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THEORETICAL AND EXPERIMENTAL STUDY FOR CONTROLLING VIBRATION OF A PARTICULAR SYSTEM USING TUNED DAMPER

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Abstract:

The vibration problems are accomplished in many application areas, such as in automobiles, aircrafts, helicopters, machines...etc., vibration may cause systems failure or reduce their performance. This paper studies the experimental and theoretical control of vibration using a developed tuned damper (based on variable piston area) to provide optimal vibration reduction at different conditions. The experimental work was carried out on a particular system, which mainly composes of beam, harmonic excitation source (exciter), passive spring and tuned damper. Also, the system is provided by an electronics system to acquire the sensor signal and drive the damper according to control algorithm. The neural network system identification (NNSI) was utilized to identify all system components as one model based on the inputs - output data that is collected from the experiment. In addition, the mathematical model of the system was derived and used with experimental damping ratio at different damper piston angles (0-60°) to verify the experimental results. The implemented controller is a model predictive controller (MPC), which was designed based on NNSI in MATLAB Toolbox to control the system vibration. The MPC explicitly handles the measurable disturbance (excitation frequency) of the predictive model. The results showed that the model predictive controller with a semi-active tuned damper is effectively reduced the root mean square (RMS) of acceleration in the resonance region in which the maximum time of vibration reduction from higher RMS to minimum level is 1.39 second. The experimental and theoretical works are relatively converged and the total range of percentage error between the experimental and theoretical RMS of acceleration at resonance is (4.9-10) %.

Keywords: *vibration control; tuned damper; semi-active damper; model predictive controller.*

دراسة نظرية وعملية للسيطرة على اهتزاز منظومة معينة باستخدام مخمد متغير التخميد

الخلاصة:

ان مشاكل الاهتزازات متواجدة في مختلف التطبيقات كالسيارات والطائرات والمروحيات ومختلف الآلات ...الخ، حيث ان الاهتزازات قد تؤدي الى تعطيل الانظمة او التقليل من ادائها. يدرس البحث السيطرة على الاهتزازات نظرياً وعملياً باستخدام مخمد متغير التخميد (بالاعتماد على تغير

مساحة مكبس المخمد) وذلك لتقليل الاهتزازات بصوره حسنه لمختلف الظروف. طبق الجانب العملي على نظام معين والذي يتألف بصورة اساسية من عتبة ، هزاز اثارة توافقي، نابض ومخمد متغير التخميد. جهز النظام بمنظومة الكترونيات للحصول على اشارة المتحسسات ولتحريك اجزاء المخمد وفقا لخوارزمية مسيطر معين. لقد استخدام التمييز بالشبكة العصبية لوصف جميع مكونات النظام كنموذج واحد بالاعتماد على بيانات للمدخلات والمخرجات التي تم جمعها من الجزء العملي . بالإضافة الى ذلك، تم انشاء أنموذج رياضي للنظام واستخدم مع نسبة التخميد المحسوب من الجانب العملي باختلاف زوايا المكبس (٠-٦٠°) وذلك لغرض التحقق من صحة النتائج العملية. ونفذت السيطرة باستخدام مسيطر تنبؤي نموذجي والذي تم تصميمهُ بالاعتماد على التمييز بالشبكة العصبية بواسطة ادوات الماتلاب وذلك للسيطرة على اهتزاز النظام وهذا المسيطر قد استخدم لكونه يأخذ بنظر الاعتبار تأثير الاضطراب المقاس في أنموذج التنبؤ. وأظهرت النتائج ،ان المسيطر التنبؤي النموذجي مع المخمد المتغير شبه الفعال قد خفض قيمة جذر متوسط التربيع للتعجيل بشكل فعال في منطقة الرنين ، إذ ان اطول مدة زمنية لتقليل الاهتزاز من اعلى قيمة لجذر متوسط التربيع للتعجيل الى اقل مستوى هي ١,٣٩ ثانية، إذ ان قيم جذر متوسط التربيع للتعجيل العملية والنظرية متقاربة نسبيا والنسبة الاجمالية للخطأ في منطقة الرنين هي (٤,٩-١٠)%.

