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A PROCEDURE FOR OPTIMAL DESIGN OF A DYNAMIC VIBRATION ABSORBER INSTALLED IN THE DAMPED PRIMARY SYSTEM BASED ON TAGUCHI'S METHOD

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Abstract. The dynamic vibration absorber (DVA) has been widely applied in various technical fields. Using the Taguchi's method, this paper presents a procedure for designing the optimal parameters of a dynamic vibration absorber attached to a damped primary system. The values of the optimal parameters of the DVA obtained by the Taguchi's method are compared by the results obtained by other methods.

Keywords: dynamic vibration absorber, tuned mass damper, damped structures, Taguchi's method.

Classification numbers: 5.4.2; 5.5.3; 5.6.2

1. INTRODUCTION

Taguchi's method for the product design process may be divided into three stages: system design, parameter design, and tolerance design [1-7]. Taguchi's method of parameter design is successfully applied to many mechanical systems: an acoustic muffler, a gear/pinion system, a spring, an electro-hydraulic servo system, a dynamic vibration absorber. In each system, the design parameters to be optimized are identified, along with the desired response.

The dynamic vibration absorber (DVA) or tuned-mass damper (TMD) is a widely used passive vibration control device. When a mass-spring system, referred to as primary system, is subjected to a harmonic excitation at a constant frequency, its steady-state response can be suppressed by attaching a secondary mass-spring system or DVA. Design of DVA is a classical topic [8-12]. The first analysis was reported by Den Hartog [10]. The damped DVA proposed by Den Hartog is now known as the Voigt-type DVA, where a spring element and a viscous element are arranged in parallel, and has been considered as a standard model of the DVA. Thenceforth, the DVA has been widely used in many fields of engineering and construction. The reasons for those applications of the DVA are its efficiency, reliability and low-cost characteristics.

Basically, the solution of optimization problem is the minimization of the maximum vibration amplitude over all excitation frequencies. In the design of the Voigt-type DVA, the main objective is to determine optimal parameters of the DVA so that its effect is maximal. Because the mass ratio of the DVA to the primary structure is usually few percent, the principal parameters of the DVA are its tuning ratio (i.e. ratio of DVA's frequency to the natural frequency of primary structure) and damping ratio.

There have been many optimization criteria given to design DVAs for undamped primary structures [10-12]. The study of optimal design of parameters of dynamic vibration absorber installed in damped primary system becomes interesting problem in recent years [13-20]. In this paper, a procedure for optimal design of the DVA installed in damped primary system based on Taguchi's method is presented and discussed.

The remaining contents of this paper are organized as follows. Section 2 presents briefly the structural mathematical model and the optimization problem. In Section 3, a procedure for optimal design of the DVA parameters for damped primary system is described in detail. In Section 4, some obtained results by Taguchi's method are compared with the results obtained by other methods to verify the proposed procedure. Section 5 includes some concluding remarks and future work proposals.

2. CALCULATION OF VIBRATION OF DAMPED PRIMARY SYSTEM AND DYNAMIC VIBRATION ABSORBER

A system shown in Fig.1 is a dynamic vibration absorber installed in the primary structure. The primary structure includes a main mass m_s, a spring element k_s and a damping element c_s and is subjected an external force $f_e(t) = F_0 \sin \Omega t$. The mass of the DVA is m_a and its spring and damping coefficients are k_a and c_a respectively.

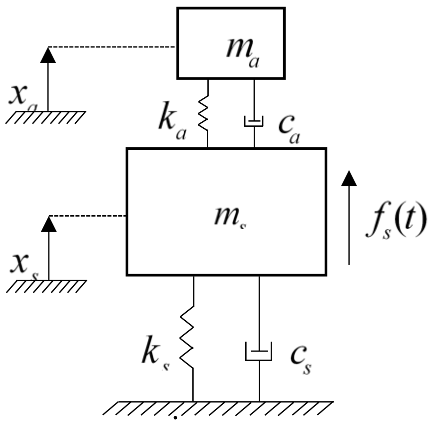


Figure 1. Dynamic vibration absorber applied to a force-excited system with damping.

