

PAPER • OPEN ACCESS

Application of SA-PSO Algorithm in Parameter Optimization of Dynamic Vibration Absorber

To cite this article: Chen Zhe-ming *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **267** 042174

View the [article online](#) for updates and enhancements.

Application of SA-PSO Algorithm in Parameter Optimization of Dynamic Vibration Absorber

Chen Zhe-ming^{1,5}, Wang Peng², Fu Jiang-hua³ and Chen Bao⁴

¹Department of Vehicle Engineering, Chongqing University of Technology, No. 69 Hongguang Avenue, Banan District, Chongqing, P.R.China

²Department of Vehicle Engineering, Chongqing University of Technology, No. 69 Hongguang Avenue, Banan District, Chongqing, P.R.China

³Department of Vehicle Engineering, Chongqing University of Technology, No. 69 Hongguang Avenue, Banan District, Chongqing, P.R.China

⁴Department of Vehicle Engineering, Chongqing University of Technology, No. 69 Hongguang Avenue, Banan District, Chongqing, P.R.China

⁵Fund project: project of Chongqing Science and Technology Commission (cstc2018jcyjAX0109)

E-mail: czm_415@qq.com

E-mail: 812827642@qq.com

Abstract: Focus on the noise problem in a 7-segment midsize SUV in the condition of wide open throttle in gear 2, noise testing and finite element analysis were combined to analyze the problem, and it was found that there was a significant resonance band around the frequency of 214 Hz, which was particularly evident under the fourth-order excitation. The frequency of resonance band is close to the second-order modal natural frequency of the sub-frame, The resonance band needs to be eliminated to reduce the noise inside the car. Adopting the scheme of installing a dynamic vibration absorber on the sub-frame to make the second-order modal of the sub-frame avoids the resonance frequency. The simulated annealing algorithm is combined with the particle swarm optimization algorithm to optimize the parameters of the dynamic vibration absorber. A dynamic vibration absorber was designed with optimized parameters and verified by simulation and installed on the car. It was verified that The designed dynamic vibration absorber eliminates the 214Hz resonance band and obviously reduce the noise inside the car.

1. Introduction

As people's requirements for comfort of car are getting higher and higher, low-vibration, low-noise cars are often more attractive to customers, and major car manufacturers are paying more and more attention to the NVH performance of cars. The main cause of high noise and obvious vibration of the vehicle is the resonance of the automobile parts. There are mainly three schemes to eliminate the resonance band at this frequency as shown below:(1)The resonance will be eliminated by changing the structure and mass distribution of the engine. This scheme changes the natural frequencies of each order of the automobile engine so as to avoid the natural frequencies of the parts, but the cycle of this scheme is



longer and more difficult; (2) Changing the structure of the part, thereby changing the natural frequency of each modal of the part, however, this scheme changes the structure of the part and thus affected other properties of the car. (3) A dynamic vibration absorber is added to the automobile component, and the reaction force generated by the dynamic vibration absorber can reduce the vibration of the active part, thereby reducing the vibration noise of the automobile in a specific frequency band. The dynamic vibration absorber technology was first proposed by Frahm H^[1], and its research has been studied from single degree of freedom to multiple degrees of freedom^[2], passive to active^[3-5]. Because the passive dynamic vibration absorber can achieve the purpose of improving the resonance of a certain frequency segment without changing the structure of the part, and has the advantages of low cost, simple structure and easy installation, it becomes an important vibration damping means.

In this paper, the noise problem of a 7-segment midsize SUV in the condition of wide open throttle in gear 2 is analyzed. The reason of the noise in the car was found out by using the method of finite element analysis and modal experiment, the reason is that the sub-frame causes a resonance band, in order to solve the problem, a method of installing a dynamic vibration absorber on a sub-frame is proposed. The dynamic vibration absorber is designed and optimized by using the simulated annealing particle swarm optimization algorithm, and made a sample, mounted it on the sub-frame, and finally the test verified that the method can effectively solve the problem of noise inside the car.

2. Analysis of car noise

2.1 problem diagnosis

The SUV is equipped with a 1.5T four-stroke engine. Under the condition of wide open throttle in gear 2, when the engine speed is around 3200 rpm, there is a large noise in the car, which affects the passenger's ride comfort. In order to solve the problem of excessive noise inside the car, the LMS test system is used to do a vibration and noise test, and a microphone is placed at the driver's right ear. It can be seen from the colormap of the interior noise of Fig 1 that there is an obvious resonance band at 213.9 Hz, which is extremely obvious under the fourth-order excitation, so measures should be taken to avoid the resonance band of this frequency.

