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An Investigation on Optimal Designing of Dynamic Vibration Absorbers Using Genetic Algorithm

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Abstract. In this study reducing of the unwanted phenomenon in machine tools, Vibration, has been investigated. The machine tools has been simulated as a five-degree-of-freedom discrete system consisting of cutting tool, work piece, body, head and cantilever of the machine. In order to reduce the undesired vibrations, a dynamic vibration absorber (DVA) with three unknown parameters (mass absorber $\boldsymbol{M_D}$, damping coefficient $\boldsymbol{C_D}$, stiffness absorber $\boldsymbol{K_D}$) has been added to the system in different positions. Utilizing Newton's second law, the governing equations have been obtained and solved. A genetic algorithm is proposed to efficiently achieve the optimum value for each 3-fold parameters in each positions of the system. The effectiveness of the proposed algorithm and the designed DVA is evaluated through comparing the vibration amplitude of the machine tool in the presence and absence of the DVA, and the paper concludes the best place to situate the vibration absorber and its specifications.

Keywords: Vibration Control, Dynamic Vibration Absorber, Optimize, Genetic Algorithm.

1. INTRODUCTION

The machine tools, the machines which are using to shape, cut, bore, grind, shear or other forms of deformation, in the other word the machines used to machine metal or other rigid materials have diverse applications in this industrial world. All machine tools have some means of constraining the work piece and provide guided movements of the parts. Thus the relative movement between the work piece and the cutting tool (which is called the tool path) is controlled or constrained by the machine to at least some extent, rather than being entirely "offhand" or "freehand".

As early as 1907, Frederick W. Taylor described machining vibrations as the most obscure and delicate of all the problems facing the machinist, an observation still true today, as shown in many publications on machining [1].

Mathematical models make it possible to simulate machining vibration quite accurately, but in practice it is always difficult to avoid vibrations and there are basic rules for the machinist:

- Rigidify the work piece, the tool and the machine as much as possible
- Choose the tool that will excite vibrations as little as possible (modifying angles, dimensions, surface treatment, etc.)
- Choose exciting frequencies that best limit the vibrations of the machining system (spindle speed, number of teeth and relative positions, etc.)

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Vibration problems generally result in noise, bad surface quality and sometimes tool breakage. The main sources are of two types: forced vibrations and self-generated vibrations.

- Forced vibrations are mainly generated by interrupted cutting (inherent to milling), run out, or vibrations from outside the machine.

Self-generated vibrations are related to the fact that the actual chip thickness depends also on the relative position between tool and work piece during the previous tooth passage. Thus increasing vibrations may appear up to levels which can seriously degrade the machined surface quality.

Undesirable vibration descents the performance of machineries. Nowadays machines are required to take smaller foot print, weight less, more durable while having higher accuracy and operation speed. One way to achieve some of these requirements is by reducing undesirable vibrations. A great deal of engineering efforts is employed to minimize undesirable vibration of machineries. Different solutions may be used such as changing electrical and mechanical stiffness but these kind of solutions are utilized at the design stage for new machines that are not built yet. However, making any fundamental changes to existing machines is usually difficult to implement, therefore, a method to improve the performance of an existing machine, with a given mechanical and electrical stiffness, is highly desirable [2].

One of the most popular solutions for machine tools vibration is using the vibration absorber. Vibration absorbers are devices attached to flexible structures in order to minimize the vibration amplitudes at a specified set of points. Frahm [3] in 1909 proposed a vibration absorber consists of a second mass-spring device attached to the main device, also modeled as a mass-spring system, which prevents it from vibrating at the frequency of the sinusoidal forcing acting on the main device [4].

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. This idea was pioneered by Watts [5] in 1883 and Frahm [3] in 1909. However, a DVA consisting of only a mass and spring has a narrow operation region and its performance deteriorates significantly when the exciting frequency varies. The performance robustness can be improved by using a damped DVA that consists of a mass, spring, and damper. The key design parameters of a damped DVA are its tuning parameter and damping ratio. The first mathematical theory on the damped DVA was presented by Ormondroyd and Den Hartog [6]. Since then, many efforts have been made to seek optimum parameters for the damped DVA. Den Hartog [4] first tackled the optimum solution of a damped DVA that is attached to a classical primary system, i.e., a system free of damping. His study utilized the feature of 'fixed-point' frequencies, i.e., frequencies at which the response amplitude of the primary mass is independent of the absorber damping. Based on the ''fixed-points'' theory, Den Hartog found the optimum tuning parameter and defined the optimality for the optimum absorber damping. Based on this optimality, Brock derived an analytical solution for the optimum damping ratio of the damped DVA [7].

Wang et al. in 1985 proved that the vibration amplitude of the main mass cannot be made zero at the forcing frequency by employing a damped absorber, but the sensitivity of the system to variations in the forcing frequency decreases. Furthermore with damped absorbers, the vibration amplitude of the absorber mass decreases considerably [8].