In the design optimal procedure, the desired response is a level of vibrational amplitude, the control factors are mass ratio μ, damping ratio δ, and tuning ratio ξ. Let x_s and x_a denote the displacements of the primary structure and the DVA, respectively. By using Lagrange's equations, we get the equations of motion

$$\begin{aligned} &m_s\ddot{x}_s + (c_s + c_a)\dot{x}_s - c_a\dot{x}_a + (k_s + k_a)x_s - k_a x_a = f_s(t) = F_0 \sin\Omega t, \\ &m_a\ddot{x}_a - c_a\dot{x}_s + c_a\dot{x}_a - k_a x_s + k_a x_a = 0. \end{aligned} \tag{1}$$

2.1. Frequency response of the damped primary system

We find the solution of Eq. (1) in the form

$$f_s(t) = F_0 e^{i\Omega t},\ x_s(t) = \tilde{u}_s e^{i\Omega t},\ x_a(t) = \tilde{u}_a e^{i\Omega t}\quad . \tag{2}$$

Substitution of Eq. (2) into Eq. (1) yields

$$\begin{cases} \left[k_s + k_a - m_s\Omega^2 + i(c_s + c_a)\Omega\right]\tilde{u}_s - (k_a + ic_a\Omega)\tilde{u}_a = F_0, \\ -(k_a + ic_a\Omega)\tilde{u}_s + (k_a - m_a\Omega^2 + ic_a\Omega)\tilde{u}_a = 0. \end{cases} \tag{3}$$

Eq. (3) denotes a set of linear algebraic equations with two unknowns $\tilde{u}_s$ and $\tilde{u}_a$. It follows that

$$\tilde{u}_s = \frac{\Delta_s}{\Delta} = \frac{F_0(k_a - m_a\Omega^2 + ic_a\Omega)}{(k_s - m_s\Omega^2)(k_a - m_a\Omega^2) - (k_a m_a + c_s c_a)\Omega^2 + i\Omega\left[c_s k_a + c_a k_s - (c_s + c_a)m_a\Omega^2 - c_a m_s\Omega^2\right]},$$

$$\tilde{u}_a = \frac{\Delta_a}{\Delta} = \frac{F_0(k_a + ic_a\Omega)}{(k_s - m_s\Omega^2)(k_a - m_a\Omega^2) - (k_a m_a + c_s c_a)\Omega^2 + i\Omega\left[c_s k_a + c_a k_s - (c_s + c_a)m_a\Omega^2 - c_a m_s\Omega^2\right]},$$

(4)

and

$$H = u_s = |\tilde{u}_s| = \frac{F_0\sqrt{(k_a - m_a\Omega^2)^2 + c_a^2\Omega^2}}{\sqrt{\left[(k_s - m_s\Omega^2)(k_a - m_a\Omega^2) - (k_a m_a + c_s c_a)\Omega^2\right]^2 + \Omega^2\left[c_s k_a + c_a k_s - (c_s + c_a)m_a\Omega^2 - c_a m_s\Omega^2\right]^2}}.$$

(5)

The formula (5) will be chosen as the target function of the Taguchi's optimization problem.

2.2. The matrix form of the differential equations for the motion of primary system and dynamic vibration absorber

Equation (1) can then be written in the compact matrix form as

$$\mathbf{M\ddot{x} + D\dot{x} + Kx = f(t)}, \tag{6}$$

where

$$\begin{aligned} &\mathbf{x} = \begin{bmatrix} x_s \\ x_a \end{bmatrix};\ \mathbf{M} = \begin{bmatrix} m_s & 0 \\ 0 & m_a \end{bmatrix};\ \mathbf{D} = \begin{bmatrix} c_s + c_a & -c_a \\ -c_a & c_a \end{bmatrix}, \\ &\mathbf{K} = \begin{bmatrix} k_s + k_a & -k_a \\ -k_a & k_a \end{bmatrix};\ \mathbf{f} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix}\sin(\Omega t). \end{aligned} \tag{7}$$

Eq. (6) can also be written in the following matrix form as follows

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{D}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{a}\sin\Omega t + \mathbf{b}\cos\Omega t. \tag{8}$$