الكلمات الدليلة : السيطرة على الاهتزازات، مخمد متغير التخميد، مخمد شبه فعال، مسيطر تنبؤي انموذجي .

Nomenclature

Symbol	***Description***	***Unit***
C_s	State damping coefficient	N.s/m
F_o	Maximum value of excitation force	N
g	Gravitational acceleration	m/s^2
I_A	Mass moment of inertia of the system about fulcrum point of beam	kg.m^2
K	Stiffness of spring	N/m
L	Length of beam	M
L_1	Distance between the damper and fulcrum point of beam	M
L_2	Distance between the motor and fulcrum point of beam	M
L_3	Distance between the spring and fulcrum point of beam	M
M	Mass of beam	Kg
M	Mass of the combined part (motor and disk)	Kg
m_d	Mass of damper	Kg
m_u	Unbalance mass corresponding to hole of disk	Kg
r_u	Radius to center of hole	M
x	Displacement of beam end	M
T	Time of vibration reduction	Sec
T_D	Time period	Sec
$\ddot{x}_x$ & $\ddot{x}_y$	Two consecutive peak acceleration	m/s^2
Ω	Angular velocity of rotation for the disk	rad/s

$\boldsymbol{\omega_d}$	Angular velocity of damped system	rad/s
$\boldsymbol{\omega_n}$	Natural frequency of the system	rad/s
$\boldsymbol{\zeta_s}$	State damping ratio	-
Θ	Angular displacement of beam	Rad
ϕ	Damper disc angle	deg.
$\boldsymbol{\delta}$	Logarithmic decrement	-

1. Introduction

The problem of reducing the level of vibration in constructions and structures arises in various branches of engineering, technology and industry. In most of today's mechatronic systems, a number of possible devices, such as reaction or momentum wheels, rotating devices, and electric motors are essential to the system's operation and performance. These devices can also be sources of detrimental vibrations that may significantly influence the mission performance, effectiveness, and accuracy of operation. Therefore, there is a need for vibration reduction. Several techniques are utilized to either limit or alter the vibration response characteristics of such systems. Generally, there are three main approaches used to control vibrations; passive vibration control, active vibration control and semi-active vibration control. A passive system comprises springs and dampers which are the most traditional elements used to avoid unwanted vibrations. In active vibration control, an actuator and sensors are used instead of damping elements to react against unwanted vibration, but require a high-energy consumption. While in semi-active vibration control, it takes the advantage of the best features of both passive and active control systems with low power requirement [1] [2].