In order to design an optimal damper with minimum relative vibration between the cutting tools and work piece during stable machining, Shin and Wang employed an approximate normal mode method to calculate the response of a machine tools system with non-proportional damping subject to random excitation [9]. Their approach significantly lessened the amount of computation especially for higher-order systems when responses have to be calculated repeatedly in the process of optimization [9].

Huang and Chen [10] applied the dynamic absorber vibration concept in aircraft fuselages. They tried to obtain the optimum design of the absorber to minimize the vibration and the interior noise level of the fuselage.

P. Varpasuo [11] studied one mass dynamic vibration absorber. The damping characteristics of the absorber are assumed stochastic. Both viscous and internal hysteretic type damping is considered. The system is optimized to yield the minimum response in steady state for pre-selected frequency band when excitation frequency sweeps over the resonance point of the base system.

Duncan et al. [12] examined the effects of dynamic vibration absorber for high speed machinery. They investigated dynamic absorber effect in spindle-holder-tool assemblies and showed that the system dynamic stiffness could be improved and critical stability limit in machining will be increased, and therefore higher material removal rate in high speed machining achieved.

The application of DVAs is growing widely in machine tools. Ultrasonic machining (USM) is an effective technique specifically for hard to machine materials. Amer [13] investigated the coupling of two non-linear oscillators of the main system and absorber representing ultrasonic cutting process to control the system behavior at basic parametric resonance condition where the system damage is plausible. In continuation of previous research Kamel et al. [14] examined the system behavior at simultaneous primary and internal resonance condition.

Elhami and Heydari [15] depicted two methodology to reduce the vibrations of the head of the working machine tools over a frequency range, (a): by using dynamic vibration absorber, (b): by changing material body type of the machine. In the second method machine tools simulated in a system with three degree-of-freedom and its related equations were obtained parametrically.

Cheung and Wong [16] used the exact transfer function to find out the approximated optimal tuning of the absorber in multi degree of freedom systems including the optimization of displacement, velocity, acceleration and kinetic energy of the system.

Tootoonchi and Gholami [2] suggested another approach to modification to the tool holder. They employed a time delay resonator (DR) and without changing the existing control system or mechanical structure of the machine tried to minimize the undesirable vibration of machine tools with specific working frequency.

Brown and Singh [17] proposed a minimax problem formulation to determine the parameters of a passive vibration absorber in a condition that the frequency forcing is an uncertainty.

In the case of tuning a DVA parameters Zilletti et al. [18] illustrated that maximizing the absorbed power of the DVA has the same effect as minimizing the kinetic energy of the host structure.

Tigli [19] studied dynamic absorber vibration (DVA) or tuned mass damper (TDM) installed on a linear damped systems that are exposed to random loads. He considered three cases of minimizing the variance of the displacement, velocity and acceleration of the main mass in his optimization exploration; and compared his more accurate estimates obtained by his proposed design formula with published approximate expressions.

Jang et al. [20] suggested a similar procedure proposed by Hartog to determine the parameters of a 2dof tuned vibration absorber (TVA). And recruited a numerical example to compare the performance of their approach with the optimal 2dof TVA.

The behavior of a three degree of freedom system consisting of a linear primary structure (PS), nonlinear tuned mass damper (NTMD) and a semi-active tuned mass damper (STMD) connected in series is scrutinized by Eason et al. [21] and the results showed that adding a small STMD to a vibration absorber system with a small stiffness nonlinearity can increase the attenuation of the primary structure and may be an effective and cost-efficient alternative to replacing the original system.

In extension of previous studies to reduce the vibration of the machine tools during its working by using passive DVA, this research tried to optimize the system to produce minimum vibration. The machine tool simulated by a five-DOF, and all of the dynamic equations parametrically is derived and solved by Newton's second law. Due to the equations and technical characteristics of materials of the machine tool, vibration amplitude of each components can be obtained. In order to minimize the undesired vibration, a dynamic vibration absorber is recruited, and the DOF of the system increases from five to six DOF.

According to the installation place of the dynamic absorber in system, there are four different modes and all of them will be analyzed in continue.

In order to simulate the vibration absorber an arrangement of a mass, a stiffness, and a damper is employed. Dynamic equations of the system by using three unknown parameters (mass absorber M_D, damping coefficient C_D, stiffness absorber K_D) is derived and solved by Newton's second law. An evolutionary genetic algorithm is proposed to efficiently attain the optimum value for each 3-fold parameters in each positions of the machine tool. The efficiency of the offered algorithm is evaluated, and the best place to situate the vibration absorber and its specifications is determined.

2. MATERIALS AND SIMULATING

2.1 Machine tools simulating

In this study machine tools is considered to consist of five components: head, cutting tool, work piece, body and cantilever (fig. 1).

Harmonic force of the machine tools has been exerted to the work piece. In vibrations analysis, the machine tools components has been modeled with five intensive masses that these masses adjoined by elastic components and dampers (fig. 2.):

Figure 1. The attempted machine tool.