The particular solution of Eq.(9) can be found in the form

$$\mathbf{x} = \mathbf{u}\sin\Omega t + \mathbf{v}\cos\Omega t\ . \tag{9}$$

The derivative of vector $\mathbf{x}$ by time one obtain

$$\dot{\mathbf{x}} = \Omega(\mathbf{u}\cos\Omega t - \mathbf{v}\sin\Omega t)\ ,$$
$$\ddot{\mathbf{x}} = -\Omega^2(\mathbf{u}\sin\Omega t + \mathbf{v}\cos\Omega t)\ .$$

Substituting the terms of $\mathbf{x}, \dot{\mathbf{x}}, \ddot{\mathbf{x}}$ into Eq.(8), then comparing the coefficients of $\sin\Omega t$ and $\cos\Omega t$, we obtain the system of linear algebraic equations to determine the vectors $\mathbf{u}$ and $\mathbf{v}$

$$\begin{bmatrix} \mathbf{K} - \Omega^2\mathbf{M} & -\Omega\mathbf{D} \\ \Omega\mathbf{D} & \mathbf{K} - \Omega^2\mathbf{M} \end{bmatrix} \begin{bmatrix} \mathbf{u} \\ \mathbf{v} \end{bmatrix} = \begin{bmatrix} \mathbf{a} \\ \mathbf{b} \end{bmatrix}. \tag{10}$$

If the determinant of the coefficient matrix in Eq.(10) is not zero, then the vectors $\mathbf{u}$ and $\mathbf{v}$ are uniquely determined. The solution of Eq.(8) is given by

$$\mathbf{x} = \mathbf{u}\sin\Omega t + \mathbf{v}\cos\Omega t, \tag{11}$$

where $\mathbf{u}$ and $\mathbf{v}$ are determined from Eq.(10).

3. APPLICATION OF TAGUCHI'S METHOD TO PARAMETER DESIGN OF DVA

3.1. Background of a procedure

Taguchi developed his methods in the 1950s and 1960s. Taguchi's methods provide a means to determine the optimum values of the characteristics of a product or process such that the product is robust (insensitive to sources of variation) while requiring less experiments than is required by traditional methods. The mathematical basis of the Taguchi method is mathematical methods of statistics. The Taguchi method allows to determine the optimal condition of many parameters of the research object. This method is applied to solve the multi-objective optimization problem in mechanical engineering, civil engineering, and transportation engineering. In this paper, Taguchi's method is applied to optimize the parameters of DVA to reduce the vibration amplitude of primary system. By using the Taguchi method, we must note the following two important points. The first is that it needs to determine the quality characteristics of the problem. The second option is that we need to select the orthogonal arrays. The Taguchi's methods begin with the definition of the word quality. Taguchi employs a revolutionary definition: "Quality is the loss imparted to society from the time a product is shipped" [5]. In this paper the quality characteristics are also called the signal-to-noise ratio (SNR). It is defined for a nominal-the-best procedure as [2]

$$\eta = SNR = -10\log_{10}(H_{actual} - H_{\min})^2\ , \tag{12}$$

where H_{actual} is the target function in experiment j, and $H_{\min}$ is desired value of target function. Taguchi developed the orthogonal array method to study the systems in a convenient and rapid way, whose performance is affected by different factors when the considered system becomes more complicated with increasing number of influence factors [1-4].

3.2. Determine optimal parameters of a DVA at the resonant frequency

Now Taguchi's method is applied to optimize the parameters of a DVA to reduce the vibration amplitude of primary system. The parameters of primary system are listed in Table 1.

Table 1. Parameters of primary system.

Parameter	Variable	Value	Unit
mass	m_s	250	kg
damping coefficient	c_s	200	Ns/m
spring coefficient	k_s	$1.5\text{x}10^6$	N/m
amplitude of ext. force	F_0	250	N
frequency of ext. force	Ω	77.46	rad/s

Step 1: Selection of control factors and target function

Because the mass ratio of the DVA to the primary structure is usually few percent, the principal parameters of the DVA are its tuning ratio (i.e. ratio of DVA's frequency to the natural frequency of primary structure) and damping ratio. The mass of the DVA firstly selected as $m_{a=}$12.5 kg. The control factors are chosen as follows

$$\mathbf{x} = x_1\ x_2\ x_3\ ^T = m_a\, c_a\, k_a\ ^T\ . \tag{13}$$

The target function H is chosen according to the formula (5). Damping and spring coefficients of the DVA , c_a and k_a, are control factors. Three levels of each control factor are given in Table 2.