Karnopp et al. [3] defined the semi active systems as fully active systems that exclude the external power source. The proposed semi-active damper consisted of a fluid viscous damper combined with a variable orifice on a by-pass pipe containing two valves work independently of damping control. **Wolfe et al.** [4] invented a semi-active controllable shock absorber for reducing the energy transmitted between two bodies. The valve of shock absorber is attached to a control rod that is located down inside the piston rod. The valve changes the flow of fluid between the variable volume chambers of the shock absorber, thus accomplishing a fluid restriction to increase or decrease the damping coefficient. **M. Canale et al.** [5] introduced the design and the analysis of a control strategy, for semi-active suspensions in road vehicles, based on Model Predictive Control (MPC) techniques. The result showed that the predictive technique decreases significantly the maximal peak accelerations of the sprung mass leading to a more comfortable behavior than skyhook (on-off) strategy. **Khajavi and Abdollahi** [6] compared the passive and semi-active suspension with proposed fuzzy logic controller for a quarter car model. It was seen from the simulation results of semi-active system based on fuzzy logic could improve the ride comfort and handling characteristics over a passive suspension system. **Praveen Kumar et al** [7] studied the semi-active variable magnetorheological (MR) dampers to mitigate seismic response and vibration control in piping system used in the process industries.The analytical results demonstrated that the semi-active dampers are very effective and practically implementable for the seismic response mitigation, vibration control and seismic requalification of piping system. **William C. and Alberto L.** [8] studied the control techniques for the problem of vibration in mechanical structures using artificial neural networks based on its inverse model. The control technique was tested through the simulation of the active vibration control of a mass-spring-damper model with three degrees of freedom. The results obtained showed that the control with neural networks controller is effective for vibration suppression as compare with H∞ control technique. **Mahmut Paksoy et al.** [9] studied intelligent controllers for vibration reduction of a vehicle by a semi-active suspension system with a variable damper (MR-type). The simulation results showed that both Fuzzy Logic and Self-Tuning Fuzzy Logic controllers perform better compared to uncontrolled case. **Dubay R. et al.** [10] introduced a unique approach for active vibration control of a one-link flexible manipulator. The method combines a finite element model of the manipulator and an advanced model predictive controller (MPC) to suppress vibration at its tip. The experimental and simulation results demonstrated that the finite element based predictive controller provides improved active vibration suppression in comparison with a standard predictive control strategy.

In this paper, the vibration control of a single degree of freedom system by using a semi-active tuned damper will be studied, which starting by developing the passive damper into a tuned damper based on variable piston area. The controller will be design theoretically and used in the experimental work to control the system vibration.

2.Theoretical analysis

The used vibrating system consists of a beam (AB) which freely pivoted in ball bearings at the left end and considered sensible rigid as shown in figures (1) & (2). A spring of stiffness K attached to the beam at point E. While the damper with variable damping coefficient Cs attached to the beam at point D. A motor with out of balance disk attached to the beam at point F, which used as vibration exciter.