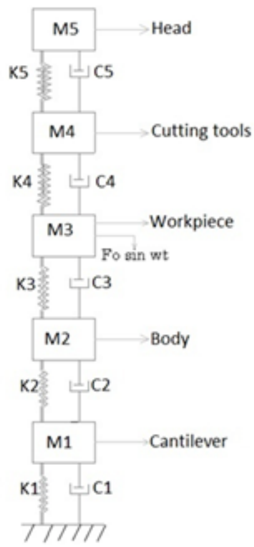

Figure 2. Model A: The model of the main system consisting five components.

As shown in fig. 2, linear coordinates has been used to describe machine tools intensive motion masses, Newton's second law has been employed for deriving motion equation.

Equation of motion that subjected to the force is as follows:

$$\left[M_{ij}\ddot{x}_j(t) + C_{ij}\dot{x}_j(t) + K_{ij}x_j(t)\right] = F_i(t) \quad , \; i,j = 1,2,\dots,n \tag{1}$$

Where $M_{ij}$is the mass of machine tools components in kg; C_{ij} is the viscous damping coefficient between the machine tools components in $N.s/m$; K_{ij} is the equivalent spring constant(stiffness coefficient) of the machine tools components in Nm^{-1}; x_j is the machine tools components displacements in its horizontal direction in m; F_0 is the cutting force in *N*; ω is the excitation frequency in $rads^{-1}$; t is the time in s and dot denotes derivative with respect to time. Technical specifications of machine tools material is as follows:

$Mass(kg)$	$M_1 = 100$	$M_2 = 80$	$M_3 = 8$	$M_4 = 5$	$M_5 = 25$
$Dampping = C\ \left(\frac{Nm}{s}\right)$	$c_1 = 500$	$c_2 = 400$	$c_3 = 300$	$c_4 = 200$	$c_5 = 250$
$Stiffness(\frac{N}{m})$	$k_1 = 100e^4$	$k_2 = 80e^4$	$k_3 = 30e^4$	$k_4 = 20e^4$	$k_5 = 90e^4$

2.2 Computation of vibration amplitude for machine tools component

Equation obtained by Newton's second law method from simulating mass-stiffness system of machine tools that represented at phase matrix is as follows:

$$\sum_{i=0}^{n} F_i = \sum_{i=0}^{n} M_i \frac{d^2x_i}{dt^2} \qquad i = 1,2,\dots,5 \tag{2}$$

$$\left[M_{ij}\ddot{x}_j(t) + C_{ij}\dot{x}_j(t) + K_{ij}x_j(t)\right] = F_i(t) \quad i,j = 1,2,\dots,5 \tag{3}$$

$$\left[M_{ij}\right]\left[\ddot{x}_j\right] + \left[C_{ij}\right]\left[\dot{x}_j\right] + \left[K_{ij}\right]\left[x_j\right] = \left[F_i(t)\right] \tag{4}$$

Where$[\boldsymbol{M}_{\text{ij}}]$ is the mass matrix, $[\boldsymbol{C}_{\text{ij}}]$ is the damping matrix, $[\boldsymbol{K}_{\text{ij}}]$ is the stiffness matrix and $\boldsymbol{F}_i(t)$ is the force matrix, the governing equation of motion for Model A can be derived as:

$$\left[M_{ij}\right] = \begin{bmatrix} M_1 & 0 & 0 & 0 & 0 \\ 0 & M_2 & 0 & 0 & 0 \\ 0 & 0 & M_3 & 0 & 0 \\ 0 & 0 & 0 & M_4 & 0 \\ 0 & 0 & 0 & 0 & M_5 \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \\ \ddot{x}_4 \\ \ddot{x}_5 \end{bmatrix} \tag{5}$$

$$\left[F_i(t)\right] = \begin{bmatrix} 0 & 0 & F_0 sin\,\omega t & 0 & 0 \end{bmatrix}^T \tag{6}$$

The solution of Eq. (3) is assumed to be harmonic as:

$$\left[C_{ij}\right] = \begin{bmatrix} c_1 + c_2 & -c_2 & 0 & 0 & 0 \\ -c_2 & c_2 + c_3 & -c_3 & 0 & 0 \\ 0 & -c_3 & c_3 + c_4 & -c_4 & 0 \\ 0 & 0 & -c_4 & c_4 + c_5 & -c_5 \\ 0 & 0 & 0 & -c_5 & c_5 \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \end{bmatrix} \tag{7}$$

$$\left[K_{ij}\right] = \begin{bmatrix} k_1 + k_2 & -k_2 & 0 & 0 & 0 \\ -k_2 & k_2 + k_3 & -k_3 & 0 & 0 \\ 0 & -k_3 & k_3 + k_4 & -k_4 & 0 \\ 0 & 0 & -k_4 & k_4 + k_5 & -k_5 \\ 0 & 0 & 0 & -k_5 & k_5 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \end{bmatrix} \tag{8}$$

$$x_i(t) = A_i \sin(\omega t - \varphi_i) \quad i = 1,2,\dots,5$$

$$x_i(t) = A_i\, e^{i\omega t}.e^{-i\varphi} = \bar{A}_i\, e^{i\omega t} \tag{9}$$

$$\dot{x}_i(t) = A_i\, i\omega\, e^{i\omega t} \quad , \quad \ddot{x}_i(t) = -\omega^2 A_i\, e^{i\omega t} \tag{10}$$