Table 2. Control factors and levels of control factors.

Levels	Control factors		
	m_a [kg]	c_a [Ns/m]	k_a [N/m]
1	2	80	$1.0\text{x}10^5$
2	4	100	$2.0\text{x}10^5$
3	8	120	$3.0\text{x}10^5$

Step 2*: *Selection of orthogonal array and calculation of signal-to-noise ratio (SNR)

Three levels of each control factor are applied, necessitating the use of an L9 orthogonal array [1, 4]. Coding stage 1, stage 2, stage 3 of the control parameters are the symbols 1, 2, 3. By performing the experiments and then calculating the corresponding response results, we have the values of actual target function H as shown in the Table 3, in which a minimal target value of $H_{min} = 0$ is selected.

The experimental results are then analyzed by means of the mean square deviation of the target function for each control parameter, namely the calculation of the SNR of the control factors according to the formula

$$\eta_j = (SNR)_j = -10\log_{10}(H_j - H_{\min})^2,\ j = 1,...,9\ , \qquad (14)$$

where H_j is the actual target function in experiment j, and $H_{\min}$ is desired value of target function.

Step 3*: *Analysis of signal-to-noise ratio (SNR)

Table 3. Experimental design using L9 orthogonal array.

Trial	Factor			Result	
	m_a	c_a	k_a	H	SNR
1	1	1	1	0.0120609292	38.3723846061
2	1	2	2	0.0124341452	38.1076813202
3	1	3	3	0.0125461249	38.0298078553
4	2	1	2	0.0079540421	41.9882422795
5	2	2	3	0.0082290805	41.6929738354
6	2	3	1	0.0070425160	43.0454431023
7	3	1	3	0.0042178069	47.4982661359
8	3	2	1	0.0026607102	51.5000484940
9	3	3	2	0.0038334828	48.3281297317

From Table 3 we can calculate the mean value of the SNR of the control parameter of $m_a = x_1$ corresponding to the levels 1,2,3

$$SNR(x_1^1) = (SNR(1) + SNR(2) + SNR(3))/3 = 38.1699579272$$
$$SNR(x_1^2) = (SNR(4) + SNR(5) + SNR(6))/3 = 42.2422197391$$
$$SNR(x_1^3) = (SNR(7) + SNR(8) + SNR(9))/3 = 49.1088147872$$

In which $SNR(x_1^1),\ SNR(x_1^2), SRN(x_1^3)$ are the mean square deviation of the control parameter m_a at the levels 1,2,3, respectively. Similarly we calculate the mean square deviation of the SNR for the levels 1,2,3 of the control parameter $c_a = x_2,\ k_a = x_3$

$$SNR(x_2^1) = (SNR(1) + SNR(4) + SNR(7))/3 = 42.6196310072$$
$$SNR(x_2^2) = (SNR(2) + SNR(5) + SNR(8))/3 = 43.7669012165$$
$$SNR(x_2^3) = (SNR(3) + SNR(6) + SNR(9))/3 = 43.1344602298$$

$$SNR(x_3^1) = (SNR(1) + SNR(6) + SNR(8))/3 = 44.3059587341$$
$$SNR(x_3^2) = (SNR(2) + SNR(4) + SNR(9))/3 = 42.8080177771$$
$$SNR(x_3^3) = (SNR(3) + SNR(5) + SNR(7))/3 = 42.4070159422$$

Then we draw the SNR Ratio Plot for optimization of seat displacement as shown in Figure 2.

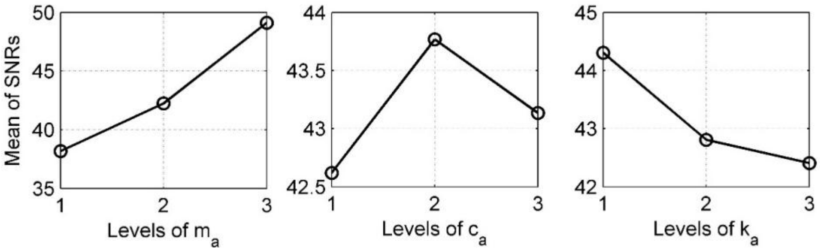


Figure 2. SNR Ratio plot for optimization of seat displacement of $m_a = x_1$, $c_a = x_2$ and $k_a = x_3$.

