GUJARAT TECHNOLOGICAL UNIVERSITY,
Ahmedabad, Gujarat, India

A

S Y N O P S I S

On

DESIGN AND IMPLEMENTATION OF PASSIVE LINEAR DYNAMIC VIBRATION ABSORBER FOR VIBRATION CONTROL OF ROTATING MASS UNBALANCE SYSTEM

Proposed to be submitted in partial fulfillment of the award of

Doctor of Philosophy

in

Mechanical Engineering

By

SHETH ANANT JITENDRAKUMAR
Enrollment No.: 179999919056

Under the Supervision of

Dr. Utkarsh.A.Patel
Assistant Professor in Mechanical Engineering
L.D College of Engineering, Ahmedabad, Gujarat, India.

August 2022

Table of Content		
Sr. No.	**Content**	**Page No.**

1. Title of the thesis and abstract:

Title: DESIGN AND IMPLEMENTATION OF PASSIVE LINEAR DYNAMIC VIBRATION ABSORBER FOR VIBRATION CONTROL OF ROTATING MASS UNBALANCE SYSTEM

Abstract: Dynamics of any system does play its crucial role whenever the working or operating conditions of the system are changing. Though utmost care is being taken at the design stage, the system may exhibit vibrations in its operating condition. Thus to attenuate vibration problem of a system in its working/running condition, Passive Dynamic Vibration Absorber (DVA) are the best possible solutions. The exact solution of damping factor as a function of mass ratio has been determined by using l'Hospital rule in its higher orders. The effect of exact solution have been compared numerically with approximate solutions obtained by using H∞ method, Equivalent Linearization method, solution proposed by Ioi & Ikeda and also system without DVA. Compared to all the listed methods, the proposed exact DVA damping factor required is far away from the approximate results. For a mass ratio range 0.05 to 0.15 with the exact solution, we appreciated 75% to 85% percentage reduction in the required damping factor compared to the approximate solution. The effect of tuned DVA with exact damping factor was experimentally tested on industrial blower functioning at two different speeds. In 1st case the DVA was designed for an industrial air blower with rotating mass unbalance and operating speed 2550 RPM. To design a DVA, damping material was predetermined with known damping factor and accordingly mass ratio was determined from the exact solution. From the calculated mass ratio, the DVA mass was calculated. The proposed DVA was a hollow ring with side edge to support rubber strips and 6800-2RS bearing. Using dual channel FFT analyzer the vibration signals were recorded in horizontal, vertical and axial directions. The system with DVA was found in "good" condition according to ISO 10816-3. In the 2nd case the same DVA was made an integral part of the blower system. DVA was attached to the casing. Vibration reduction was much more convincing compared to the 1st case arrangement. In the 3rd case air blower operating at 720 RPM with unbalance was considered. In as it is condition vibration signals exhibits least possibility of unbalance and according to ISO 10816-3 Vibration level is found within "Permissible" limit. Thus, to incorporate unbalance, 10 gm mass was added to the impeller. Using exact solution

the mass ratio value was determined and accordingly the inbuilt DVA was designed. The vibration level according to stated ISO standard was found within the permissible limit for the 10 gm added mass case with DVA.

2. State of the art of the research topic: The study of vibrations of a mechanical system is one of the challenging problems in the industries. Suppression of unwanted vibration is an important goal in many applications such as machine tools, press tools, turbines, pumps, automobiles, marine applications, pipelines, aircrafts and also for the human operated tools. Even though lots of work has been done to avoid the unwanted vibrations, still we are lacking in solving this problem for rotating equipments. It is not possible to remove completely the vibrations from the system; however, one can reduce or minimize its effects on the system. Thus, Vibration control is one of the important areas needs to be explored to minimize the ill effects of the vibration. Such effects are the problem of chatter in the machining application that affects the quality of product, the unbalance of rotating parts of automobiles, turbines, pumps that infer the efficiency or comfort of user, and precision of the measurement or the accuracy of the measuring instruments, which are affected largely by the external excitations.

Mitigation of vibration in rotary system is challenging area of research. Designing a Dynamic Vibration Absorber (DVA) is one of the best and economic solution. DVA are secondary system with mass, stiffness and damping properties attached to a primary system and tuned with its natural frequency. The designing parameters are (1) mass ratio (2) Damping factor (3) Frequency ratio. The concept and design methodology of undamped DVA was first introduced by Watts (1883) and Frahm (1909) where the undamped primary system vibrations were controlled ideally by attaching undamped secondary system (DVA). Here the frequency tuning was done to mitigate vibrations of primary system. Den Hartog (1934) firstly fined the optimum solution of a damped DVA that is attached to an Undamped primary system subjected to harmonic excitation. He utilized ‘‘fixed-points theory’’ and concluded that the irrespective of frequencies the response amplitude of the primary mass is independent of the DVA absorber damping. Based on this optimality, Brock (1946) derived an analytical solution for the optimum damping ratio of the damped DVA.

Table 1: Optimal tuning parameter values obtained by various researchers.

Year	Method	f_{opt}	ξ_{dopt}	Result
1928 1934 1946	Fixed point theory	$\frac{1}{1+\mu}$	$\sqrt{\frac{3\mu}{8(1+\mu)}}$	Here all the vibration responses are passing through two common points irrespective of their different damping factor and mass ratio on either side of forcing frequency=1. The optimal damping ratio could be achieved when both the points become aligned.
1956	Fixed point theory	$\frac{1}{1+\mu}$	$\sqrt{\frac{3\mu}{8(1+\mu)^3}}$	The optimal damping factor is having value lesser compared to the previous study.

Liu,K., and Coppola,G (2010) a geometrically modified DVA where, damping agency of DVA is connected to ground directly was emerged. The optimal parameters were determined by using two numerical approaches, the first approach solves a set of nonlinear equations established by the Chebyshev's equioscillation theorem. And the second approach minimizes a compound objective. Both the methods were applied to a classical systems which has no damping agency and the results are compared with those from the analytical solutions. Then the modified Chebyshev's equioscillation theorem method was applied to find the optimum damped DVAs for the damped primary system . Noori and Farshidianfar (2013)The solutions to H∞ and H2 optimization problems of a variant dynamic vibration absorber (DVA) applied to suppress vibration in beam structures are derived analytically. The reduction in maximum amplitude responses and mean square motion of a beam using the traditional vibration absorber is compared with the proposed dynamic absorber. Numerical results show the non-traditional DVA under optimum conditions has better vibration suppression performance on beam structures than the traditional design of DVA. Furthermore, comparing H∞ and H2 optimization procedures shows that for a beam under random force excitation, use of H2 optimum parameters resulting in smaller mean square motion than the other optimization. Argentini et.al. (2015) had determined approximate tuning parameter solution for DVA attached to the rotating mass unbalance system. With the damping element it is very difficult to determine exact solution for optimal damping factor. Using perturbation the approximate solution determine by considering lower order terms. Here the damping of the primary system is neglected. Thus, by using perturbation method and l'Hospital rule with higher orders the optimal damping factor as a function of mass ratio power series could be found for the rotating mass unbalance system.

