0.3483z^{-1} - 0.002182z^{-2}}{1 - 1.414z^{-1} + 0.9931z^{-2}} \tag{1}$$

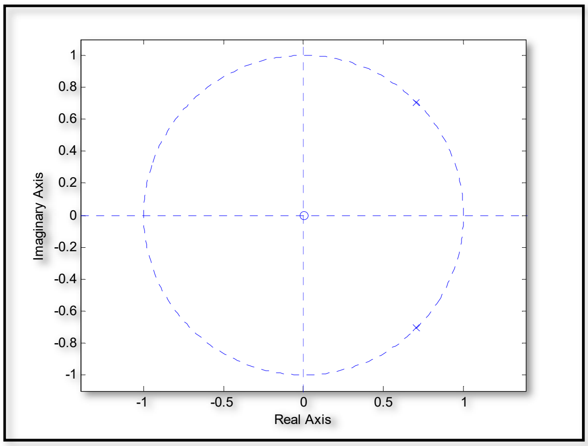


Figure 7: Pole-zero diagram of the system using PSO modeling

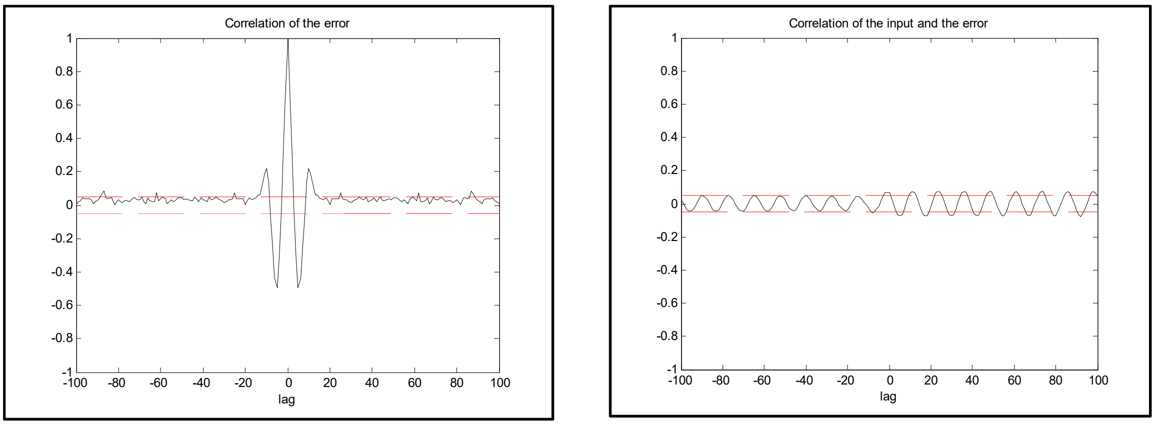


Figure 8: Correlation tests

4. ACTIVE VIBRATION CONTROL

In this section, intelligent control schemes of the horizontal flexible plate structure are introduced. AVC technique was applied in this research to suppress the vibration on the plate system. The concept of AVC is reducing the amplitude of vibration by using the superposition of waves where the secondary vibrational signal generated destructively interfere with the unwanted vibration at the desired location, mitigating vibration [4]. The AVC responds when the sensors applied at the system detects unwanted disturbances through the controller to analyze the vibration. The actuators were supplied with information to produce the secondary signal to superimpose the disturbance and interference with the primary signal which led to vibration cancellation [27].

Due to its robustness and reliability in controlling process, PID controller was employed in this research paper. This type of controller is widely used in the industry because it is cheap, easy to understand, maintain and implement in control structures of various application. However, it is difficult to find an optimal gain to be applied for an acceptable performance.

Many strategies have been proposed by other researchers to obtain the best tuning method because improper tuning method may lead the poor robustness, cyclic and slow system recovery [28]. Therefore, this paper presents the development of a PID controller tuned by an intelligent algorithm via PSO for vibration suppression of a horizontal flexible plate system. The performance of the proposed controller will be compared with the conventional ZN method. The purpose of this study to achieve the best tuning method and improve control strategy for further enhancement of control performance. Then, the proposed controller was validated by employing different disturbance into the system. A Matlab/SIMULINK was utilized to assess and verify the proposed controller schemes in this paper. The corresponding controller parameters were fed to the closed loop PID controller in Matlab/SIMULINK. The error for each sample was calculated and the MSE was evaluated. MSE was set as the fitness value in this algorithm. This is to adjust the PID parameters in order to achieve a lower fitness value. Figure 9 demonstrates the block diagram of PID based controller tuned by PSO and ZN methods.