From Figure 2 we derive the optimal signal-to-noise ratio of the control parameters as follows

$$(SNR)x_1 = 49.1088147872,\ (SNR)x_2 = 43.7669012165,\ (SNR)x_3 = 44.3059587341 \quad (15)$$

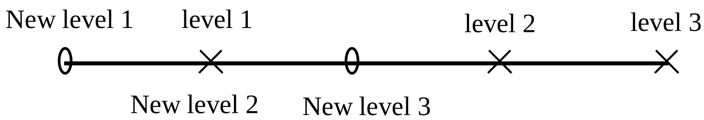
Step 4: Selection of new levels for control factors

By the formula (15), we see that the optimal signal-to-noise ratio of the control parameters is different. This makes it easy to perform iterative calculation. First we must select new levels for control parameters. Based on the level distribution diagram of the parameter (Figure 2), we choose the new levels of control parameters as follows. The optimal parameters are levels with the largest value of the parameters, namely: m_a level 3, c_a level 2, k_a level 1. Therefore, we have the values of the new levels as follows:

- Level 2 of the control parameter m_a = 8 Ns / m (level 3 of the previous parameter set),
- Level 2 of the control parameter c_a = 100 Ns / m (level 2 of the previous parameter set),
- Level 2 of the control parameter $k_a = 1.0 \times 10^5\ N/m$ (level 1 of the previous parameter set).

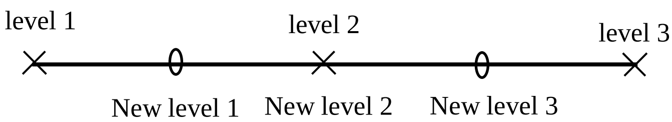
We use these values as central values of the next search, (these values are levels 2 for the next search). The levels of the control parameters of the following search are created according to the following rule:

If level 1 is optimal then the next levels are



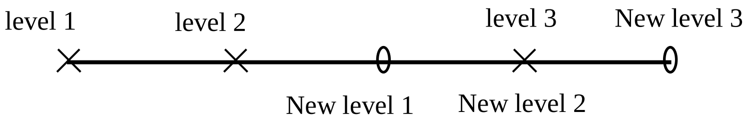
$$\begin{cases} level2\_new = level1\_old \\ level1\_new = level1\_old - \dfrac{level2\_old - level1\_old}{2} \\ level3\_new = level1\_old + \dfrac{level2\_old - level1\_old}{2} \end{cases}$$

If level 2 is optimal then the next levels are



$$\begin{cases} level2\_\text{new} = level2\_old \\ level1\_new = level2\_old - \dfrac{level2\_old - level1\_old}{2} \\ level3\_new = level2\_old + \dfrac{level3\_old - level2\_old}{2} \end{cases}$$

If level 3 is optimal then the next levels are



$$\begin{cases} level2\_\text{new} = level3\_old \\ level1\_new = level3\_old - \dfrac{level3\_old - level2\_old}{2} \\ level3\_new = level3\_old + \dfrac{level3\_old - level2\_old}{2} \end{cases}$$

According to the above rule, we have the new levels of control parameters as shown in Table 4.

Table 4. Control factors and new levels of control factors.

Levels	Control factors		
	m_a [kg]	c_a [Ns/m]	k_a [N/m]
1	6	90	50000
2	8	100	100000
3	10	110	150000

Then the analysis of signal-to-noise ratio (SNR) is performed as the step 2.

Step 5: Check the convergence condition of the signal-to-noise ratio and determine the optimal parameters of the DVA

Table 5. Noise values of the control parameter $(SNR)_i$ of the control parameters.

Trial	Optimal noise values $(SNR)_i$		
	$(SNR)x_1$	$(SNR)x_2$	$(SNR)x_3$
1	49.1088147872	43.7669012165	44.3059587341
2	55.9272740956	54.3189064302	58.3951488545
3	56.1664058168	57.2013745970	60.8209778901
4	64.2795638094	64.2062018597	65.5283802774
…	…	…	…
38	68.1299451384	68.1299451384	68.1299451384
39	68.1299451384	68.1299451384	68.1299451384
40	68.1299451385	68.1299451384	68.1299451384
41	68.1299451385	68.1299451385	68.1299451385

After 33 iterations, we obtain the optimal noise values of the control parameters $x_1 = m_a,\ x_2 = c_a,\ x_3 = k_a$. The calculation results are recorded in Table 5.