For harmonic solution Eq. (3) matrix methods are indispensable, so that considering $x_i(t) = A_i e^{(i\omega t - \varphi_i)}$, $i = 1,2,3,\dots,5$ is response subjective, A_i , $i = 1,2,\dots,5$ is amplitude of oscillation, φ is the phase angle and ω is the frequency of excitation, Eq. (3) is solvable:

$$[\boldsymbol{A}_{ij}] = \left[-\omega^2[\boldsymbol{M}_{ij}] + i\omega[\boldsymbol{C}_{ij}] + [\boldsymbol{K}_{ij}]\right]^{-1} [\boldsymbol{f}_{ij}] \qquad i,j = 1,2,\dots,5 \tag{11}$$

2.3 Vibration amplitude reduction by using vibration absorber

In this method a vibration absorber has been recruited in the system to reduce vibration in machine tools (figure 3). The Vibration absorber is simulated by a mass-stiffness system with six degree-of-freedom and the placement of the vibration absorber is variable in the system. Equations solved by instituting different value for absorber mass (M_D), damping coefficient of the absorber(C_D), and absorber stiffness (K_D) are as follows:
Fig. 3 demonstrates the case in which the vibration absorber attached to the head of the machine tools.

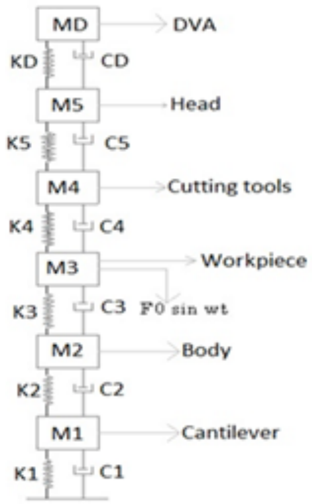

Figure 3. Model B: A vibration absorber is recruited to the main system

$$[M_{ij}\ddot{x}_j(t) + C_{ij}\dot{x}_j(t) + K_{ij}x_j(t)] = F_i(t)\ i,j = 1,2,\dots,6 \tag{12}$$

$$[M_{ij}][\ddot{x}_j] + [C_{ij}][\dot{x}_j] + [K_{ij}][x_j] = [F_i(t)] \tag{13}$$

Where$[\boldsymbol{M}_{ij}]$ is the mass matrix, $[\boldsymbol{C}_{ij}]$ is the damping matrix, $[\boldsymbol{K}_{ij}]$ is the stiffness matrix and $\boldsymbol{F}_i(t)$ is the force matrix, the governing equation of motion for Model B can be written as:

$$[M_{ij}] = \begin{bmatrix} M_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & M_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & M_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & M_4 & 0 & 0 \\ 0 & 0 & 0 & 0 & M_5 & 0 \\ 0 & 0 & 0 & 0 & 0 & M_D \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \\ \ddot{x}_4 \\ \ddot{x}_5 \\ \ddot{x}_6 \end{bmatrix} \tag{14}$$

$$[C_{ij}] = \begin{bmatrix} c_1+c_2 & -c_2 & 0 & 0 & 0 & 0 \\ -c_2 & c_2+c_3 & -c_3 & 0 & 0 & 0 \\ 0 & -c_3 & c_3+c_4 & -c_4 & 0 & 0 \\ 0 & 0 & -c_4 & c_4+c_5 & -c_5 & 0 \\ 0 & 0 & 0 & -c_5 & c_5+c_D & -c_D \\ 0 & 0 & 0 & 0 & -c_D & c_D \end{bmatrix} \begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \end{bmatrix} \tag{15}$$

$$[K_{ij}] = \begin{bmatrix} k_1+k_2 & -k_2 & 0 & 0 & 0 & 0 \\ -k_2 & k_2+k_3 & -k_3 & 0 & 0 & 0 \\ 0 & -k_3 & k_3+k_4 & -k_4 & 0 & 0 \\ 0 & 0 & -k_4 & k_4+k_5 & -k_5 & 0 \\ 0 & 0 & 0 & -k_5 & k_5+k_D & -k_D \\ 0 & 0 & 0 & 0 & -k_D & k_D \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \\ x_5 \\ x_6 \end{bmatrix} \tag{16}$$

$$[F_i(t)] = [0 \quad 0 \quad F_0 sin\ \omega t \quad 0 \quad 0 \quad 0 \quad]^T \tag{17}$$