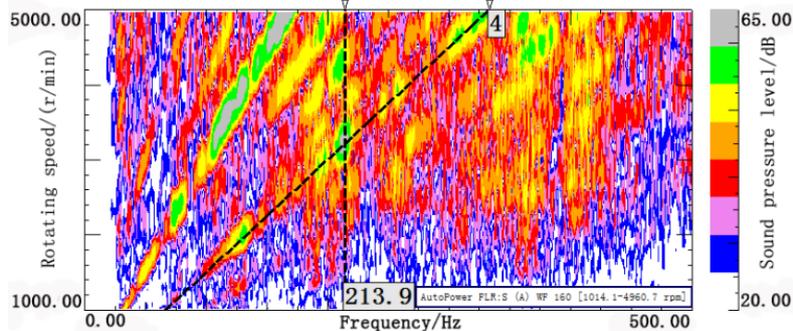


Fig 1. Interior Noise colormap

2.2 modal analysis

After a series of systematic investigations on the intake and exhaust system and the powertrain suspension system, it was found that the Resonance is caused by vibration of the sub-frame. The modal of the sub-frame was tested by LMS Test.Lab software, and found the second-order modal natural frequency is 214.2Hz. On this resonance band, it can be determined that the interior noise was caused by the resonance of the fourth-order excitation frequency and the second-order modal natural frequency of the sub-frame. The modal shape is shown in Fig 2.

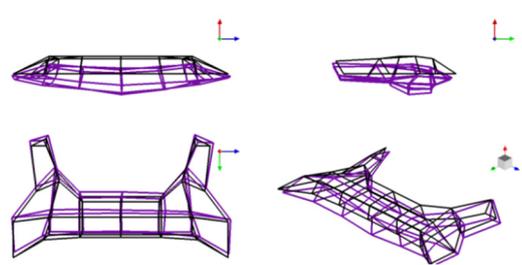
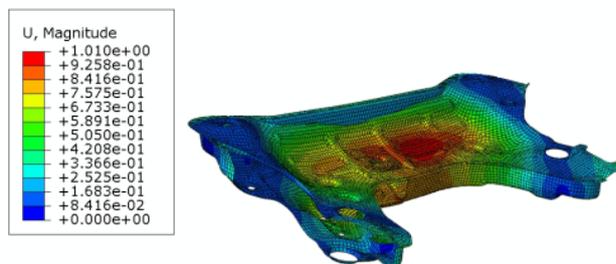


Fig 2.Second-order modal shape of sub-frame

To verify the accuracy of the test, the finite element method is used for verification. The material properties of the sub-frame are defined according to the actual material: elastic modulus is 210000 MPa, Poisson's ratio is 0.3, density is $7850 \text{ kg} \cdot \text{m}^3$, the thickness of each component modal is defined according to the actual thickness value, modal analysis and post-processing of the sub-frame are applied by ABAQUS. Due to the first six orders are rigid body modal, the seventh order is the first-order modal, and found the second-order modal frequency is 212.85Hz, which is basically consistent with the frequency of fourth-order in-vehicle noise 214Hz resonance band. Resonance will occur at the frequency. The specific modal frequency is shown in table 1, and the second-order modal cloud image is shown in Fig.3.

Table 1.Modal frequency of sub-frame

Order	frequency(Hz)
1	187.98
2	212.85
3	233.03



X ODB: Job-1.odb Abaqus/Standard 6.13-1 Thu Apr 19 17:05:43 GMT+08:00 2018
 Y Step: loadstep1
 Z Mode 8: Value = 1.45928E+06 Freq = 212.85 (cycles/time)
 Primary Var: U, Magnitude

Fig 3.Second-order modal shape cloud map of Sub-frame

Therefore, it is confirmed that the roaring sound of the engine speed around 3200 r/min is caused by the second-order modal of the sub-frame, that is, the fourth-order excitation frequency of the engine is the same as the second-order modal natural frequency of the sub-frame, and resonance occurs, producing roaring sounds in the car.