3. Problem Definition: Frequently it is found in industry that unbalance of the rotating components of primary damped or Undamped structure generates centrifugal force which is found to be a primary cause of forced vibration. As time elapse this unbalance induced vibration causes catastrophic failure of the system's important and costly components or the system itself. Mostly through condition monitoring the presence of unbalance is identified. Industry people are going for offline balancing where, a huge loss in terms of money and production occurs. DVA especially passive one are the cheapest solution if properly tuned. Thus to determine exact optimal damping factor of the DVA required to control vibration of the rotating unbalance system and its practical application are the aim of this research work.

4. Objective and Scope of work: following are the objectives set for the research work,

- To determine numerically a modified mass ratio value for the conventional passive DVA to attain same performance as obtained with the geometrically modified DVA.
- To carryout analytical analysis of the rotating unbalance system with DDVA.
- To determine exact optimal tuning parameters of Damped DVA for the vibration control of rotating unbalance system.
- To identify the presence of rotating Unbalance in the blower system using FFT analyzer.
- To design a passive DVA tuned with exact optimal damping factor for the rotating unbalance blower system.
- Determining the effect of tuned DDVA on rotating unbalance system experimentally and analyze its performance using ISO 10816-3.

From the literature review on Derivations of optimal tuning parameters and its implementation on the experimental setup it was found that the approximate tuning parameters performance was not up to the mark. Argentini et.al. (2015) was Only one researcher who had determined approximate tuning parameters for the unbalance system. And practically only static magnets and dual mass beam type DVA were used to control vibration induced due to unbalance. up to best of my knowledge. So the scope of work are as below,

- Deriving exact optimal damping factor using higher orders in perturbation method and L'Hospital rule.
- Design of passive DVA by implementing the derived exact optimal damping factor.

5. Original contribution by the thesis:

(1) The required tuning parameter modification in the conventional DVA was determined to attain same performance as offered by the geometrically modified DVA. The approximate solution was already determined by various researchers.

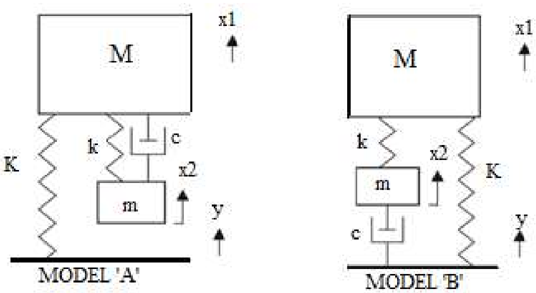


Figure.1.Conventional and ground hook (geometrically modified) lump configurations.

Table 2.Optimal parameters for both the models[11]

Model	forcing frequency ratio	optimum damping factor
A	$\frac{1}{(1+\mu)}$	$\sqrt{(\,3\mu\,/\,8(1+\mu))}$
B	$\frac{1}{\sqrt{(1-\mu)}}$	$\sqrt{(\,\mu(3-\mu)\,/8)}$

From the numerically investigation using MATLAB we found that performance of the ground hook DVA was better compared to conventional design. So the rise required with the tuning parameter of the conventional DVA was determined numerically using the same software to attain same performance as with the ground hook DVA. With the application of tuning parameter modification following result was obtained.

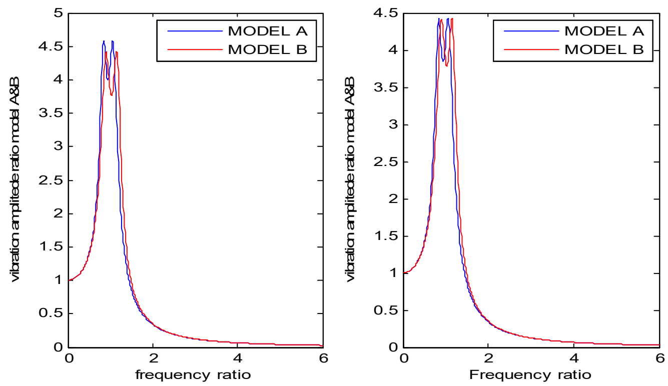


Figure 2: Amplitude ratio to frequency ratio for mass ratio 0.1 (left) equal amplitude (right) of Model A' and 'B' with mass ratio 0.1081 and 0.1 respectively.

Limitations of research with this topic:

(A) The **system is classical** and it is under the **exposure of ground motion excitation**.

(B) There is **absence of unbalance excitation**.

Due to above limitations we look forward in the following direction for research.

(2) The exact optimal tuning damping factor of DVA is determined analytically using *L'Hospital* rule with the higher orders of the polynomials used in the perturbation method. The performance of the tuning damping factor of DVA is compared with the existing solutions like,

(a) Solution proposed by Ioi & Ikeda.

(b) H∞ optimization method solution.

(c) Equivalent Linearization method solution

(d) System without DVA.

The optimal mass ratio range for designing DVA is 0.05-0.15 for which damping factor differs by larger amount compared to approximate solution.

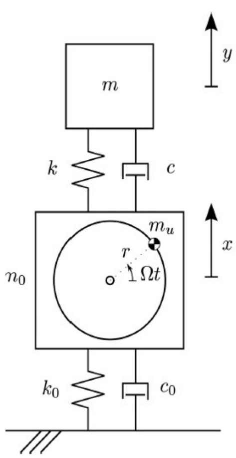
(3) DVA have been designed for two different rotating unbalance blower system (1) operating speed 2550 RPM (2) 720 RPM operating speed. For 1st case, the DVA is externally connected to the system while in the 2nd case it is an integral part of the system. In both the cases vibration responses before and after attaching DVA was determined using FFT analyzer. According to ISO 10816-3 for both the cases DVA had worked satisfactorily and vibration was under the controlled condition.