The solution of Eq. (12) is assumed to be harmonic as:

$$x_i(t) = A_i \sin(\omega t - \varphi_i) \quad i = 1,2,\dots,6$$

$$x_i(t) = A_i\, e^{i\omega t}.\, e^{-i\varphi} = \bar{A}_\iota\, e^{i\omega t} \tag{18}$$

$$\dot{x}_i(t) = A_i\, i\omega\, e^{i\omega t} \qquad \ddot{x}_i(t) = -\omega^2 A_i\, e^{i\omega t} \tag{19}$$

For harmonic solution Eq. (12) matrix methods are indispensable, so that considering $x_i(t) = A_i e^{(i\omega t - \varphi_i)}$, $i = 1,2,3,\dots,6$ is response subjective, A_i , $i = 1,2,\dots,6$ is amplitude of oscillation, φ is the phase angle and ω is the frequency of excitation, Eq. (12) is solvable:

$$[\boldsymbol{A}_{ij}] = \left[-\omega^2[\boldsymbol{M}_{ij}] + i\omega[\boldsymbol{C}_{ij}] + [\boldsymbol{K}_{ij}]\right]^{-\mathbf{1}} [\boldsymbol{f}_{ij}] \qquad i,j = 1,2,\dots,6$$
(20)

In the following section, the solution procedure using a genetic algorithm (GA) to obtain the optimum parameters of the DVA, in different placement in different frequencies is explained.

3. GENETIC ALGORITHM
3.1 Approach

Genetic algorithms (GA) is a stochastic global search technique [22, 23] based on Darwin's evolution theorem of 'survival of fittest'; and due to the fact that they do not employ analytical properties of the function to be optimized, it is recruited in a wide range of optimization problems. In this procedure an initial population, a set of random feasible solutions, is used to find better solutions. Each individual of the population which is called chromosome is assessed according to a predetermined fitness function and based on some genetic operators (selection, crossover, mutation, etc.) and through some evolutionary iterated processes, better solutions is found.

The outline of the genetic algorithm practiced in this paper is condensed as the following steps:
1. According to the placement of the DVA (head, cutting tool, body and cantilever), an initial population of chromosomes is randomly produced in conformity with a predefined population size.
2. Each chromosome in the population is assessed through a predefined fitness function.
3. The roulette wheel technique is employed to pick out the parents and then the crossover and mutation operators are applied to generate new offspring from the picked out parents. According to the fitness function the newly generated offspring are evaluated.
4. The roulette wheel and elitism policy is applied to select and create the new generation among the parents and offspring.
5. The pre-specified stopping criterion for the algorithm is being checked. If the algorithm reaches to a predefined number of iterations, the search process terminates,
Otherwise the algorithm goes to Step 3. The flow chart of the GA procedure is shown in Fig. 4. [24].

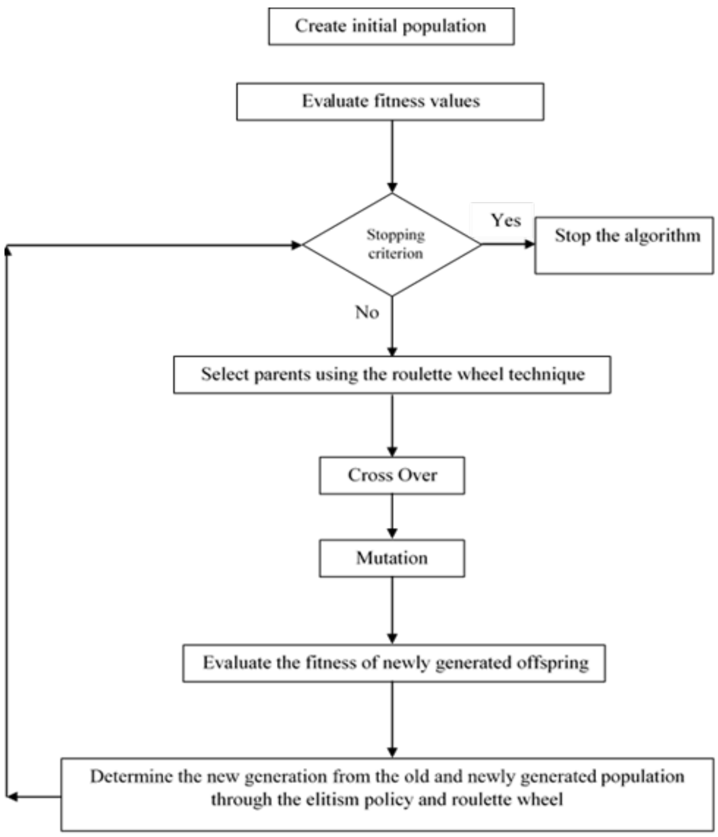

Figure 4. Flow chart of the proposed genetic algorithm [24]

3.2 Chromosome Representation

Genetic algorithm requires a string representation scheme (chromosome) to encode solutions of the problem. The chromosome designed in this paper is made up of four genes which considered to code mass, damping and stiffness coefficient of the absorber (M_D, C_D, K_D), in which each of the these genes is a matrix itself consisting machine tools components specifications. In other words, the first gene is the mass matrix of the machine tool components, the second gene is the damping coefficient matrix and the third gene is the stiffness coefficient matrix of the machine tool components. The fourth gene illustrates the fitness of the chromosome.