If the optimal SNR of the control parameters is equal (or approximately equal) we move on to step 5. If otherwise we return to step 2. According to the above analysis, after 41 iterations, we obtain the optimal values of the dynamic vibration absorber:

$$m_a = 11.5kg,\quad c_a = 100Ns/m,\ k_a = 6.9819\times10^4\ N/m \tag{16}$$

Step 6: Determining the vibration of the primary system

Knowing the parameters of the damper, using equation (11) we can easily calculate the vibration of the main system and of the dynamic vibration absorber. Using the optimum parameters (16), we plot the compliance curve in frequency domain for the damped primary system in Fig.3. Numerically we can find the peak values H(A) and H(B) of the normalized amplitude and their corresponding frequency ratios $f_A / f_S = 0.901,\ f_B / f_S = 1.116$.

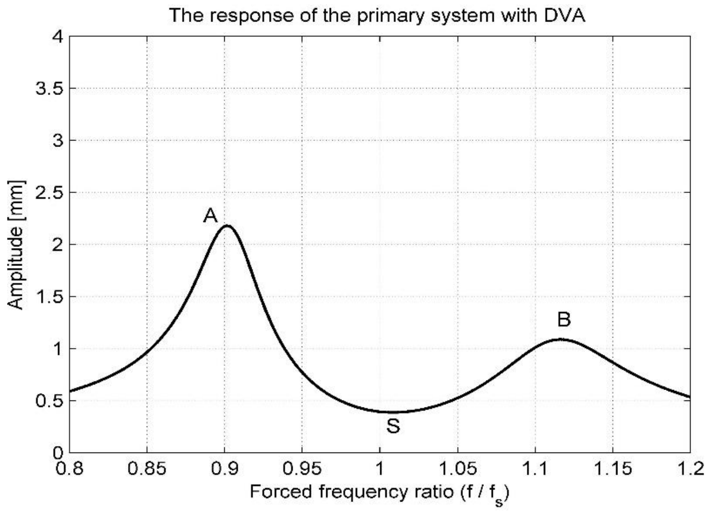


Figure 3. The compliance curve in frequency domain.

3.3. Problem formulation for determining optimal parameters of a DVA in a frequency domain

When a primary system is damped, the “fixed-points” feature no longer exists. However, as shown in [13], when there is viscous damping on both masses, the design problem can be formulated as follows: Given a primary mass m_s, connected to the ground with a spring-dashpot element and subjected to the force $F_O \sin\Omega t$, compute the values of secondary mass m_a, stiffness k_a and viscous damping c_a such that the frequency response curve of the primary mass has two maximum amplitudes. Therefore, it is justified to assume that the “fixed-point” theory also approximately holds even for the case when a damped DVA is attached to a lightly or moderately damped primary system. Based on this assumption, it is reasonable to assume that $H(\Omega)$ has two distinct resonance points. These are denoted A and B, with frequencies Ω_A and Ω_B $(\Omega_A < \Omega_B)$. This leads to the equations

$$H(\Omega_A) = \max|H(\Omega)| \text{ and } H(\Omega_B) = \max|H(\Omega)| \quad (17)$$

It is well recognized that each fixed point very close to the corresponding resonance point, and that the trade off relation between $H(\Omega_A) = \max|H(\Omega)|$ and $H(\Omega_B) = \max|H(\Omega)|$ can be postulated. On this assumption, it is guaranteed that the optimum design is derived using equivalent resonance magnification factors

$$\max|H(\Omega)| = |H(\Omega_A)| = |H(\Omega_B)| \quad (18)$$

The problem can also be formulated as the one that minimizes the following two functions [16]

$$f_1 = \frac{1}{2}\left|H\ \ \Omega_A\ -H\ \ \Omega_B\ \right|, \qquad f_2 = \frac{1}{2}\left|H\ \ \Omega_s\ +H\ \ \Omega_B\ \right| \quad (19)$$

A target function can be defined as

$$f = \mathrm{w}_1 f_1 + \mathrm{w}_2 f_2 \ \to \min \ , \quad (20)$$

where w_1 and w_2 are weighting factors used to impose different emphasis on each of the target functions. The optimum solution can be found by using the Taguchi's method. The calculation results of the equation (20) are given in Fig. 4 and in Table 6.