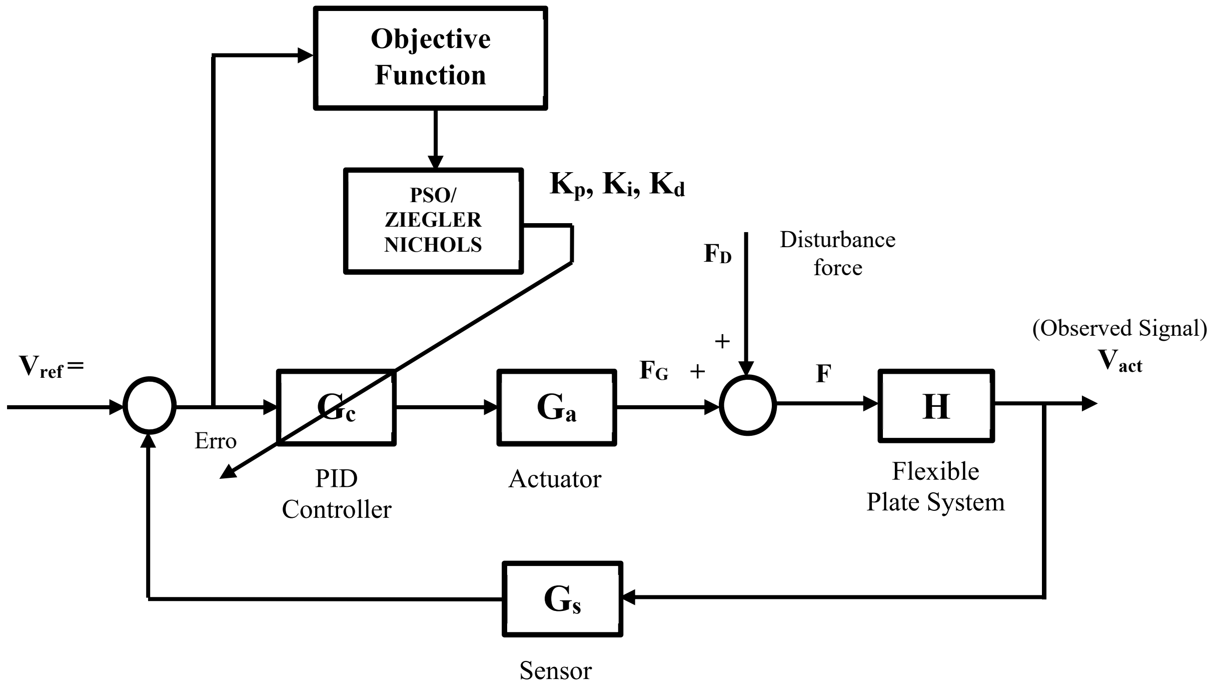


Figure 9: The block diagram of PID based controller tuned by Ziegler-Nichols and particle swarm optimization methods.

4.1 PID controller tuned by ZN

In 1942, John G. Ziegler and Nathaniel B. Nichols developed the well-known loop tuning technique to tune PID parameters, which is also known as the ZN tuning rules. This method has been successfully implemented into feedback control strategies and widely employed in many industrial applications. There were two types of tuning method developed by Ziegler and Nichols, which are called as first method and second method. The first method was developed for open loop system while the second method was developed for closed loop system. In this paper, the second method was applied to tune the PID controller parameter. Initially, the gain value of integral, *I* and derivative, *D* are set as zero. Then, the gain value of proportional, *P* is increased from zero until an optimum value of K_P, defined as K_U is reached. The condition of oscillation response was in sustaining mode and its oscillation frequency period was recorded and defined as T_U. The value of K_P, K_I and K_D were calculated based on ZN formulation as listed in Table 2 [29].

Table 2: Ziegler-Nichols formulation for tuning the PID parameter

Controller	K_P	K_I	K_D
PID Controller	0.6 K_U	$2K_P/T_U$	$K_PT_U/8$

4.2 PID controller tuned by PSO

PSO is an intelligent algorithm which was proposed by Kennedy and Elberhart in 1995. This algorithm was inspired from the social behavior of organisms such as bird flocks and schooling fishes. This optimization become popular among researcher due to a number of advantages exhibited by this algorithm as compared to other optimization methods. This algorithm does not require derivation calculation to develop it. On the other hand, the algorithm can be easily programmed and only requires the tuning of a few parameters [30].

In PSO algorithm, firstly, the number of particles is randomly initialized. The particle flies into the space problem where every particle will remember the best position it has seen. The position of each particles is represented as *i*-th while the particle in *d*-dimensional vector in problem space as $x_i = (x_{i1}, x_{i2}, ..., x_{id})$ where $i = 1,2,...,n$ (n is number of particles) while the velocity vector of *i*th particle $v_i = (v_{i1}, v_{i2}, ..., v_{id})$ is known as the change of its position. Its performance is evaluated based on the predefined objective function, where in this research, the objective function is to achieve the lowest MSE. The flying direction of each particle is the dynamical interaction of individual and social flying experience. The algorithm completes the optimization by determining the personal best solution for each particle, also known as *pbest* while the global best value for the whole swarm is known as *gbest*.

The particles will then use the memories of their best position it has seen to modify their own velocities and positions. Form this part, the particles will always fly through the problem space to search and achieve its own previous best position and the previous best position attained by any particle is the swarm. The best fitness value represents as $p_i(p_{i1}, p_{i2}, ..., p_{id})$ while the fittest particle found at time t represents as $p_g(p_{g1}, p_{g2}, ..., p_{gd})$. Then, the velocities and positions of particles are updated using the following formulas [30-32]:

$$v_{id}(t+1) = wv_{id}(t) + c_1 rand1(p_{id} - x_{id}(t)) + c_2 rand2(p_{id} - x_{id}(t)) \tag{2}$$

$$x_{id}(t+1) = x_{id}(t) + v_{id}(t+1) \tag{3}$$

where t is number of iterations, c_1 and c_2 are acceleration coefficients, *rand*1 and *rand*2 are random number in range [0, 1], w is inertia weight. The inertia weight, w is updated according to the following equation due to reduction in the weight over number of iterations [32]:

$$w = w_{\max} - \frac{w_{\max} - w_{\min}}{iter_{\max}} \times t \tag{4}$$

where $w_{\max}$, $w_{\min}$ are the maximum and minimum values of inertia weight while $iter_{\max}$ is the maximum number of iterations. Figures 10 and 11 presents the velocity and position updates for PSO in two dimensional parameter space and the computational flowchart of PSO algorithm, respectively.