6. Methodology of Research, Results / Comparisons

From the literature review it is clear that most of the researchers have given approximate tuning parameters as it is difficult to determine tuning parameters of damped primary system. Very few researchers have given focus in the area of rotating mass unbalanced damped primary system vibration mitigation using DVA. The entire work have been divided in to following steps,

1. To determine analytically the exact optimal tuning parameters of DVA.
2. Experimental application of the optimally tuned DVA to bring vibration control as per ISO 10816-3 for rotating equipments.

6.1 Determination of Exact Optimal Tuning of Conventional Dynamic Vibration Absorbers to Control Vibration of undamped primary system due to Rotating Mass Unbalance



$$\therefore \begin{bmatrix} (1+\mu)\ddot{x}+\mu\ddot{y} \\ \mu\ddot{x}+\mu\ddot{y} \end{bmatrix} + \begin{bmatrix} 2\zeta\omega\dot{x} \\ 2\zeta_d\omega_d\mu\dot{y} \end{bmatrix} + \begin{bmatrix} {\omega_0}^2 x \\ {\omega_d}^2\mu y \end{bmatrix} = \begin{bmatrix} \frac{m_u}{m_0} r\Omega^2 e^{i\omega t} \\ 0 \end{bmatrix}$$

$$\therefore (1+\mu)\ddot{x}+\mu\ddot{y}+2\zeta\omega\dot{x}+{\omega_0}^2 x = \frac{m_u}{m_0} r\Omega^2 e^{i\omega t} \qquad (1)$$

$$\mu\ddot{x}+\mu\ddot{y}+2\zeta_d\omega_d\mu\dot{y}+{\omega_d}^2\mu y = 0 \qquad (2)$$

Figure 3: Rotating mass unbalance system with damped DVA.

To eliminate time variable from the equation of motion some characteristic parameters of the coupled system are introduced, and they are as follows,

$$\tilde{x} = \frac{x}{\frac{m_u}{m_0}r},\ \tilde{y} = \frac{y}{\frac{m_u}{m_0}r},\ \tilde{t} = \Omega t,\ f = \frac{\omega_d}{\omega_0},\ g = \frac{\Omega}{\omega_0}$$

Where, $\tilde{x}$, $\tilde{y}$ and $\tilde{t}$ are non-dimensional expressions of x, y and t

$$\begin{bmatrix} (1+\mu)g^2 & \mu g^2 \\ \mu g^2 & \mu g^2 \end{bmatrix} \begin{Bmatrix} \ddot{\tilde{x}} \\ \ddot{\tilde{y}} \end{Bmatrix} + \begin{bmatrix} 2\zeta g & 0 \\ 0 & 2\zeta_d\mu f g \end{bmatrix} \begin{Bmatrix} \dot{\tilde{x}} \\ \dot{\tilde{y}} \end{Bmatrix} + \begin{bmatrix} 1 & 0 \\ 0 & f^2\mu \end{bmatrix} \begin{Bmatrix} \tilde{x} \\ \tilde{y} \end{Bmatrix} = \begin{Bmatrix} g^2 \\ 0 \end{Bmatrix} e^{i\tilde{t}}$$

According to the fixed point theory irrespective to the DVA damping factor all responses will pass through two fixed points Let's say a and b. This condition persists when $\left(\frac{\partial \text{IXI}}{\partial g}\right)$ a, b= 0. This means that the magnification factor value at a and b will be equal as the slope at those points becomes zero. This theory performs well for the classical systems with small amount of damping factor.

The solution of the above matrix is done by considering conventional $\left|\tilde{X}\right|_{a,b}$ we have considered $\left|\tilde{X}\right|^2{}_{a,b}$ and the solution is as follows,

$$\left\{\begin{matrix} {\zeta_d}^2 = \frac{A^2 - C^2|\widetilde{X}|^2}{4(D^2|\tilde{X}|^2 - B^2)} \\ \text{and} \\ \left|\tilde{X}\right|^2 = \left|\tilde{X}\right|^2{}_{a,b} = \frac{2}{\mu(1+\mu)} \end{matrix}\right\}$$ Where, $A = g^2(f^2 - g^2), B = fg^3$,

$$C = g^4 - \left(1 + f^2 {f_\infty}^{-2}\right)g^2 + f^2,\ D = fg(1 - g^2 {f_\infty}^{-2})$$

Applying perturbation method, with the assumption $\delta \to 0$

$$g^2 = {g_{a,b}}^2 + \delta$$

$$\therefore \lim_{\delta \to 0} {\zeta_d}^2 = \lim_{\delta \to 0} \frac{P_0 + P_1\delta + P_2\delta^2 + P_3\delta^3 + \cdots}{Q_0 + Q_1\delta + Q_2\delta^2 + Q_3\delta^3 + \cdots}$$

For more accurate optimal damping factor of DVA the L'hospital rule is applied with higher orders of 'δ' as follows,

$$= \lim_{\delta^2 \to 0} \frac{P_0 + P_1\sqrt{\delta^2} + P_2\delta^2 + P_3(\delta^2)^{3/2}}{Q_0 + Q_1(\delta^2)^{1/2} + Q_2\delta^2 + Q_3(\delta^2)^{3/2}}$$

$${\zeta_d}^2 = \frac{\left[g^2(f^2 - g^2)\right]^2 - \left[g^4 - \left(1 + f^2 {f_{opt}}^{-2}\right)g^2\right]^2 \frac{2}{\mu(1+\mu)}}{4\left[\left\{fg\left(1 - g^2 {f_{opt}}^{-2}\right)\right\}^2 - \left(\frac{2}{\mu(1+\mu)}\right)^2 - (fg^3)^2\right]}$$

$$= \frac{g^4(f^2 - g^2)^2 - (g^8 - 2g^4\left(1 + f^2 {f_\infty}^{-2}\right)g^2 + \left(1 + f^2 {f_\infty}^{-2}\right)g^4\,)\frac{2}{\mu(\mu+1)}}{4(\left(f^2g^2\left(1 - 2g^2 {f_\infty}^{-2} + g^4 {f_\infty}^{-4}\right)\frac{4}{\mu^2(\mu+1)^2}\right) - f^2g^6)}$$

$$= \frac{16(\mu(1+\mu))^{1/2} - \frac{2}{\mu(\mu+1)} - 4(\mu)^{3/2}(1+\mu)^{1/2} + \frac{2\mu}{(\mu+1)}}{\frac{32\sqrt{2}}{\mu^{3/2}(1+\mu)^{1/2}} - \frac{16\sqrt{2}}{(1+\mu)^{1/2}} - \frac{64}{(1+\mu)} + \frac{16}{(1+\mu)}\left(\frac{2}{\mu(\mu+1)}\right)^{1/2}}$$