3.3 Selecting Operator

The selection process in which two chromosomes is selected from the current population to be the parent to give birth to new chromosomes (offspring) employed in this study is the roulette wheel selection mechanism in which the selection probability of each chromosome is proportional to its fitness value.

$$p_{selection}(i)=\frac{Ft(i)}{\sum_j Ft(j)} \tag{21}$$

Where $Ft(i)$ is the fitness of chromosome i.

It is worth mentioning that the fitness function employed in this study is a maximization, meaning that the algorithm tries to maximize the difference between the total vibration of the system with a DVA and the total vibration of the original system employing some importance coefficients.

3.4 Crossover Operator

The cross over operator, the mechanism which is employed generates offspring from parents, used in this study is the well-known single point crossover operator.

3.5 Mutation Operator

The mutation operator is used to empower the diversification of the solution searching area, and brings unexpected changes to the content of the chromosomes. In the mutation operator used in the genetic algorithm proposed in this paper, one gene is selected randomly and the content of the gene is produced randomly again due to the placement position of the DVA.

3.6 Reproduction

After producing offspring and evaluating them through fitness function, the new population is selected and produced among old and newly generated chromosomes (parents and offspring). Elitism policy and roulette wheel technique is used in the proposed algorithm. A pre-defined percent of the most excellent chromosomes are selected based on elitism policy and the rest of population is produced based on roulette wheel technique.

3.7 Stopping Criteria

In order to terminate the computation process of the proposed algorithm a stopping criterion is required. The criterion used here is a pre-defined maximum number of iterations.

4. RESULT

In this section first, vibration amplitude diagram of components of the original system (machine tool model A) are depicted, and second, a DVA is designed through a genetic

algorithm and an analytical comparison of vibration amplitude of components of the system in the presence and absence of the DVA is implemented.
The vibration amplitude diagram of the original system with the specification announced in the section 1.2. is as follows (fig. 5):

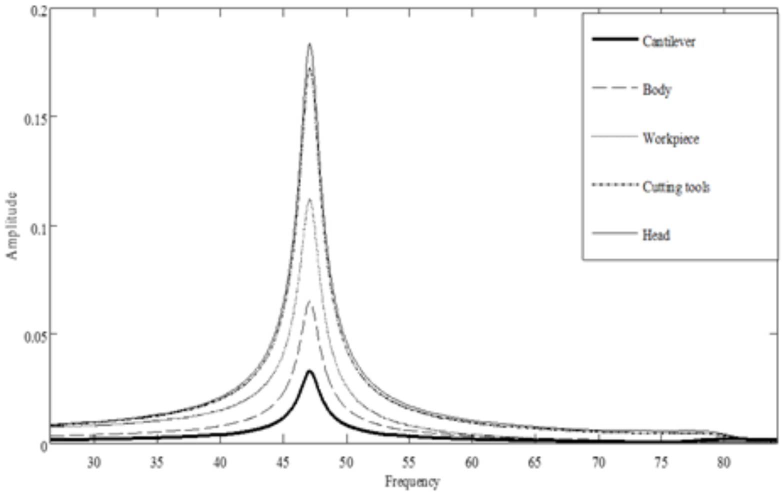

Figure 5. Vibration amplitude diagram of each components of the machine tools (model A)

As it can be inferred from the diagram, drastic and vicious vibration occurs at the head and cutting tool of the machine tool. Furthermore this vibration comes about at the 45-50 frequencies in all states, therefore the solution provided must straighten out this problem.

In this study in order to resolve this vicious vibration, a DVA is recruited, but the question is a DVA with what specifications (M_D, C_D, K_D) minimizes this vicious and drastic undesired vibration? And where it should be situated?

So as to answer the question, a genetic algorithm is designed to find the optimum specification of the DVA. The GA is employed in different positions of the machine tool components (head, cutting tool, body and cantilever), and the question is answered in the following manner (fig. 6, table 1):

Table 1. The Specification derived for the DVA in different positions of the machine tool in the range of 0 to 60 frequency.

Specification / Position	M_D	C_D	K_D	Fitness Function[a]
Head	5.2	49.8	10825.2	268
Cutting tool	9.9	49.9	5685.5	419
Body	0.0666	49.98	15067477.3	299
Cantilever	0.153	49.94	80771269.9	309

a. The difference between the total vibration of the system with a DVA and the total vibration of the original system employing some importance factors

Attaining the specification of the DVA in different position, the vibration amplitude of components of the system in the presence and absence of the DVA is compared in the following. Figure 7, 8, 9, 10 shows this comparison in different positions of head, cutting tool, body and cantilever.