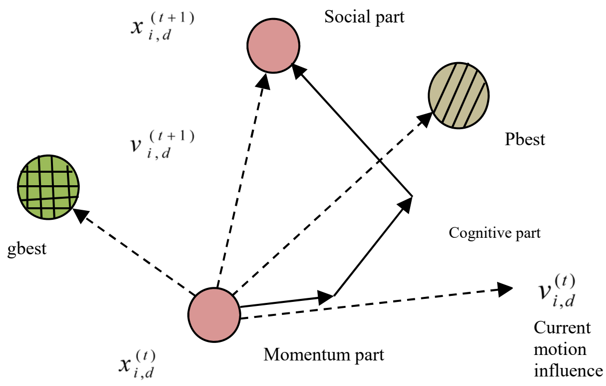


Figure 10: description of velocity and position updates in PSO for a two dimensional parameter space

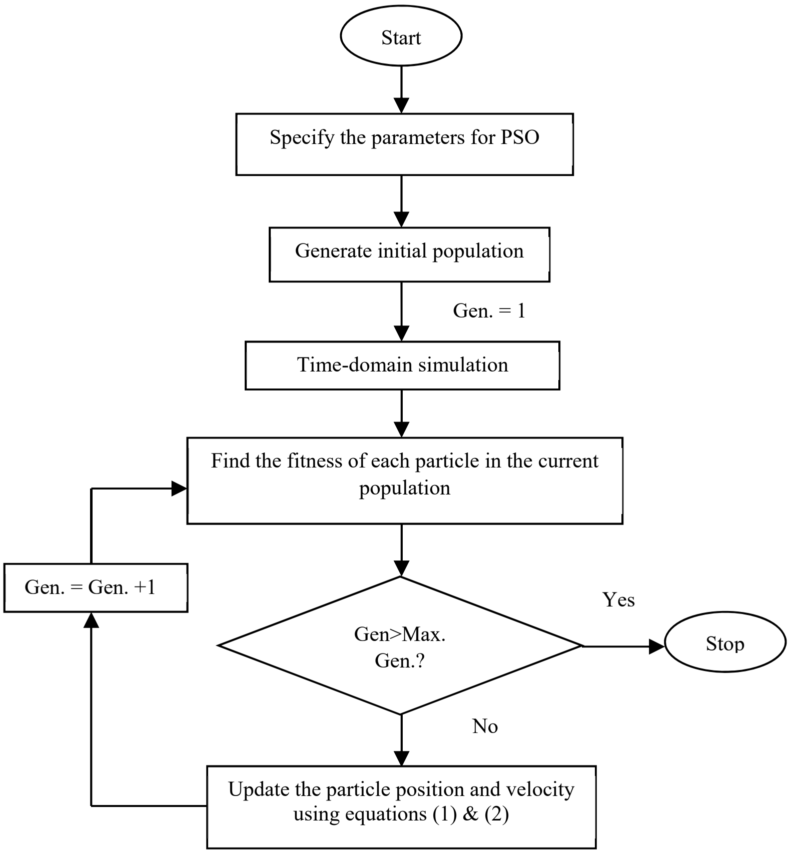


Figure 11: The computational flowchart of PSO algorithm

5. SIMULATION RESULTS AND ANALYSIS

In this paper, PID controller was tuned using ZN and PSO methods. The performances for both controller will be compared and explained briefly in this section. Two different disturbance was introduced into the system as means of validation of the developed controller. The disturbance used in this study is known as multiple sinusoidal and multiple real disturbance. For ZN method, the value of K_P, K_I and K_D will be obtained through the calculations based on ZN formulation as tabulated in Table 2. For PSO method, the tuning was initialized by setting the number of iterations to 100 and varying the number of particles from 10 to 100. Then, the same procedure was repeated by varying the number of iterations and fixing the value of number of particles to 40 since this number of particles achieved the best attenuation during the first tuning session.

The parameters tuning result for proposed controllers using multiple sinusoidal disturbance in this paper using ZN and PSO algorithm methods were elucidated in Table 3. By referring to Table 3, PSO algorithm achieved the lowest MSE as compared to ZN method for vibration suppression in a horizontal flexible plate system. The MSE achieved by the PID controller tuned by PSO algorithm (PID-PSO) is 0.0734 while the MSE for PID controller tuned by ZN method (PID-ZN) is 0.0216. The attenuation level and percentage reduction in vibration suppression were achieved by both controllers as described in Table 4. From Table 4, it can be revealed that the PID controller tuned by PSO algorithm achieved the highest attenuation level at the first mode of vibration which is the dominant mode in the horizontal plate system.

This can be further illustrated in Figure 12 and 13, where thevibration suppression through the proposed controllers in this study using multiple sinusoidal disturbance in time and frequency domains are elucidated, respectively. Figures 14 and 15 shows the enlarged view of attenuation achieved for both controllers in time and frequency domains, respectively. The PID-PSO controller provided better vibration suppression in the horizontal plate system by achieving an attenuation of 47.36 dB at the first mode of vibration in the system with 40.37 % percentage of reduction, while, the PID-ZN controller only achieving 34.30 dB attenuation with 29.24 % reduction. From the results, it was shown that the evolutionary algorithm using PSO proposed in this paper successfully suppressed higher unwanted vibration in the horizontal flexible plate system by achieving higher percentage of reduction in vibration as compared to conventional method via ZN tuning rules.

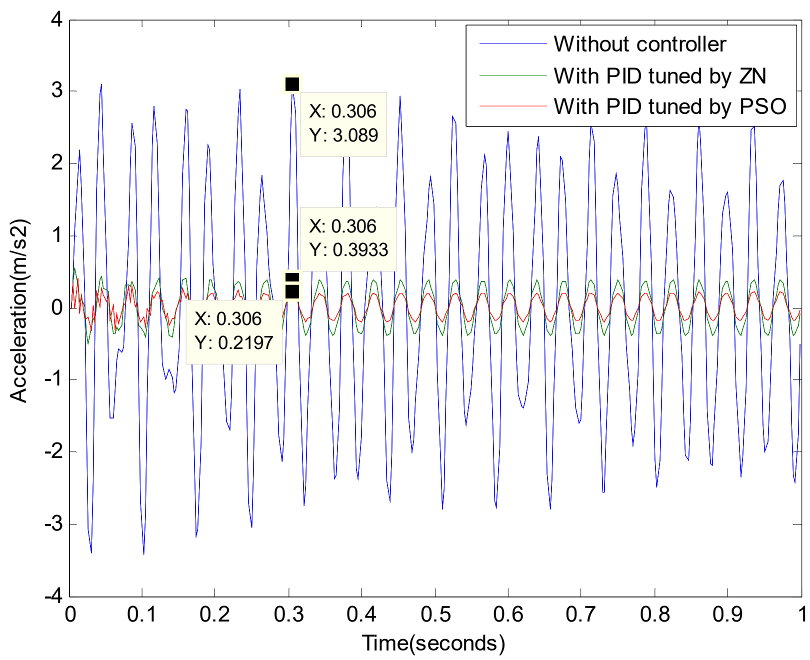


Figure 12: PID controller tuned by ZN and PSO methods for vibration suppression in time domain using multiple sinusoidal disturbance