$${\xi_d}^2 = \frac{\frac{16(\mu(\mu+1))^{\frac{3}{2}} - 2 - 4\mu^{\frac{5}{2}}(\mu+1)^{\frac{3}{2}} + 2\mu^2}{\mu(\mu+1)}}{\frac{32\sqrt{2}(\mu+1) - 16\sqrt{2}\mu^2(\mu+1) - 64\mu^{\frac{1}{2}}(\mu+1)^{\frac{1}{2}} + 16\sqrt{2}\mu}{\mu^{\frac{3}{2}}(\mu+1)^{\frac{3}{2}}}}$$

$$\xi_{d(opt)} = \sqrt{\frac{4\mu^{10} + 12\mu^9 - 52\mu^8 - 188\mu^7 - 181\mu^6 - 52\mu^5 - 192\mu^4 - 129\mu^3 - 60\mu^2}{128\mu^6 + 128\mu^4 + 384\mu^2 + 1024\mu - 512}}$$

Going with the exact solution the requirement of Optimal damping factor value for 0-30% mass ratio is considerably less compared to the approximate solution. Up to 6% mass ratio, the optimal damping factor required for the classical system and approximate solution are almost equal and after that as shown in the following figure the requirement differs. Referring to Fig. 4 we can observe that the blue colored line is showing relation of mass ratio and DVA damping factor. Appreciable difference between approximate result and exact solution is observed between mass ratio ranges 0.05 to 0.15 as the required Optimal damping factor with exact solution found to be reduced in the range of 75% - 85%.

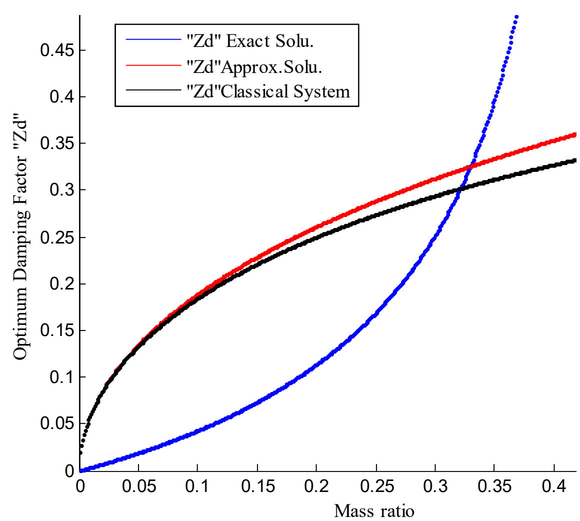


Figure 4: Comparison of Exact DVA Optimal damping factor (Zd=ξ_d) to o ther solutions for the Rotating mass unbalance system with DVA.

In the following Fig.5 and 6, the presented exact analytical solution is numerically compared with the approximate solution, solution proposed by Ioi & Ikeda, H∞ method, Equivalent Linearization method, and system without DVA.

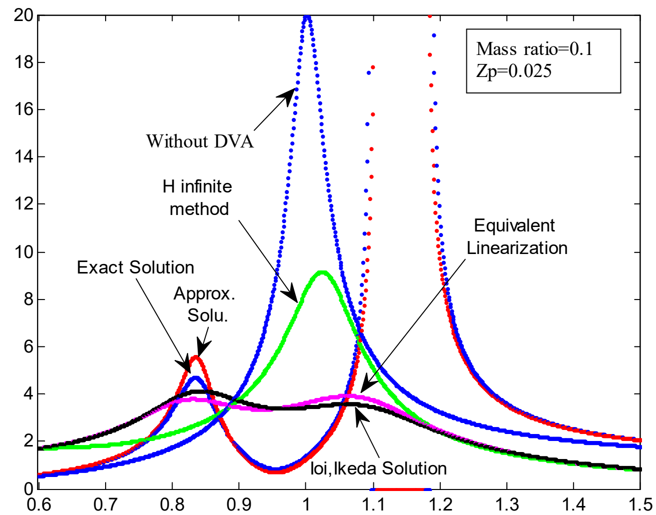


Figure 5: Comparison of Exact DVA solution to other solutions for the Rotating mass unbalance system with DVA mass ratio 0.1.

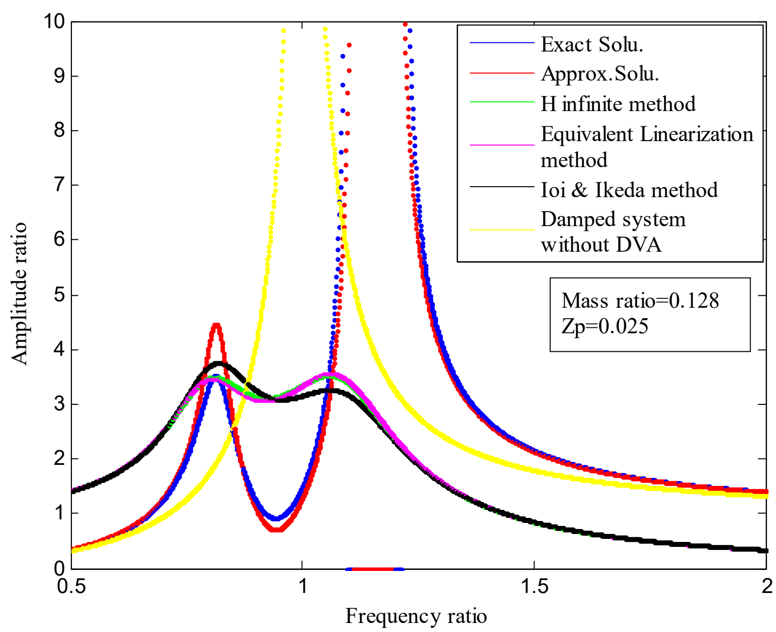


Figure 6: Comparison of Exact DVA solution to other solutions for the Rotating mass unbalance system with DVA mass ratio 0.128.

6.2 Experimental Implementation

Case 1: A setup has been made by using a blower which runs at 2550 rpm with the help of a motor of 0.5 HP as an external drive. The blower rotor mass is 1.168 Kg. with one blade mass 0.015 Kg. Totally there are 36 blades. The blower rotor rotates in a casing mounted 6300 pre-lubed with grease and sealed precision single-row ball bearing.