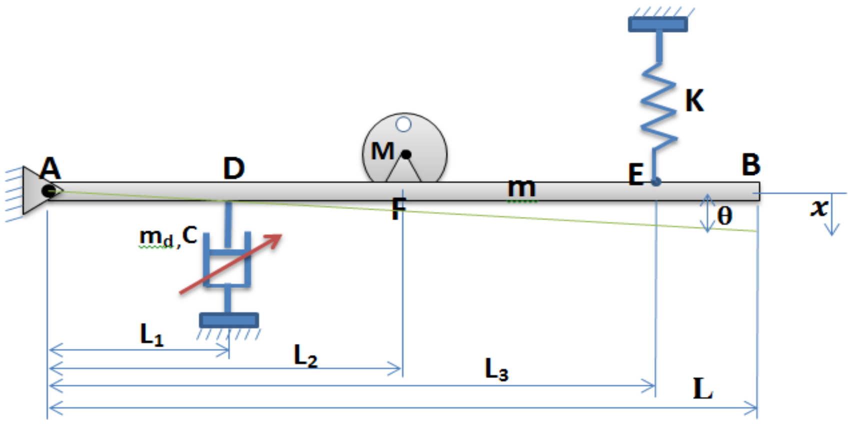


Figure (1): Schematic diagram of dynamic system

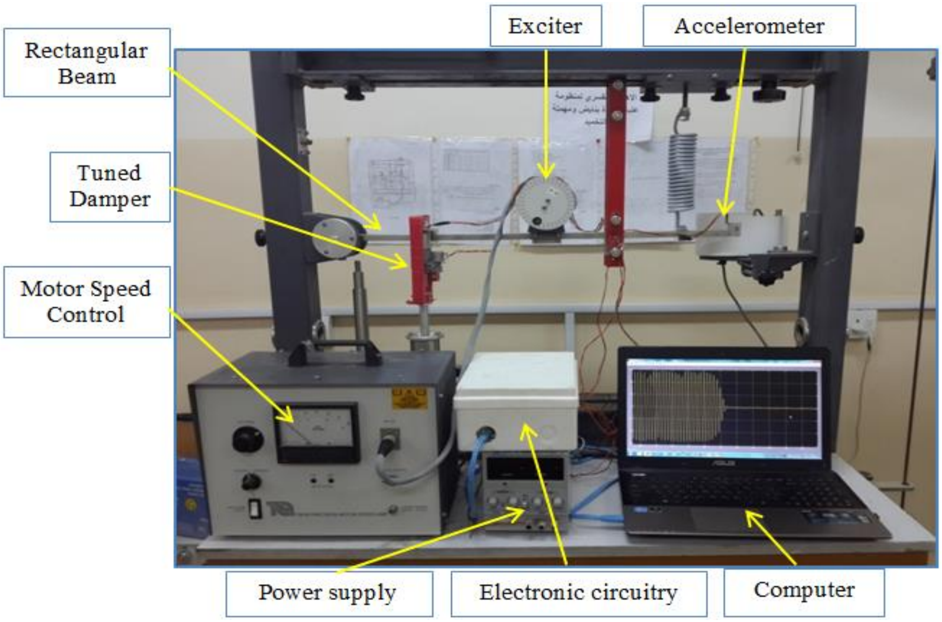


Figure (2): Main components of test rig

Establishment of the equation of motion involves forming the moment equilibrium about the fulcrum point A of the beam:

$$\sum M_A = I_A \frac{d^2\theta}{dt^2} \tag{1}$$

$$(F_0 \sin\omega t)\,L_2 - (K\,L_3\theta)\,L_3 - \left(C_s\,L_1\frac{d\theta}{dt}\right)L_1 = I_A\frac{d^2\theta}{dt^2} \tag{2}$$

Where:

$$I_A = \frac{m\,L^2}{3} + ML_2^2 + m_d\,L_1^2 \tag{3}$$

Re-arrange equation (2), we obtain:

$$\frac{d^2\theta}{dt^2} + \frac{C_s\,L_1^2}{I_A}\frac{d\theta}{dt} + \frac{K\,L_3^2}{I_A}\,\theta = \frac{F_0\,L_2}{I_A}\sin\omega t \tag{4}$$

When $x = L\,\theta$

$$\frac{d^2x}{dt^2} + \frac{C_s\,L_1^2}{I_A}\frac{dx}{dt} + \frac{K\,L_3^2}{I_A}\,x = \frac{m_u\,r\,\omega^2\,L\,L_2}{I_A}\sin\omega t \tag{5}$$

$$\frac{d^2x}{dt^2} + 2\,\zeta_s\,\omega_n\,\frac{dx}{dt} + {\omega_n}^2\,x = F_1\,\sin\omega t \qquad (6)$$

By comparing eq. (5) with eq. (6)

$$\frac{C_s\,L_1^2}{I_A} = 2\zeta_s\,\omega_n \qquad \& \qquad \frac{K\,L_3^2}{I_A} = {\omega_n}^2 \qquad (7)$$

$$\frac{d^2x}{dt^2} + 2\zeta_s\sqrt{\frac{K\,L_3^2}{I_A}}\,\frac{dx}{dt} + \frac{K\,L_3^2}{I_A}\,x = \frac{m_u\,r\,\omega^2\,L\,L_2}{I_A}\,\sin\omega t \qquad (8)$$

Or

$$\frac{d^2x}{dt^2} + 2\zeta_s\,\omega_n\,\frac{dx}{dt} + {\omega_n}^2\,x = \frac{m_u\,r\,\omega^2\,L\,L_2}{I_A}\,\sin\omega t \qquad (9)$$

The logarithmic decrement is calculated as the ratio of two consecutive peak acceleration that measured by accelerometer [11].