The scheme of installing a dynamic vibration absorber on the sub-frame can not only solve the fourth-order noise in the vehicle but also does not change the sub-frame structure. The principle of the dynamic vibration absorber is to change the vibration state of the main vibration system by mounting a vibration absorber on the main vibration system, appropriately selecting the structural form of the vibration absorber, the dynamic parameters and the coupling relationship with the main vibration system, so as to reduce the forced vibration response of the main vibration system in the expected frequency band^[6].

3.parameter optimization of absorber

Properly select the parameters of the dynamic vibration absorber can make the absorber absorb most of the energy of the main vibration system, and can limit the amplitude of the system within a certain range,

so that the resonance peak is small to achieve the purpose of vibration reduction. With the development of intelligent algorithms, more and more intelligent algorithms are applied to the parameters optimization of dynamic vibration absorbers, such as genetic algorithm and particle swarm optimization algorithm, but the genetic algorithm has convergence and mutation operations, it's convergence speed is slower than the particle swarm optimization algorithm, and the particle swarm optimization algorithm is easy to fall into the local optimal solution, so it is necessary to find a new intelligent optimization algorithm. In this paper, the simulated annealing particle swarm optimization algorithm is used to optimize the parameters, and study the influence of various factors on the damping effect under the intelligent optimization algorithm.

3.1 Mathematical Model

The dynamic vibration absorber and main system can be simplified to the two-degree-of-freedom vibration system shown in Fig 4^[7].

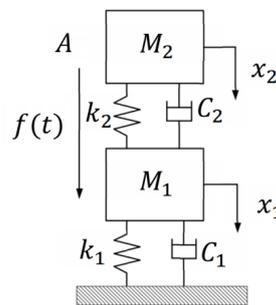


Fig 4. Two-degree-of-freedom vibration system model

The vibration differential equation can be expressed as:

$$\left. \begin{aligned} m_1 \ddot{x}_1 + (c_1 + c_2) \dot{x}_1 - (c_1 + c_2) \dot{x}_2 + (k_1 + k_2)x_1 - k_2 x_2 &= F_1 \sin \omega t \\ m_2 \ddot{x}_2 - (c_1 + c_2) \dot{x}_1 + (c_1 + c_2) \dot{x}_2 - k_2 x_1 + k_2 x_2 &= 0 \end{aligned} \right\} \quad (1)$$

Solving the equation (1) can obtain the expression equation of the amplitude amplification factor A of the system, it can be expressed as:

$$A = \left(\frac{B^2 + C^2}{D^2 + 4\lambda^2 E^2} \right)^{1/2} \quad (2)$$

Each parameter is expressed as:

$$B = 2\eta_2 f \lambda$$

$$B = 2\eta_2 f \lambda$$

$$C = \lambda^2 - f^2$$

$$D = (1 - \lambda^2)(f^2 - \lambda^2) - \mu \lambda^2 f^2 - 4\eta_1 \eta_2 f \lambda$$

$$E = \eta_2 f - \eta_2 f \lambda^2 (1 - \mu) + \eta_1 (f^2 - \lambda^2)$$

Where $\lambda = \frac{\omega}{\omega_1}$ is the system excitation frequency ratio, $f = \frac{\omega_2}{\omega_1}$ is the tuning ratio of the dynamic vibration absorber to the main system, and $\omega_1 = \left(\frac{k_1}{m_1} \right)^{1/2}$ is the natural frequency of the main system, $\omega_2 = \left(\frac{k_2}{m_2} \right)^{1/2}$ is the natural frequency of the dynamic vibration absorber, $\mu = \frac{m_2}{m_1}$ is the quality ratio, $\eta_1 = \frac{c_1}{2(m_1 k_1)^{1/2}}$ is the damping ratio of the main system, $\eta_2 = \frac{c_2}{2(m_2 k_2)^{1/2}}$ is the damping ratio of the dynamic vibration absorber.

3.2 Objective function

According to the theory of Damping damper^[8], the curve of system amplitude amplification factor has two peaks when the parameters are optimal, and the two peaks are equal at this time. Therefore, designing an algorithm to obtain the maximum value A_{maxi} of the curve of the amplitude amplification factor of the i times iteration, and to search for the combination of damping ratio and coordination ratio

that minimizes A_{maxi} in all iterations.