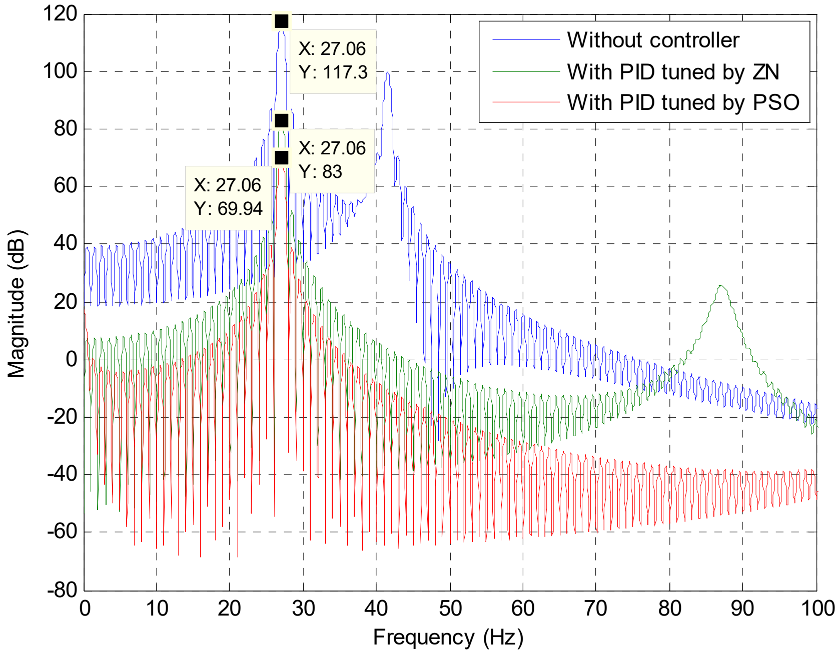


Figure 13: PID controller tuned by ZN and PSO methods for vibration suppression in frequency domain using multiple sinusoidal disturbance

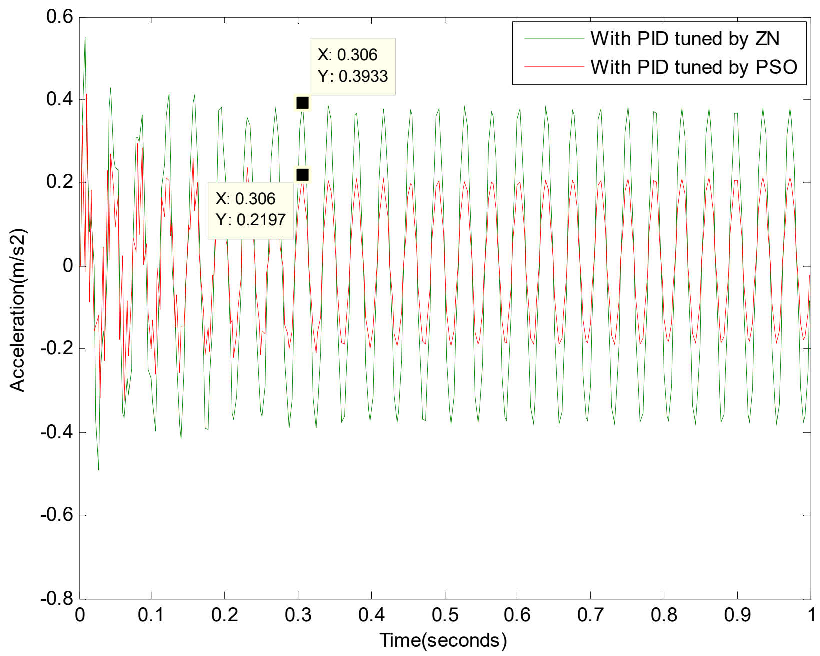


Figure 14: Comparison in time domain between PID-ZN controller and PID-ABC controller with multiple sinusoidal disturbance

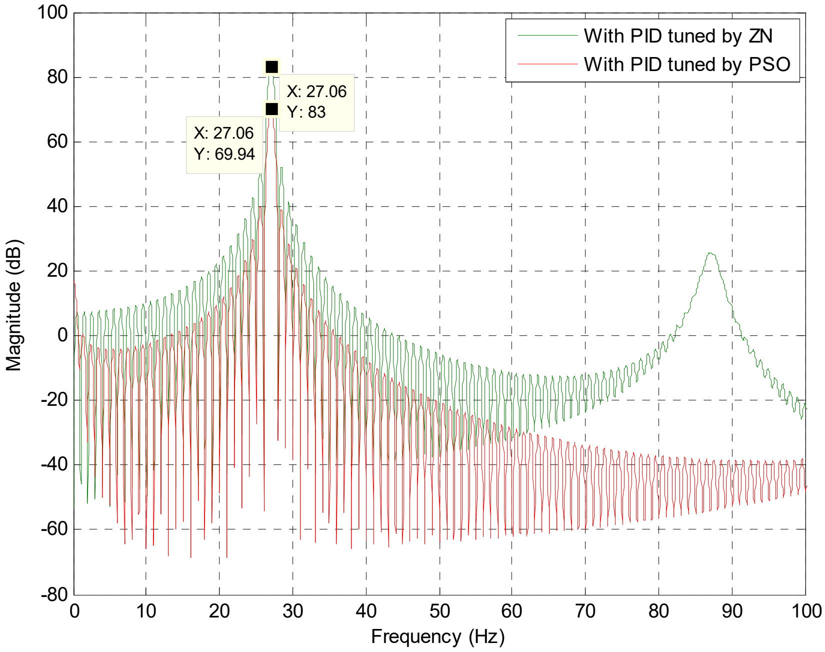


Figure 15: Comparison in frequency domain between PID-ZN controller and PID-ABC controller with multiple sinusoidal disturbance