$$\delta = \ln\frac{\ddot{x}_x}{\ddot{x}_y} = \zeta_s\,\omega_n T_D \qquad (10)$$

$$\delta = \frac{2\Pi\,\zeta_s}{\sqrt{1-{\zeta_s}^2}} \qquad (11)$$

3. MATLAB/SIMULINK Model for Vibrating System

In order to study the response of the dynamic system, the system model must be solved. MATLAB/SIMULINK was used to solve the derived model of vibrating system. Rewriting the differential equation of motion (equation (9)) in term of acceleration:

$$\frac{d^2x}{dt^2} = +\frac{m_u\,r\,\omega^2\,L\,L_2}{I_A}\,\sin\omega t - 2\zeta_S\,\omega_n\,\frac{dx}{dt} - {\omega_n}^2\,x \qquad (12)$$

The terms in right-hand side of equation (12) are added in sum block to provide the acceleration at the end of the beam, the integrators generate velocity, and displacement and the feedback terms provide the inputs to the sum block and so on. The SIMULINK program of system is shown in figure (3).

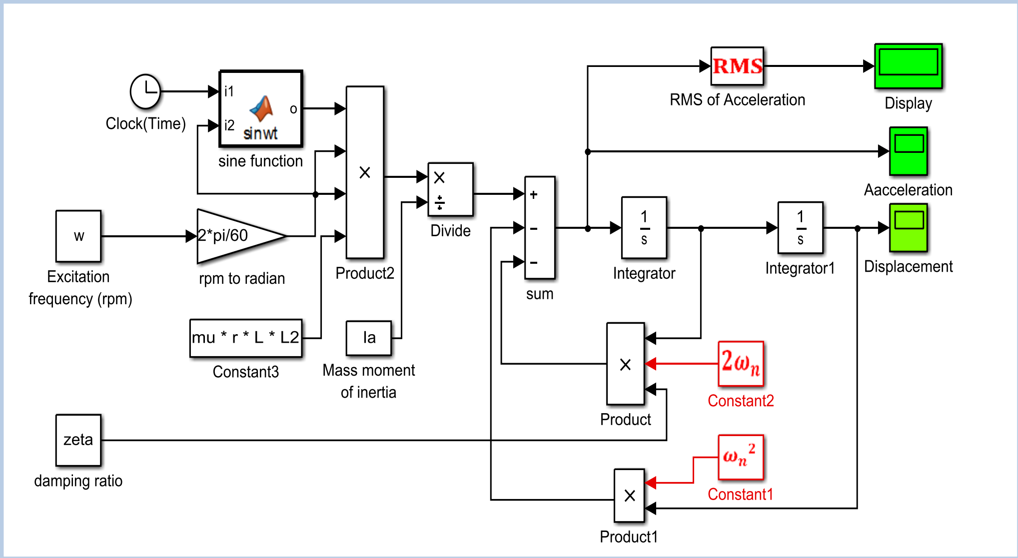


Figure (3): MATLAB/SIMULINK for vibrating system

4. Experimental Work

The test rig has a beam with mass of (2.2325) kg and dimensions (75*2.5*1.25) cm. The mass of the motor, unbalance disc and tuned damper are 2.435, 0.388 and 0.888 kg respectively. The experimental work started by developing the passive damper into a tuned damper. In this work, the changing of piston area is the method of controlling the damping characteristics. The main piston body consists of a radial array of three equally size holes with a circular arc shaped and arranged with equal interval along the circle while the rotary disc is like a fan-shape and rotates with respect to main piston as shown in figure (4). The motor is selected to change the disc position relative to the main piston part, thus allowing the effective flow area to be controlled via the DC motor. The damping is varied by changing the angle of rotation, where ϕ = zero corresponds to fully-open (minimum damping) and ϕ = 60 degrees corresponds to fully-closed (maximum damping). The main parts of the damper were made from plastic material (Acrylic) and aluminum. The CNC machine is used for cutting the Acrylic material and the developed tuned damper is shown in figure (5).