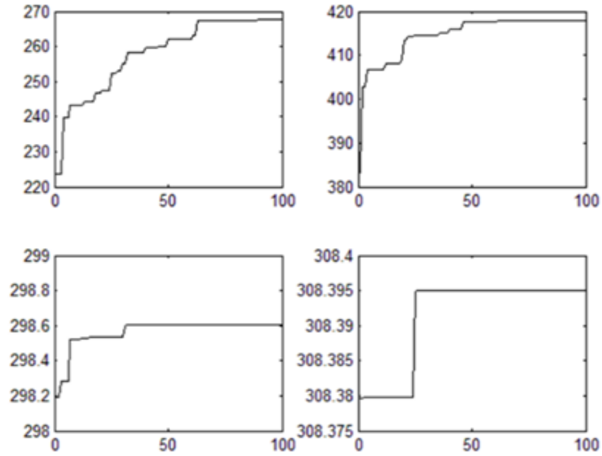

Figure 6. The progress of the fitness function of the proposed GA by recruiting different DVAs in different positions (head, cutting tool, body and cantilever).

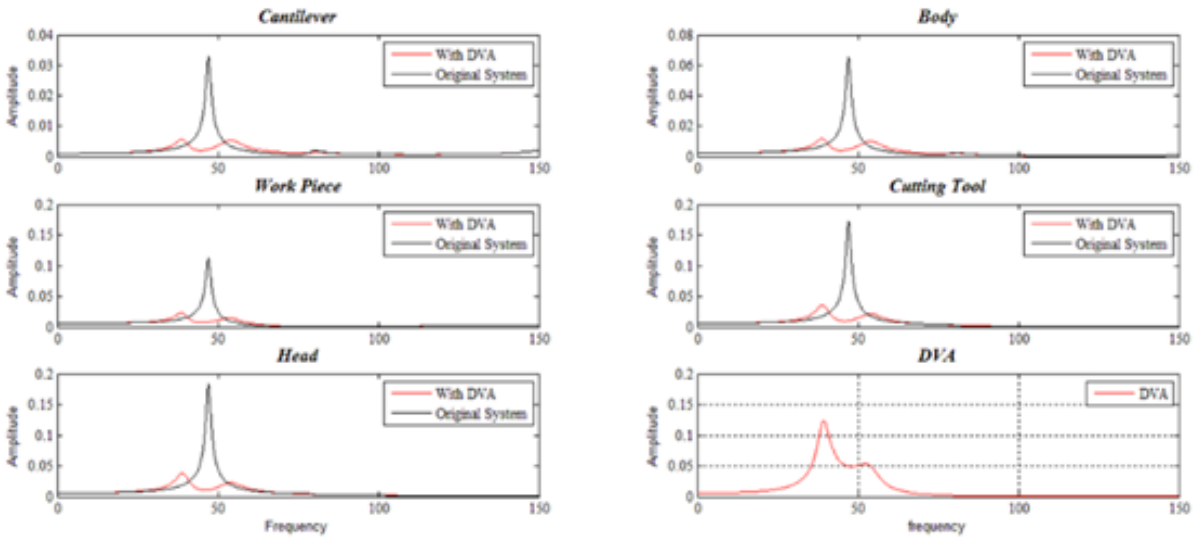

Figure 7. Comparing the vibration amplitude of components of the system in the presence and absence of the DVA in the case that the DVA is situated on the head of the system

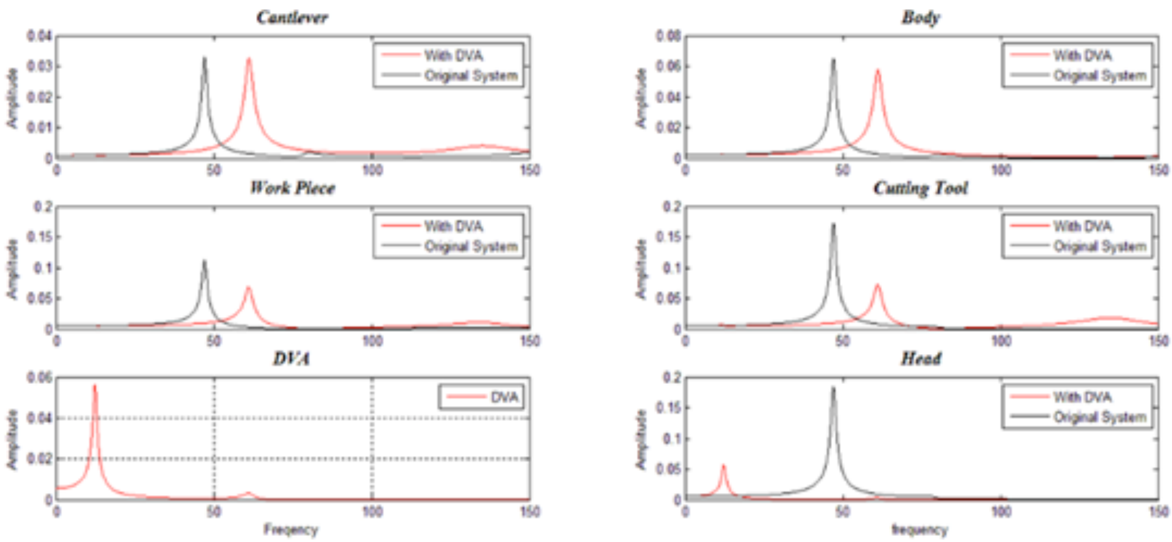

Figure 8. Comparing the vibration amplitude of components of the system in the presence and absence of the DVA in the case that the DVA is situated on the Cutting tool of the system