Figure 7: Blower setup without DVA.

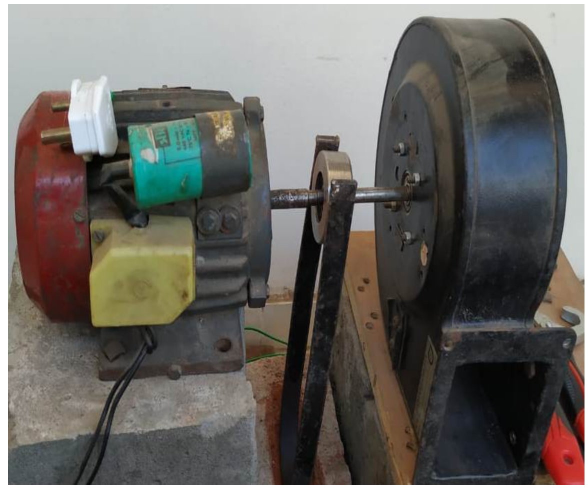


Figure 8: Blower setup with DVA.

For the placement of DVA, a little bit of modification is done. The motor drive-end shaft is extended for a length of 10mm. For creating unbalance, one of the blade is deliberately broken. Utmost care is taken for the proper foundation of the motor and the blower.

Table 3: Vibration V_{rms} using FFT analyzer

Direction	Before attachment of DVA (V_{rms}) Motor Drive End	After attachment of DVA (V_{rms}) Motor Drive End	Before attachment of DVA (V_{rms}) Motor Non- Drive End	After attachment of DVA (V_{rms}) Motor Non- Drive End
Horizontal	10.18	9.00	4.53	1.34
Axial	10.39	0.70	8.98	3.04
Vertical	4.03	0.42	4.24	3.32

From the FRF, it was seen that the peaks are mostly in a 1X RPM pattern measured both in a vertical and horizontal direction. As per the condition monitoring basics, the presence of rotating unbalance is justified. As per ISO 10816-3 below 15 KW external driver rigid support rotating systems the measured Vrms value should be less than 4.5 mm/s for unlimited long term operation allowable condition. Looking at Table 2 it is concluded that before attaching DVA all the Vrms readings are almost above 4 mm/s at the motor drive end and motor non-drive end. In the following section we will design DVA such that it will absorb vibration induced due to the broken blade.

DVA Design

Form the derived exact optimal damping factor equation it was come to the notice that to design DVA either damping material or mass ratio need to be decided first. Here we are considering rubber as a damping material having damping factor 0.09 together. Corresponding to this value the mass ratio from the damping factor v/s mass ratio curve we get mass ratio value equal to 0.18.

$$\xi_{d(opt)} = \sqrt{\frac{4\mu^{10} + 12\mu^9 - 52\mu^8 - 188\mu^7 - 181\mu^6 - 52\mu^5 - 192\mu^4 - 129\mu^3 - 60\mu^2}{128\mu^6 + 128\mu^4 + 384\mu^2 + 1024\mu - 512}}$$

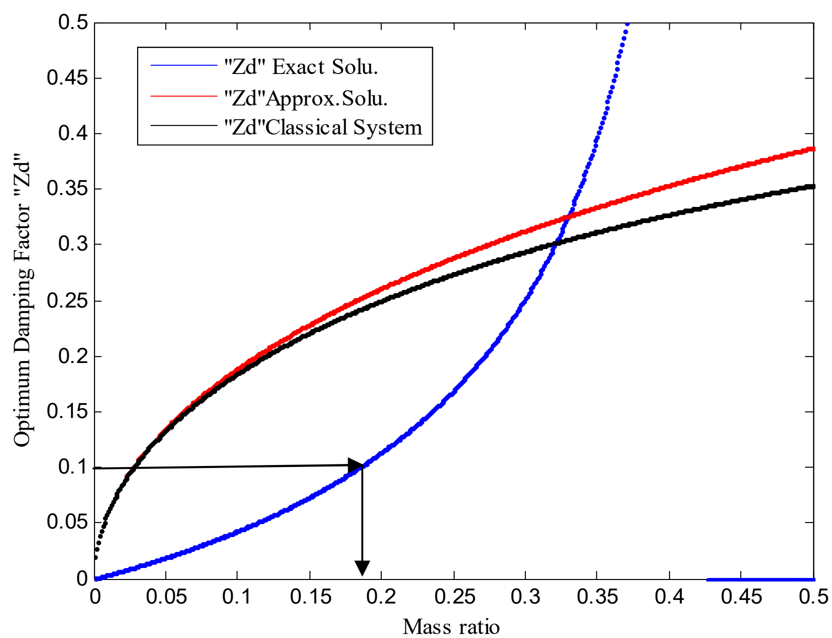


Figure 9: Expected mass ratio for DVA design.

The proposed DVA consists of an outer hollow ring with outer diameter 60 mm and inner diameter 40 mm with Wall thickness 20mm. The rubber strip is placed on the inner hollow side having side wall thickness 5 mm so the damping material strip will get adjusted. Eccentricity is 1mm. 6800-2RS (10x19x5 mm) Rubber Sealed Ball Bearing is placed at the center and the drive end extended shaft passes through the inner hole of 10mm. 2RS or 2 Rubber seals indicating they have lip seals on both sides and are pre-filled with grease for life, so no lubrication is required. The seals give some protection against ingress of moisture and contaminating particles entering the raceways. Mass ratio= DVA mass/ Rotor mass= 0.179.

Figure 10: Proposed Dynamic Vibration Absorber.

CASE 2: DVA- an integral part of casing

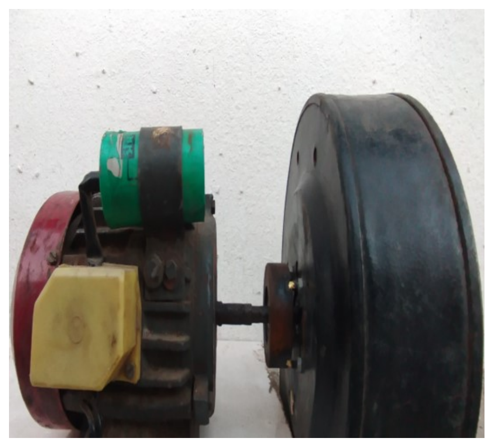


Figure 11: Detailed Setup of blower attached to DVA and Impeller with eccentric mass.