Table 3: The PID parameters and MSE obtained tuned by Ziegler-Nichols and PSO algorithm methods using multiple sinusoidal disturbance

Controller	Parameters Gains			Mean Squared Error (MSE)
	K_p	K_i	K_d	
Without Controller	-	-	-	2.6619
Ziegler-Nichols (ZN)	3.9	0.02	0.005	0.0734
PSO	8.732	-21.07	0.00455	0.0216

Table 4: The attenuation level and reduction percentage of the proposed controllers using multiple sinusoidal disturbance

Controller	Decibel Magnitude (dB)	Attenuation Level (dB)	Percentage of Reduction
	1st mode	1st mode	1st mode
Without Controller	117.30	reference	reference
Ziegler-Nichols (ZN)	83.00	34.30	29.24 %
PSO	69.94	47.36	40.37 %

The robustness of the developed controller was then test by employing multiple real disturbance to the system. Table 5 describes the MSE achieved for both controllers using multiple real disturbance. Here, it was shown that PID-PSO controller has obtained the lowest MSE in the system at 0.0216 as compared to PID-ZN which achieved the MSE value of 0.0734. The attenuation level and percentage reduction in vibration suppression were achieved for both controllers as described in Table 6. From Table 6, it was revealed that the PID controller tuned by PSO algorithm achieved the highest attenuation level at the first mode of vibration which is the dominant mode in the horizontal plate system. This can be further illustrated in Figure 16 and 17 for vibration suppression through the proposed controllers using multiple real disturbance in time and frequency domains, respectively. Figures 18 and 19 show the enlarged view of the attenuation achieved for both controllers in time and frequency domains, respectively.

The PID-PSO controller provided better vibration suppression in the horizontal plate system by achieving 47.24 dB of attenuation at the first mode of vibration in the system with 70.69 % reduction, while, the PID-ZN controller only achieved 34.21 dB of attenuation with 51.19 % reduction. From the results presented, it was clearly observed that PID controller proposed in this study can reduce unwanted vibration even when different disturbances were employed on the system. It was proven that the proposed controller is very robust and can work effectively for vibration suppression in the horizontal plate system. Both proposed methods successfully reduced unwanted vibration. However, PID-PSO was more effective in tuning PID parameters in this study to suppress unwanted vibration in the horizontal flexible plate system by achieving higher level of attenuation in the first mode of vibration.

Table 5: The PID parameters and MSE obtained tuned by Ziegler-Nichols and PSO algorithm methods using multiple real disturbance.

Controller	**Parameters Gains**			**Mean Squared Error (MSE)**
	K_p	K_i	K_d	
Without Controller	-	-	-	2.6619
Ziegler-Nichols (ZN)	3.9	0.02	0.005	0.0734
PSO	8.732	-21.07	0.00455	0.0216

Table 6: The attenuation level and reduction percentage of the proposed controllers using multiple real disturbance.

Controller	**Decibel Magnitude (dB)**	**Attenuation Level (dB)**	**Percentage of Reduction**
	1st mode	**1st mode**	**1st mode**
Without Controller	66.83	reference	reference
Ziegler-Nichols (ZN)	32.62	34.21	51.19 %
PSO	19.59	47.24	70.69 %

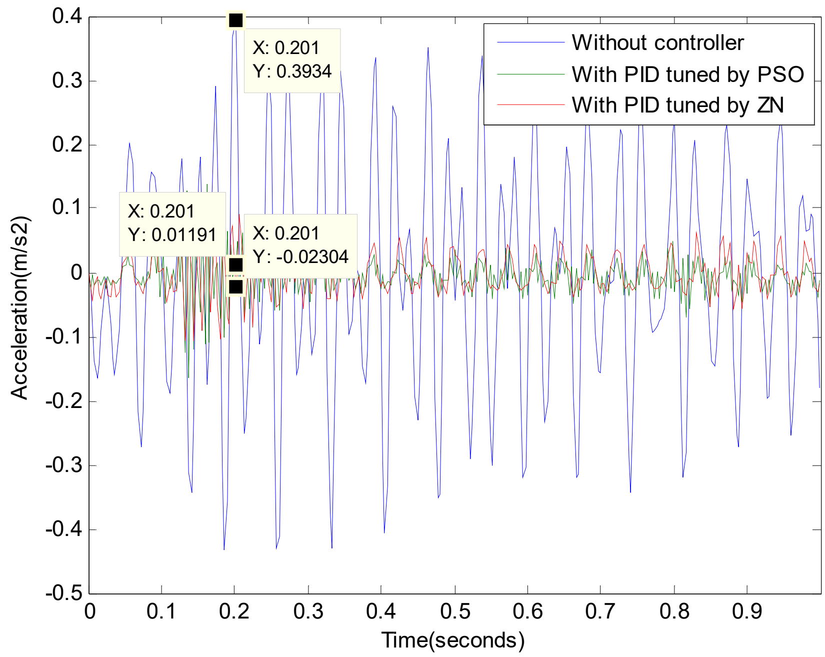


Figure 16: PID controller tuned by ZN and PSO methods for vibration suppression in time domain using multiple real disturbance.