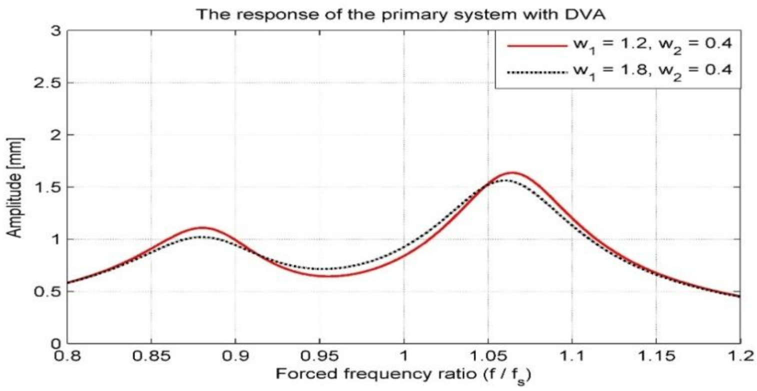


Figure 4. The compliance curves in frequency domain.

Table 6. Calculated results with different weighting factors

Weighting factors		Vibration amplitude of the primary system without DVA,[mm]	Vibration amplitude of the primary system with DVA,[mm]	Ratio %
w_1	w_2			
0.8	0.4	16.14	1.582	90.2
1.2	0.4	16.14	1.636	89.86
1.8	0.4	16.14	1.563	90.32
1.8	0.3	16.14	1.876	88.38

4. VERIFYING THE EFFECTIVENESS OF THE PROPOSED APPROACH

In this section, the optimum values of the DVA determined by the Taguchi method would be compared with the values calculated from the other methods. In [17, 18], Anh and Nguyen recently adopted the dual equivalent linearization technique for handling the variant DVA model, which transforms approximately the damped primary system to an equivalent undamped system as shown in Figure 5.

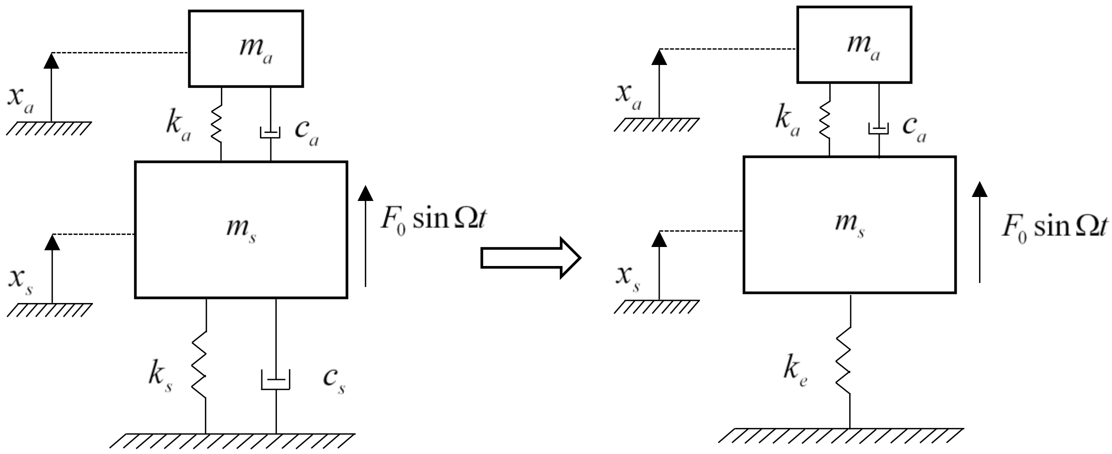


Figure 5. The approximation of the primary system [18].