In the earlier case the DVA was kept outside the blower setup now here it was attached to the casing. The purpose behind this was to investigate the effectiveness of the DVA being an integral part of the unbalanced system. The same DVA with geometrical modified attachment is used. From following table it was noticed that the DVA is more effectively working.

Table 4: Vibration V_{rms} using FFT analyzer

Measuring point	**Direction**	**Before attachment of DVA**	**After attachment of DVA**
		Vrms (mm/s)	Vrms (mm/s)
Motor Drive End	Horizontal	8.74	3.65

	Axial	4.78	1.08
	Vertical	9.34	3.64
Motor Non-Drive End	Horizontal	6.4	3.12
	Axial	5.44	2.33
	Vertical	13.66	4.13

CASE 3: Design of DVA for blower with low speed

To see the effect of exact optimal damping parameters derived in the previous chapter here a second case is considered where, fan speed is low and it is 720 RPM.

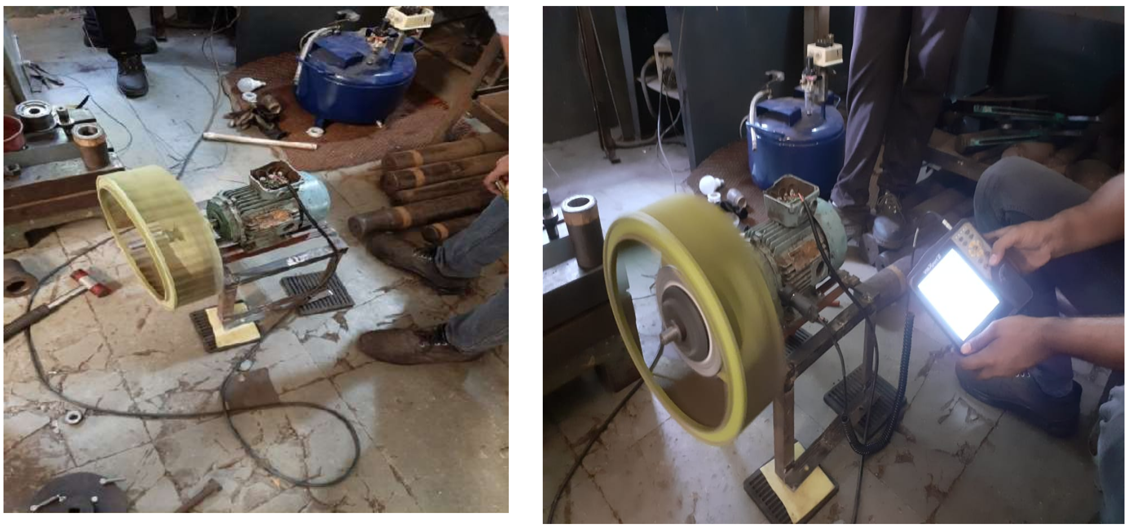


Figure 12: Experimental investigation of blower setup with DVA

The blower assembly is mounted on a rubber pad for better isolation. The FFT analyzer is used for recording the vibration peaks. Vibration spectrum is measured in Horizontal, Vertical and Axial direction.The decision regarding blower condition was estimated using ISO-10816-3.

Table 5: Vibration readings in V_{rms} without attachment of DVA

720 RPM Fan speed		V_{rms}		
		Horizontal H	Vertical V	Axial A
Without added mass	DE	3.4	1.4	2.8
	NDE	2.9	0.8	2.9
With added mass	DE	8.1	2.5	4.9
	NDE	9.5	1.5	5.4

In the above table we can see the vibration readings of the blower arrangement without DVA attachment and added mass. Without added mass condition of blower fan Vibration level is within “Permissible” limit as per ISO-10816-3. Thus, to incorporate unbalance we need to add some mass externally. So we add 10 gm mass to make the impeller unbalance. Now to reduce Vibration level the DVA need to be designed and attached to the impeller. Here as a damping agent neoprene rubber bush is taken whose damping factor is 0.05. Now using the numerical comparison graph of the exact damping factor, approximate damping factor and damping factor for classical system v/s mass ratio we are able to get mass ratio. Here we have previously selected the damping factor so accordingly we get the mass ratio value between 0.12 to 0.13.

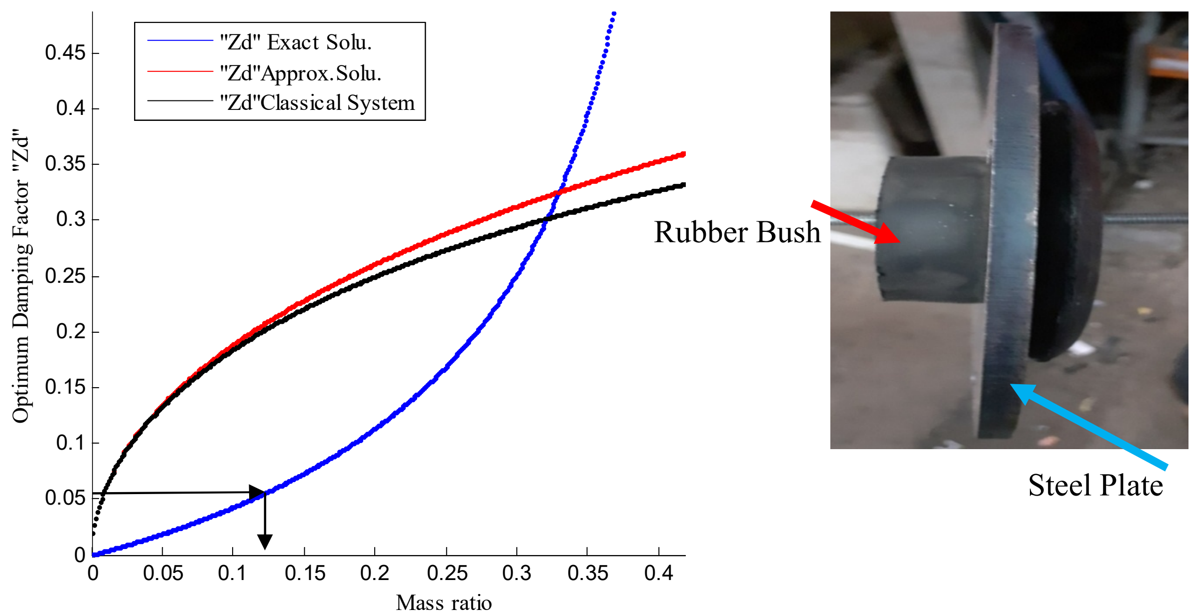


Figure 13: Determination of mass ratio value and DVA design.