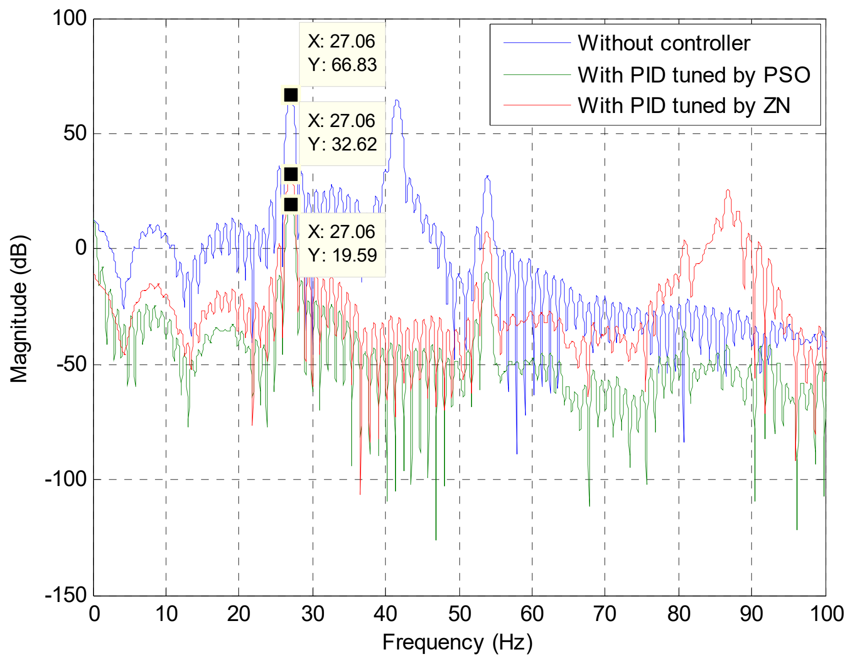


Figure 17: PID controller tuned by ZN and PSO methods for vibration suppression in frequency domain using multiple real disturbance.

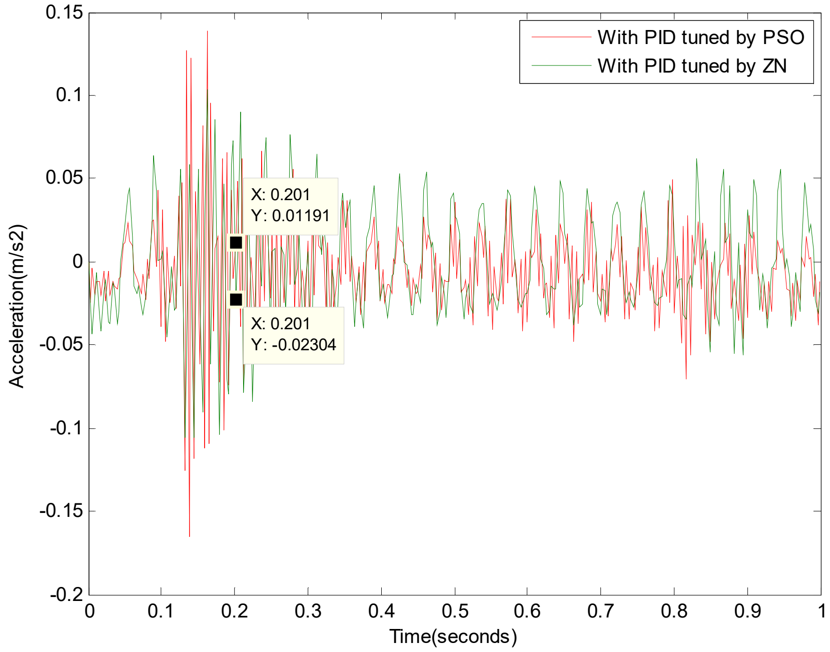


Figure 18: Comparison in time domain between PID-ZN controller and PID-ABC controller with multiple real disturbance.

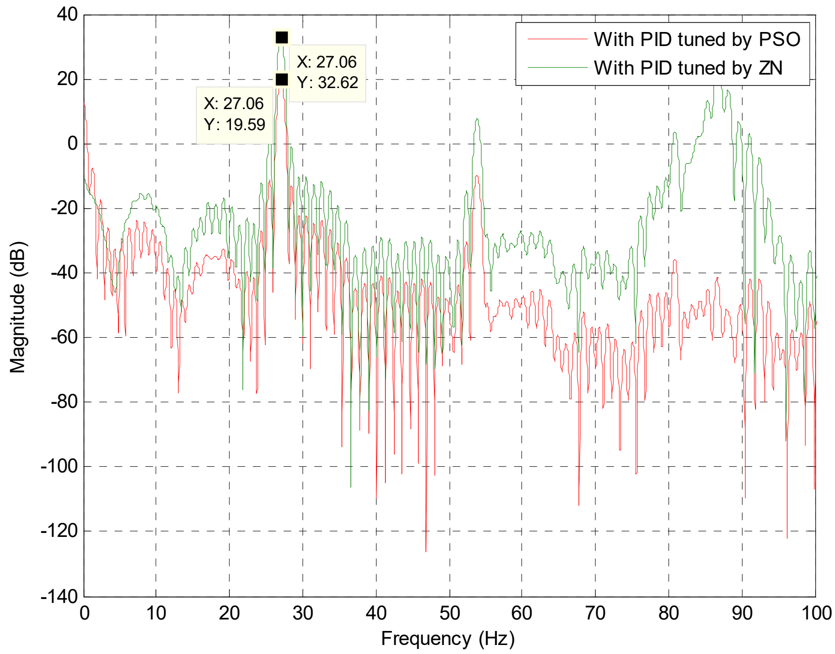


Figure 19: Comparison in frequency domain between PID-ZN controller and PID-ABC controller with multiple real disturbance.

As a validation of the developed simulated controller, the best controller acquired was experimentally tested. The optimal value of PID parameters tuned by PSO was tested on the experimental rig and the performance of the controller was evaluated. A sinusoidal signal at the frequency of 27 Hz and amplitude of 1V was employed as the disturbance signal to excite the experimental rig while conducting the experiment. The controller was utilised in a real-time computer control system using MATLAB/Simulink with a sampling rate of 0.003 s. Figure 20 shows the schematic diagram of the controller implemented in the experiment. Next, Figure 21 presents the block diagram of the controller implemented by MATLAB/Simulink.