According to Anh and Nguyen [17, 18], two design parameters of dynamic vibration absorber are determined by the following formulas

$$k_a = \frac{m_a \omega_s^2}{\left(1+m_r\right)^2 \left[\sqrt{1+\frac{\pi^2}{\left(\pi^2-2\right)^2}\zeta_s^2}+\frac{\pi}{\pi^2-2}\zeta_s\right]^2}, \quad c_a = \frac{2 m_a \omega_s}{\sqrt{1+\frac{\pi^2}{\left(\pi^2-2\right)^2}\zeta_s^2+\frac{\pi}{\pi^2-2}\zeta_s}}\sqrt{\frac{3m_r}{8\left(1+m_r\right)^3}}. \tag{20}$$

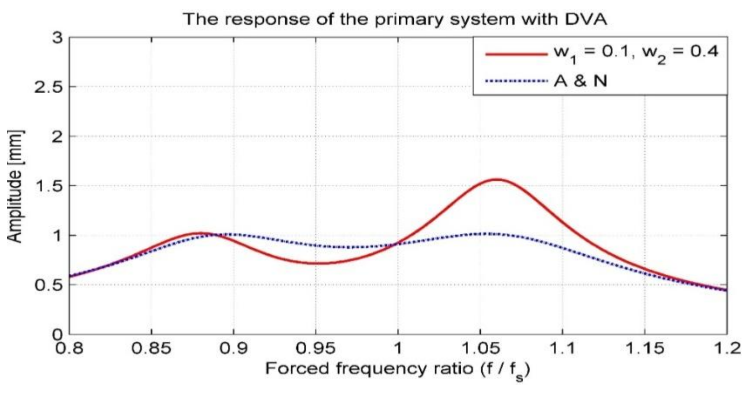


Figure 6. The compliance curves in frequency domain with $c_s = 200 Ns/m$.

The calculation results are given in Figs. 6 and 7. Fig. 6 shows the compliance curves in frequency domain. Figure 7 shows the response of the damped primary system at the resonance frequency. Figs 6 and 7 show that the compliance curves in frequency domain from the proposed in this paper are closer to the curves from the expression (20) given by Anh and Nguyen.

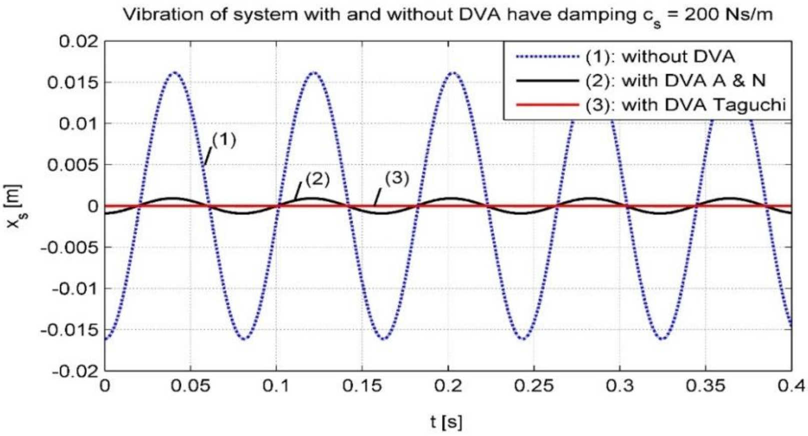


Figure 7. The response of the primary system with DVA and without DVA with $c_s = 200 Ns/m$.

5. CONCLUSIONS

When a damped primary system is excited by a harmonic force, its vibration can be suppressed by attaching a DVA. The DVA has the effect of reducing vibrations in the resonance region, and has almost ineffective far out of the resonance region. In this paper, a procedure for the optimal design of parameters of the DVA installed in damped primary system was investigated from the viewpoint of suppressing vibration amplitude in the damped primary system in the resonant region. Based on the obtained results, the following concluding remarks can be reached:

- The use of the Taguchi’s method to design the optimal parameters of the DVA installed in damped primary system is relatively simple and convenient.

- The Taguchi’s method has the good effect of reducing vibration in a narrow band of the resonant frequency (the ratio is approximately 90 %).

- In the narrow band of the resonant frequency, the vibration reduction effect of the Taguchi method is similar to that of Anh and Nguyen.

We note that Taguchi's method has the following advantages: It does not need to use the derivative of the target function to calculate the optimal parameters, allowing the determination of multiple control parameters to reduce vibration for complex structures. In addition, the control parameters can be selected the same or different. This problem is being studied at the Hanoi University of Science and Technology.

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