1. Circular Hollow disc mass : 0.3235Kg.
2. Impeller Mass: 1.875 Kg and M.S Hub Mass: 0.385 Kg.

Mass ratio = DVA mass/ Rotor mass

= 0.129

Table 6: V_{rms} values at drive and Non-drive end of blower motor.

720 RPM		Measuring Location	Vrms with DVA	Vrms with DVA and 10gm added mass to impeller
Blower Motor	DE	H	1.9	4.6
		V	0.5	1.5
		A	0.8	3.4
	NDE	H	1.3	4.0
		V	0.3	0.4
		A	0.9	2.8

7. Achievements with respect to objectives: Referring to the literature, here an attempt was made to design DVA to control rotating mass unbalance. The achievement in regards of the predefined objectives is as follows,

(1) Numerically a modified mass ratio value for the design of conventional passive DVA to attain same performance as obtained with the geometrically modified DVA was done. The outcome is the following design table which reflects modified new mass ratio and damping factor value of the conventional DVA.

Table 7: Design data for the required modification in the tuning parameter of conventional DVA to attain same performance as obtained with the ground hook DVA.

Old mass ratio model A	Model A amplitude ratio X_1	ξd model 'A'	X_2 model 'B'	ξd model 'B'	New Mass ratio Model 'A'	New Amp. Ratio model A	New ξd model 'A'
0.01	14.1796	0.0609	14.1078	0.0611	0.0101	14.1078	0.0612
0.02	10.0528	0.0857	9.9454	0.0863	0.0204	9.9453	0.0867
0.03	8.2091	0.1045	8.1378	0.1055	0.0306	8.1377	0.1055
0.04	7.1461	0.1201	7.0326	0.1217	0.0413	7.0326	0.122
0.05	6.407	0.1336	6.2883	0.1358	0.0519	6.2883	0.136
0.1	4.5901	0.1846	4.4236	0.1904	0.1081	4.4236	0.1913
0.11	4.3877	0.1928	4.213	0.1993	0.1199	4.213	0.2004
0.12	4.2116	0.2004	4.0291	0.2078	0.1319	4.0291	0.209

0.13	4.0565	0.2077	3.8667	0.216	0.1441	3.8667	0.2173
0.14	3.9187	0.2146	3.7219	0.2237	0.1565	3.7219	0.2253
0.15	3.7953	0.2212	3.5916	0.2312	0.1691	3.5916	0.2329
0.16	3.6838	0.2274	3.4736	0.2383	0.1818	3.4736	0.2402
0.17	3.5824	0.2334	3.366	0.2452	0.1948	3.366	0.2473
0.18	3.4898	0.2392	3.2674	0.2519	0.2081	3.2674	0.2542
0.19	3.4047	0.2447	3.1766	0.2583	0.2217	3.1766	0.2609
0.2	3.3262	0.25	3.0924	0.2646	0.2355	3.0924	0.2674

(2) The analytical analysis of the damped primary rotating unbalance system with damped dynamic vibration absorber was carried out. The forced vibration case has been formed having harmonic imaginary exponential frequency function. The magnitude of excitation force is influenced by the unbalance force. During simplification of two equation of motion the variable time 't' brings hurdle in simplification to derive Magnification Factor equation. Thus, this analysis should be carried out using combination of perturbation method and l'hospital rule.

(3) To determine exact tuning parameter in the analysis of the considered rotating unbalance system with DDVA, the time variable elimination from the equation of motion some characteristic parameters of the coupled system were introduced. Here the frequency response function was formed. Relation between frequency ratio and magnification factor was derived. The exact optimal tuning parameters of Damped DVA for the vibration control of rotating unbalance system was derived using Applying perturbation method followed by L'hospital rule with higher orders of 'δ' i.e. lim $\delta^2 \rightarrow 0$. After a rigorous mathematical operation the exact damping parameter solution was obtained. The solution gives relation between required damping factor of DVA and mass ratio. Using this relation a design curve was plotted which gives relation between the stated dimension less parameters. The same was utilized in the design of DVA for vibration control of unbalance rotary structure prepared for experimental investigation.

(4) The structure for experimental investigation was industrial blower with induced unbalance in the impeller. The operating speed of the blower for 1st and 2nd case was 2550 RPM and for 3rd case it was 720 RPM. The vibration spectrum and frequency peaks repetition pattern was observed using the frequency response obtained in all three directions i.e. Horizontal, Vertical and Axial at motor drive end and Motor non-drive end using FFT analyzer. The frequency

peaks mostly shows 1X peak repetition pattern. Thus, the presence of unbalance was confirmed. In the 3rd case impeller was made of Poly Propylene material. The vibration signals were recorded firstly by breaking two blades but the result was not significant. Now unbalance was introduced by adding 10gm mass to the impeller. The recorded vibration frequency response presents 1X repetition. Thus, in all cases according to the ISO 10816-3 rotating unbalance case has been reflected.

(5) The Design of passive DVA tuned with exact optimal damping factor for the rotating unbalance blower system was done using the design curve presented in Fig. 9 & 13. Firstly the damping material is decided which gives damping factor using it in design curve we would get the mass ratio. Here the mass of rotor is known in case 1, 2 & 3 so mass of dynamic vibration absorber was available using the design graph. The outer ring mass can be easily calculated from the mass ratio and primary system mass.

(6) The passive DVA tuned with exact optimal damping factor for all three rotating unbalance blower system cases were designed. For 1st case the DVA was acting externally and in the 2nd case it was a integral part of the impeller. Before attachment of DVA, system performance for 1st case and 2nd case was not acceptable as per ISO 10816-3. After attaching designed tuned DVA vibration V_{rms} value have been fall below 4.5 mm/s which is the indication of "System health in GOOD condition". In the 3rd case again DVA was designed with exact damping parameter derived equation system. The attached 10 gm mass system vibration which was above the permissible limit has been brought in permissible limit. Looking to all the V_{rms} values after attaching DDVA the system health had fall in GOOD category as per ISO 10816-3.