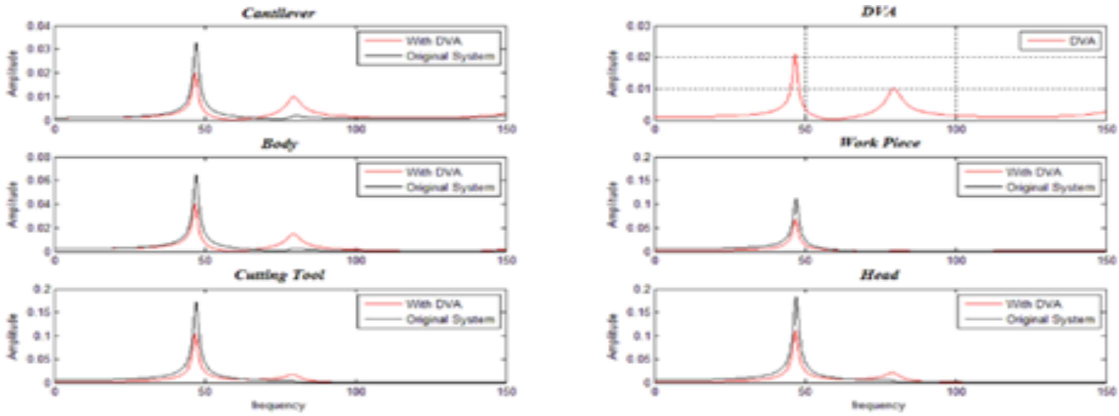

Figure 9. Comparing the vibration amplitude of components of the system in the presence and absence of the DVA in the case that the DVA is situated on the Body of the system

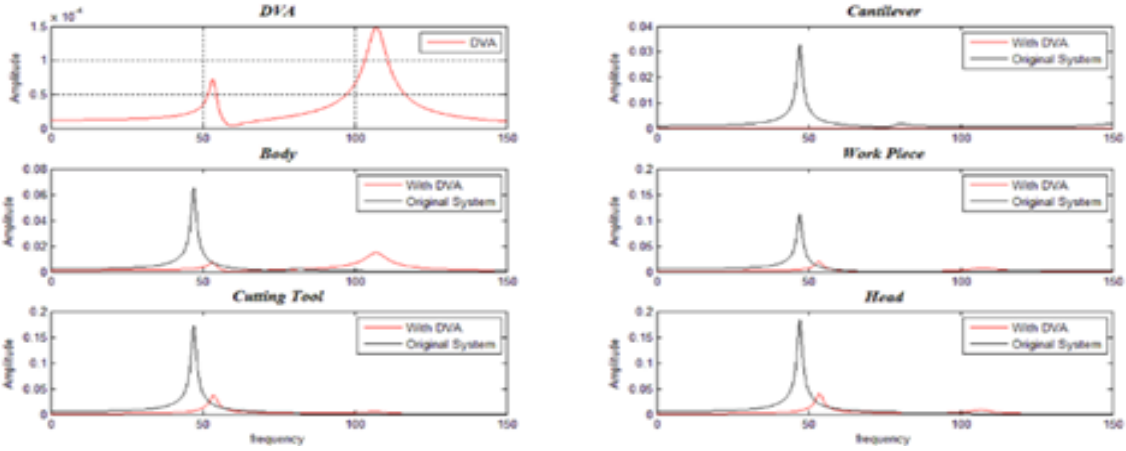

Figure 10. Comparing the vibration amplitude of components of the system in the presence and absence of the DVA in the case that the DVA is situated on the Cantilever of the system

Due to the outcomes achieved in table 1, according to the importance coefficient considered in the fitness function, the best place to situate the designed DVA to decrease the unwanted viscous vibration is on the cutting tool of the machine tool. In other words by employing the DVA on the cutting tool, maximum reduction in vibration amplitude in machine tools components obtains.

5. CONCLUSION

In this research a vibration absorber with three unknown parameters (mass absorber M_D, damping coefficient C_D, stiffness absorber K_D) was studied to reduce the vibration amplitude of a machine tool. The absorber was placed in different situations of the machine tool and the governing equation is solved in each case. Then in order to obtain the optimum specifications of the DVA, an evolutionary genetic algorithm was designed and the optimum specifications of the DVA is determined in different positions of the machine tool. The effectiveness of the proposed algorithm and the designed DVA is evaluated through comparing the vibration amplitude of the machine tool in the presence and absence of the DVA; And according to the assumptions and circumstances considered in this problem, the paper concludes that the best place to situate the vibration absorber is on the cutting tool of the machine tool.

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