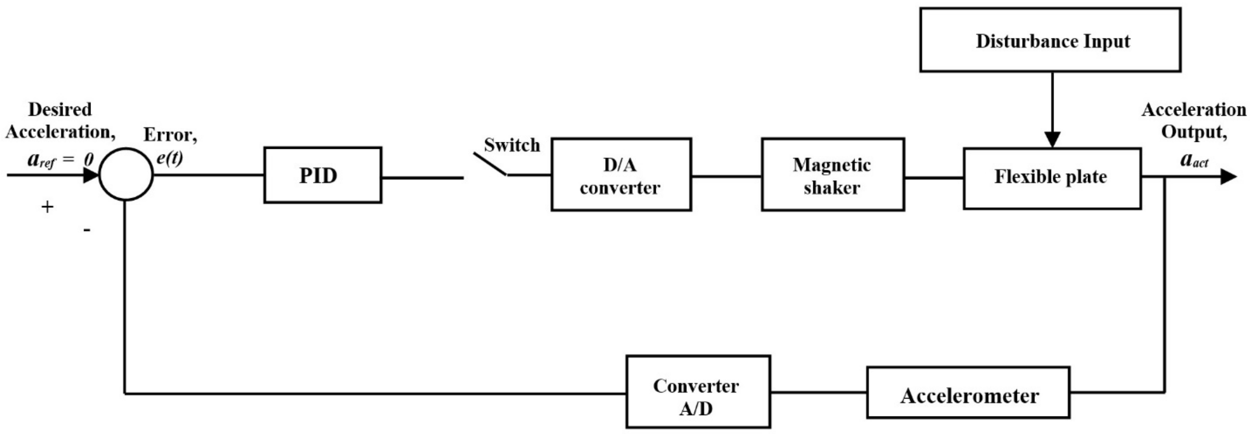


Figure 20: The schematic diagram of the controller used in the experiment

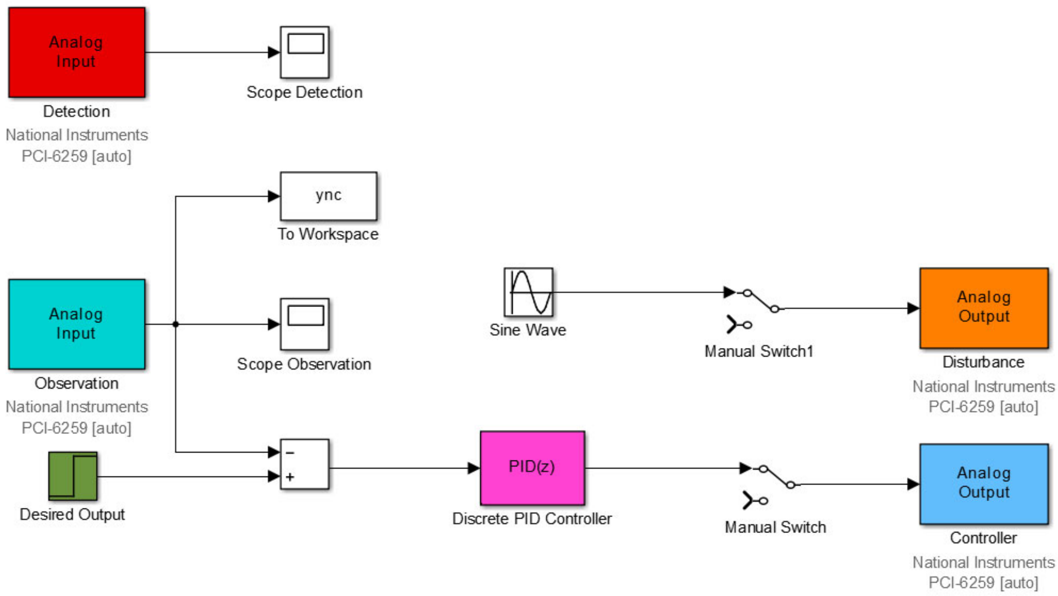


Figure 21: Block diagram of PID based controller using MATLAB/Simulink

Figures 22 and 23 illustrate the experimental performances of PID based controller tuned by PSO in time and frequency responses, respectively. It can be noticed that both PID-PSO based controllers used in this study successfully attenuated the unwanted vibration of horizontal flexible plate system. Likewise, the controller has achieved an acceptable performance in both time and frequency responses, respectively. The performance result of the controller is summarised in Table 7.