8. Conclusion:

The exact solution for damping factor as a function of mass ratio have been determined by using l'Hospital rule in its higher orders. The effect of exact solution have been compared numerically with approximate solutions obtained by using H∞ method, Equivalent Linearization method, solution proposed by Ioi & Ikeda and also system without DVA. Compared to all the listed methods the proposed exact DVA damping factor required is far away from the approximate results. For a mass ratio range 0.05 to 0.18 with the exact solution we appreciated 75% to 85% percentage reduction in the required damping factor compared to the approximate solution.

CASE 1 is of designing passive DVA for an industrial unbalanced air blower operating speed 2550 RPM. To design a passive DVA, Fig.4 is used. Mass ratio of DVA is determined using a known damping factor value in the exact solution. The vibration signals were recorded in horizontal, vertical and axial directions using FFT analyzer. Comparing both before and after attaching passive DVA vibration signals, an appreciable reduction in V_{rms} value in Vertical and axial measuring points of motor drive end and axial and horizontal measuring points of motor non-drive end. The designed DVA is placed external to blower assembly thus functioning externally. In CASE 2 we were move forward by using DVA as an integral part of the blower. The integral DVA had given remarkable vibration reduction for the unbalanced blower used in 1st case.

CASE 3 is of designing passive DVA for an industrial unbalanced air blower operating speed 720 RPM. The designed DVA was attached to the impeller directly. Vibration readings were recorded in two conditions of blower. The effectiveness of the proposed arrangement for vibration reduction was compared according to ISO 10816-3. Without added extra mass and with added 10 grams mass to impeller designed DVA performance was recorded using FFT analyzer and according to ISO 10816-3the blower vibration is within permissible limit.

9. Research Paper Publications detail

Sr. No	Title	Name of journal	SCOPUS approved	SCI/ SCIE	Status
1	Numerical Investigation For The Determination Of Equal Vibration Amplitude Of The Linear Undamped Vibrating System Using Conventional And Ground Hook Dynamic Vibration Absorber By Modification In The Damping Factor And Mass Ratio	International Journal of Advanced Science and Technology	YES	NO	Published
2	Determination of Exact Optimal Tuning of Dynamic Vibration Absorbers to Control Vibration Due to Rotating Mass Unbalance	Springer Nature Lecture Notes in Mechanical Engineering	YES	NO	Published
3	Design and Implementation of Passive Dynamic Vibration Absorber for Vibration control of rotating mass unbalanced Air Blower	Journal of Engineering Research, Kuwait University	YES	YES	Published

10. References:

[1] Watts, P. (1883), ‘‘On a method of reducing the rolling of ship at sea,’’ Transactions of the Institute of Naval Architects, Vol. 24, pp. 165–190.

[2] Frahm, H., (1909), ‘‘Device for damping vibrations of bodies,’’ U.S. Patent No. 989958.

[3] Ormondroyd, J. and Den Hartog, J.P.,(1928), ‘‘Theory of the dynamic vibration absorber,’’ Transactions of the American Society of Mechanical Engineers, Vol. 50, pp. 9–22.

[4] Den Hartog, J.P.,(1934), Mechanical Vibrations, McGraw-Hill, New York.

[5] Brock, J. E., (1946), ‘‘A note on the damped vibration absorber,’’JournalofApplied Mechanics,Vol. 68, pp. A-284.

[6] J.P.DenHartog,(1956), Mechanical Vibrations, McGraw-Hill,NewYork.

[7] Toshihiro IoI and Ken IKEDA,(1978) “On the Dynamic Vibration Damped Absorber of the Vibration system”, Bulletin of the JSME, Vol. 21, No. 151.

[8] G.B.Warburton,(1981),“Optimum absorber parameters for minimizing vibration response”, Earthquake Engineering and Structural Dynamics Vol.9.

[9] Brock, J. E.,(1946), ‘‘A note on the damped vibration absorber,’’ Journal of Applied Mechanics,Vol. 68,A-284.

[10] Liu,K., and Coppola,G.,(2010) “Optimal design of damped dynamic vibration absorber for damped primary systems,” Canadian society for mechanical engineering, Vol.34,pp. 119-135.

[11] Wong, W. and Cheung, Y.,(2008), ‘‘Optimal design of a damped dynamic vibration absorber for vibration control of structure excited by ground motion,’’ Engineering Structures, Vol.30, pp.282–286.

[12] T.Ioi and K.Ikeda,(1978), “On the dynamic vibration damped absorber of the vibration system”, Bulletin of the Japanese Society of Mechanical Engineering, Vol. 21, No. 151, pp.64-71.

[13] S.E.Randall, D.M.Halsted, D.L Taylor,(1981), “Optiomum Vibration Absorbers for linear damped system”, J.Of Mechanical Design, ASME, Vol.103, No.4, pp. 908-913.

[14] H C Tasi, G C Lin, (1993), “Optimum Tuned Mass Dampers for minimizing steady-state response of support excited and damped systems”, Earthquake Engg. And structural dynamics, Vol. 22, Nol.11, pp. 957-973.

[15] N. D. Anh and N. X. Nguyen,(2011), “Extension of equivalent linearization method to design of TMD for linear damped systems”, Struct. Control Health Monit., John Wiley & Sons, Ltd.
[16] B. Noori · A. Farshidianfar,(2013), “Optimum design of dynamic vibration absorbers for a beam based on H∞ and H2 Optimization”, Arch Appl Mech.

[17] ND Anh, NX Nguyen,(2014), “Design of non-traditional dynamic vibration absorber for damped linear structures”, Journal Of Mechanical Engineering Science, SAGE Publication.
[18] T. Argentini, M. Belloli, P. Borghesani, (2015) “A Closed-Form Optimal Tuning of Mass Dampers for One Degree-of-Freedom Systems Under Rotating Unbalance Forcing”, Journal of Vibration and Acoustics, ASME.
[19] Hongliang Yao, Tongzhao Wang, Bangchun Wen and Baijun Qiu,(2018) “A tunable dynamic vibration absorber for unbalanced rotor system”, Journal of Mechanical Science and Technology, Vol.32 (4), pp.1519~1528.
[20] Asd said Juma, AL Zadjali & G.R.Rameshkumar, (2013), “conditioning monitoring of centrifugal blower using vibration analysis”, International Journal of Multidisciplinary Sciences and Engineering, Vol.4, NO. 5.
[21] Abdullah Ozer, et.al.,(2015),"Design and Experimental Implementation of a Beam-Type Twin Dynamic Vibration Absorber for a Cantilevered Flexible Structure Carrying an Unbalanced Rotor: Numerical and Experimental Observations", Hindawi Publishing Corporation Shock and Vibration.