The design objective function is:

$$A_0 = \min \left\{ \max_{\lambda \in (0.5, 1.5)} A(\eta_2, f) \right\} \quad (3)$$

3.3 Parameter optimization

The PSO method first initializes a bunch of random particles (random solutions) and extends them into the D-dimensional space. It is assumed that the position of the particle i in the D-dimensional space is expressed as a vector: $X_i = (x_1, x_2, \dots, x_D)$, and the flying speed is expressed as a vector: $V_i = (v_1, v_1, \dots, v_D)$. Each particle has a suitable value determined by the objective function, and knows the best position: $P_i = (p_{i1}, p_{i2}, \dots, p_{iD})$ that it has found so far and the current position. This can be seen as the particle's own flight experience. In addition, each particle knows the best position found by all particles in the entire population so far: $P_g = (p_{g1}, p_{g2}, \dots, p_{gD})$ (P_g is the best value in P_i). This can be seen as the experience of companions. Particles determine the next move through their own experience and the best experience of their companions^[9].

Although the particle swarm optimization algorithm has the characteristics of simple structure, easy realization and fast convergence, it is easy to fall into local convergence. In this paper, the simulated annealing particle swarm optimization algorithm is used to optimize the parameters of dynamic vibration absorber. The algorithm combines the strong ability of the simulated annealing algorithm^[10] to jump out of the local optimal solution and the advantages of the fast optimization of the particle swarm optimization algorithm to improve the global optimization ability and algorithm accuracy.

The simulated annealing algorithm is derived from the principle of solid annealing, given a higher initial temperature, and with the temperature drop, combined with jumping of the time-varying and finally zero-probability jump out of the local optimal solution to find the global optimal solution. The solution process of simulated annealing particle swarm optimization algorithm is as follows:

- (1) Initializing the position and velocity of a group of random particles;
- (2) Calculating the suitable value of each particle, and storing the individual extremum of each particle in p_i , and store the population extremum in p_g ;
- (3) Setting the initial temperature T ;
- (4) Calculating the suitable value of each p_i at the current temperature according to the following equation:

$$TF(p_i) = \frac{e^{-(f(p_i)-f(p_g))/t}}{\sum_{i=1}^N e^{-(f(p_i)-f(p_g))/t}} \quad (4)$$

- (5) Using the roulette method to find a globally optimal alternative value p'_g from all p_i , and then iteratively update the speed and position of each particle according to equation (5) and equation (6)

$$v_i = \varphi \{ wv_i + c_1 r_1 (p_i - x_i) + c_2 r_2 (p_g - x_i) \} \quad (5)$$

$$x_i = x_i + v_i \quad (6)$$

$$\text{Where } \varphi = \frac{2}{|2 \cdot C - \sqrt{C^2 - 4C}|}, \quad C = c_1 + c_2;$$

- (6) Calculating the new suitable value of each particle, and updating the individual extreme value p_i and the population extreme value p_g of each particle;

- (7) Performing a cooling operation according to equation (7);

$$T_{k+1} = \lambda T_k \quad (7)$$

- (8) Determining if it meets the termination iteration conditions of the preset (usually the operation precision or the number of iterations), and if it is satisfied, the search is stopped, and the result is output; if not, go to fourth process and re-iterate.

4 Simulation and test verification

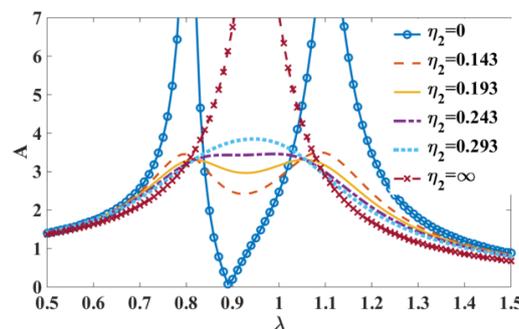
4.1 Simulation verification

Designing the mass ratio μ and the main system damping ratio η_1 of the dynamic vibration absorber, the damping ratio η_1 of main system is 0.05, and, the better the mass ratio μ is, the better the damping effect is, however, the mass of the vibration absorber will be limited by the installation space, and the general mass ratio μ is less than 0.1, so the mass ratio μ is chosen to be 0.1, As a known quantity input optimization algorithm, the number of particles of the simulated annealing particle swarm optimization algorithm is set to 40, the number of iterations to 400, and the peak of the curve of the mplitude amplification factor as the optimization result. Obtaining the amplitude amplification factor: $A_0 = 3.3095$, $\eta_2 = 0.193$, $f = 0.8926$.