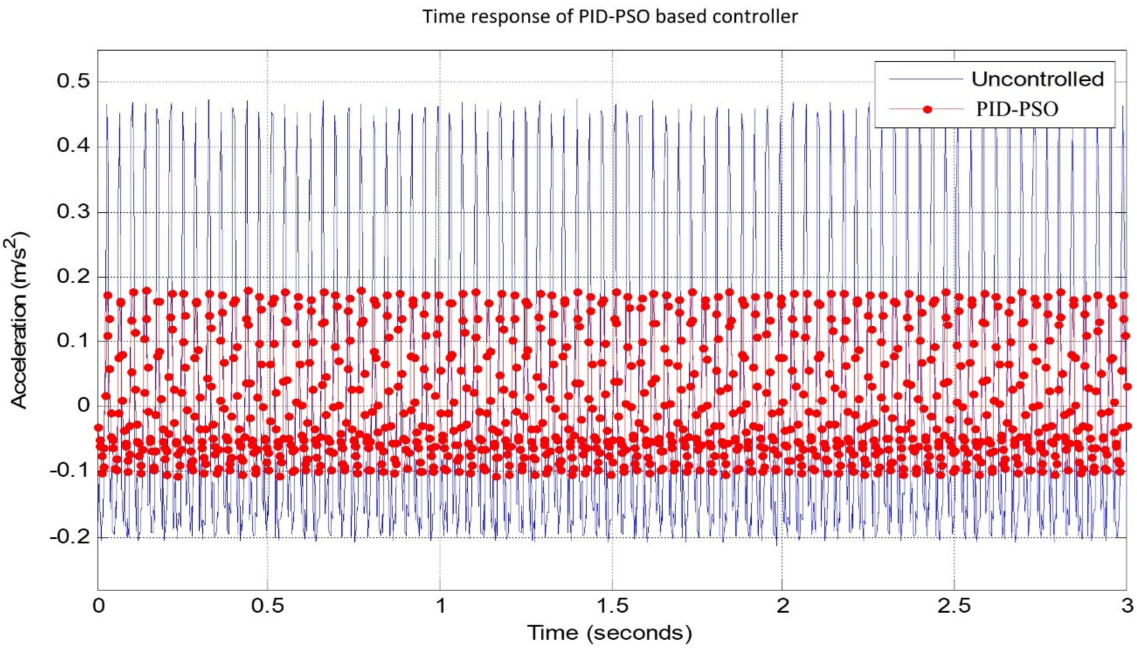


Figure 22: Experimental time response of PID based controller tuned by PSO algorithm

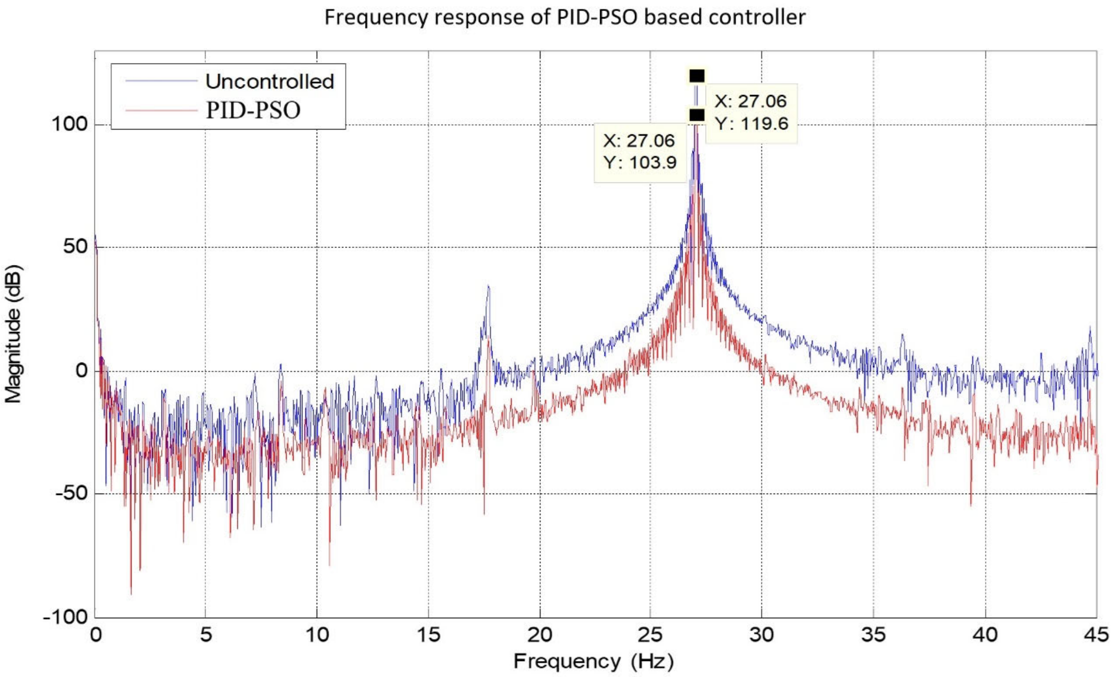


Figure 23: Experimental frequency response of PID based controller tuned by PSO algorithm

From Table 7, PID-PSO based controller has achieved 15.70 dB of attenuation at the first mode of vibration. The attenuation value has been reduced from 119.6 dB to 103.9 dB, which is equivalent to 13.13 % of attenuation, after the introduction of vibration control. Furthermore, the MSE achieved by PID-PSO is 0.0026, compared with 0.0168 before the controller activation. From these results, conclusion can be made that PID-PSO based controller has succeeded in making remarkable vibration suppression for horizontal flexible plate system. Moreover, the PSO algorithm has successfully tuned the PID controller in searching for best quality solutions. Hence, this result confirmed the simulation part that shows PID-PSO is the superior controller in suppressing the unwanted vibration as compared to the other controllers.

Table 7: The attenuation level and reduction percentage of PID-PSO based controller in experimental performance

Controller	Decibel Magnitude	Attenuation Level	Percentage of Reduction	MSE
	1st mode	1st mode	1st mode	
Without controller	119.6 dB	-	-	0.0168
PID-PSO	103.9 dB	15.70 dB	13.13 %	0.0026

6. CONCLUSION

In this paper, the development of parametric modeling using PSO algorithm and active vibration control for vibration suppression in the horizontal flexible plate system using conventional method via ZN tuning rules and intelligent algorithm via PSO was carried out. The model was developed based on Auto-regressive with exogenous (ARX) input structure and used to identify the parameters. Then, the estimated output was validated using one step-ahead prediction, correlation tests and pole-zero diagram stability. The vibration modes of the flexible plate structure was successfully detected which led to the development of an effective controller design. The performance of the developed PID controller tuned by ZN and PSO algorithm has been considered for horizontal flexible plate system. Based on the simulation result using multiple sinusoidal disturbance, the PID-PSO controller exhibited better performance compared to the PID-ZN controller. Here, the PID-PSO was able to successfully suppress vibration by 40.37 %. The performance of the developed controllers were also validated and assessed using different types of disturbances. They include multiple real disturbance, where the PID-PSO achieved higher reduction of vibration of 70.69 % compared to PID-ZN. Based on the results elucidated, it can be deduced that the PID-PSO controller successfully suppressed the vibration on horizontal flexible plate system. As a validation for the developed simulated controller, the best controller achieved was experimentally tested in this study. Likewise, the controller achieved an acceptable level of performance in both time and frequency responses, respectively.

Conflict of Interests

There are no conflicts of interest

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