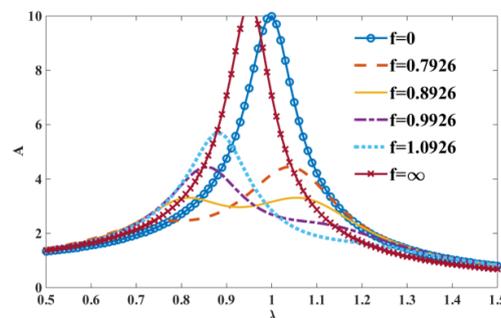
Through the influence of different dynamic vibration absorber parameters on the amplitude amplification factor of system, it can be found that curve A of the amplitude amplification factor of system and the coordination ratio f , the dynamic vibration absorber damping ratio η_2 have the following rules, as shown in Figure 5:

(1)The influence of the damping ratio η_2 of the dynamic vibration absorber on the curve of the amplitude amplification factor of the system: It can be seen from Fig 5(a) that as the η_2 increases, the peaks on both sides gradually decrease and move closer to the middle, when η_2 exceeds the optimal damping ratio η_{opt} , the peak becomes one and increases as η_2 increases;

(2)Influence of coordination ratio f on the curve of the amplitude amplification factor of the system: It can be seen from Fig 5(b) that the change of the coordination ratio affects the position of two common points. When f is gradually increased from 0 to the optimal coordination ratio f_{opt} , The peak on the right side of the curve gradually decreases and the peak on the left side gradually rises until the peaks on both sides are equal. When f is greater than f_{opt} , the left peak exceeds the right peak and gradually increases.



(a) Effect of damping ratio on amplitude amplification factor curve



(b) Effect of tuning ratio on amplitude amplification factor curve

Fig 5. Effect of dynamic vibration absorber parameters on amplitude amplification factor curve

4.2 Test verification

The mass of the sub-frame is 10.72kg. According to the actual situation, the mass of the vibration

absorber is designed to be 0.95 kg, and the mass ratio μ is 0.089. The damper ratio η_1 of the main system can be obtained by modal test, the optimal dynamic vibration absorber damping ratio η_{opt} and the optimal dynamic vibration absorber damping ratio η_{opt} can be obtained by applying the simulated annealing particle swarm optimization algorithm, f_{opt} is 0.9151, η_{opt} is 0.1801; and the natural frequency ω_2 of the dynamic vibration absorber can be calculated as 196Hz by $\eta_2 = \frac{c_2}{2\sqrt{m_2k_2}}$, $\omega_2 = \sqrt{\frac{k_2}{m_2}}$. The damping c_2 is 10.84N · m/s. Make a prototype of the dynamic vibration absorber, and install it at the maximum vibration on the sub-frame, as shown in Figure 6. The test conditions is wide open throttle in gear 2, The sound pressure level of the noise in the right ear of the driver before and after the installation of the dynamic vibration absorber is shown in Fig.7. The noise in the fourth-order vehicle is obviously improved, and In-vehicle noise sound pressure level reduced by 15.1 dB, the noise reduction effect is very obvious; the colormap diagram is shown in Fig.8. According to the analysis, the designed dynamic vibration absorber eliminates the 214 Hz resonance band.



Fig 6.Dynamic vibration absorber installation diagram

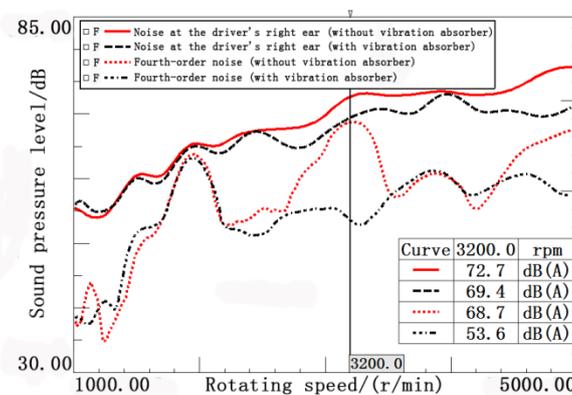
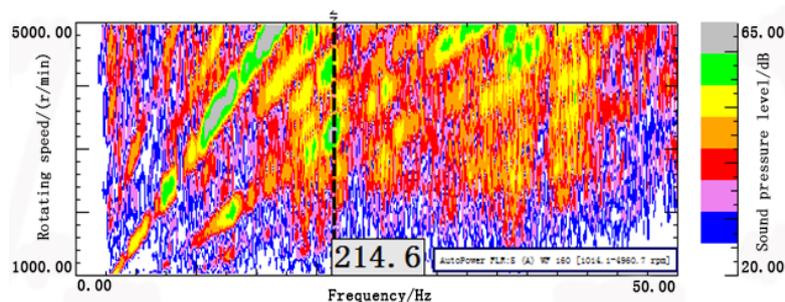


Fig 7.Comparison of sound pressure level of interior noise before and after installation of dynamic vibration absorber



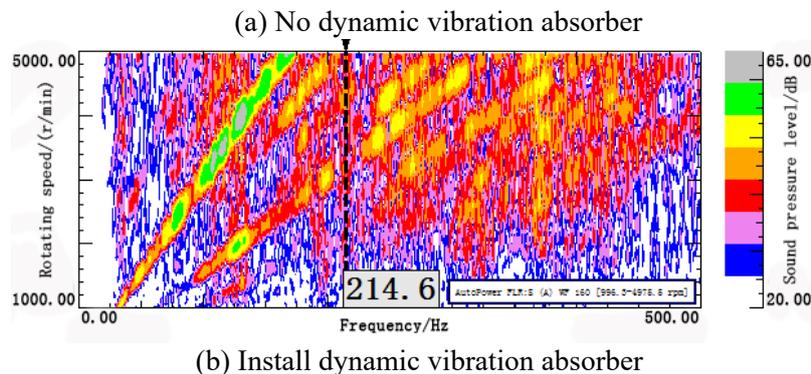


Fig 8. Comparison of the Colormap of Subframe with or without dynamic vibration absorber

5. Conclusion

1) A SUV has a large noise at 3200 rpm under the 2nd WOT. Through a series of experimental tests and finite element modal analysis, It was found that the interior noise was caused by the resonance of the fourth-order excitation frequency and the second-order modal frequency of the sub-frame;

2) Establishing a two-degree-of-freedom mathematical model of the vibration absorber and the main system, designing the objective function, and the simulated annealing particle swarm optimization algorithm is used to optimize the parameters of the dynamic vibration absorber with the mass ratio and the damping ratio of the main system as the input, and the optimal tuning ratio and the optimal damping ratio as the output. Making prototypes and conducting real vehicle tests. The test shows that the designed vibration absorber can greatly reduce the fourth-order noise and the resonance caused by the sub-frame. It shows that the intelligent optimization algorithm designed has certain guiding significance, Considering the engineering cost problem, the program has certain practical engineering significance.

References

- [1] Hermann F 1909 Device for damping vibration of Bodies, US Appl NO. 0989958.
- [2] Love J S, Tait M J 2017 Estimating the added effective damping of SDOF systems incorporating multiple dynamic vibration absorbers with nonlinear damping *Engineering Structures*. **130** p 154-161
- [3] REN M Z 2001 A variant design of the dynamic vibration absorber *Journal of Sound and Vibration*, **245**(4) pp 762-770
- [4] Komatsuzaki T, Inoue T, Terashima O 2016 Broadband vibration control of a structure by using a magneto rheological elastomer-based tuned dynamic absorber *Mechatronics*, **40**
- [5] Li Bin, Niu, Wen-Chao, Xu Zhao-jun 2016 Design and experimental research on eddy-current energy absorber *Journal of Northwestern Polytechnical University*, **34**(1) PP 18-24
- [6] Hu Yin-yin 2013 Development of vibration isolation and vibration absorption technology for power devices *Science*, **14** p 364
- [7] Huang Xiu-jin, He Li-dong, Wang Chen-yang 2015 Study on controlling forced vibration of pipeline by controlled circular dynamic vibration absorber *Journal of Beijing University of Chemical Technology*, **42**(3) pp 88-93
- [8] Zhang Yi-min 2007 *Mechanical Vibration Vol 3* (Beijing: Tsinghua University Press) pp 97-101
- [9] Liu Lang, Chen, Jian-hong, Yang Shan and Han Yu-jian 2011 PSO-BP algorithm based on grey relational analysis for predicting mine danger *Journal of Central South University*, **42**(8) pp 2400-2405
- [10] Kang Li-shan 1997 *Non-numerical parallel algorithm Vol 1* (Beijing: Science Press) pp 22-55