To control the system output, the electronic components are required. The main components are the NI Data Acquisition (USB-6009) for interface with computer, Accelerometer (type ADXL335) for measuring the acceleration at the beam end, potentiometer for angular position of damper disc and the motor driver to drive the geared DC motor. The overall system connection is shown in figure (6).

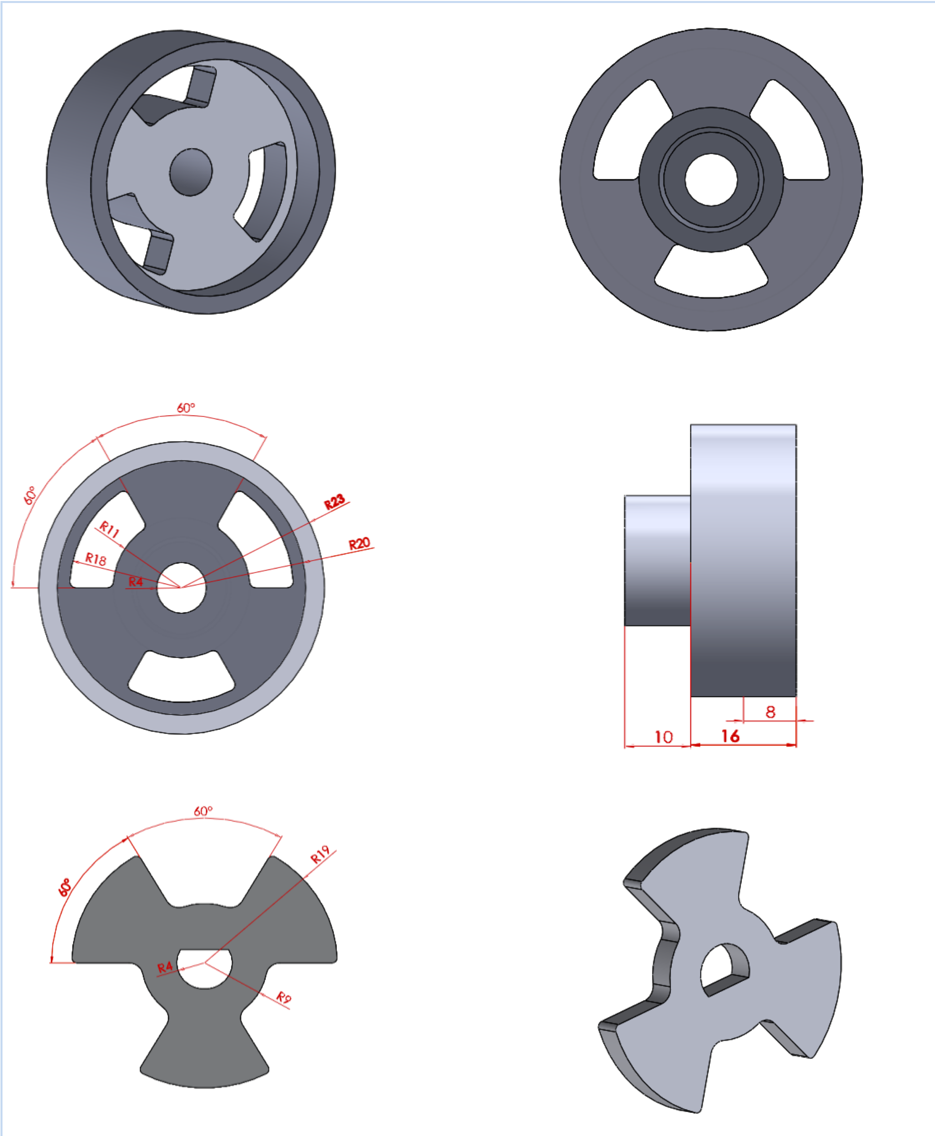


Figure (4): Main piston and rotary disc (all dimensions are in mm)

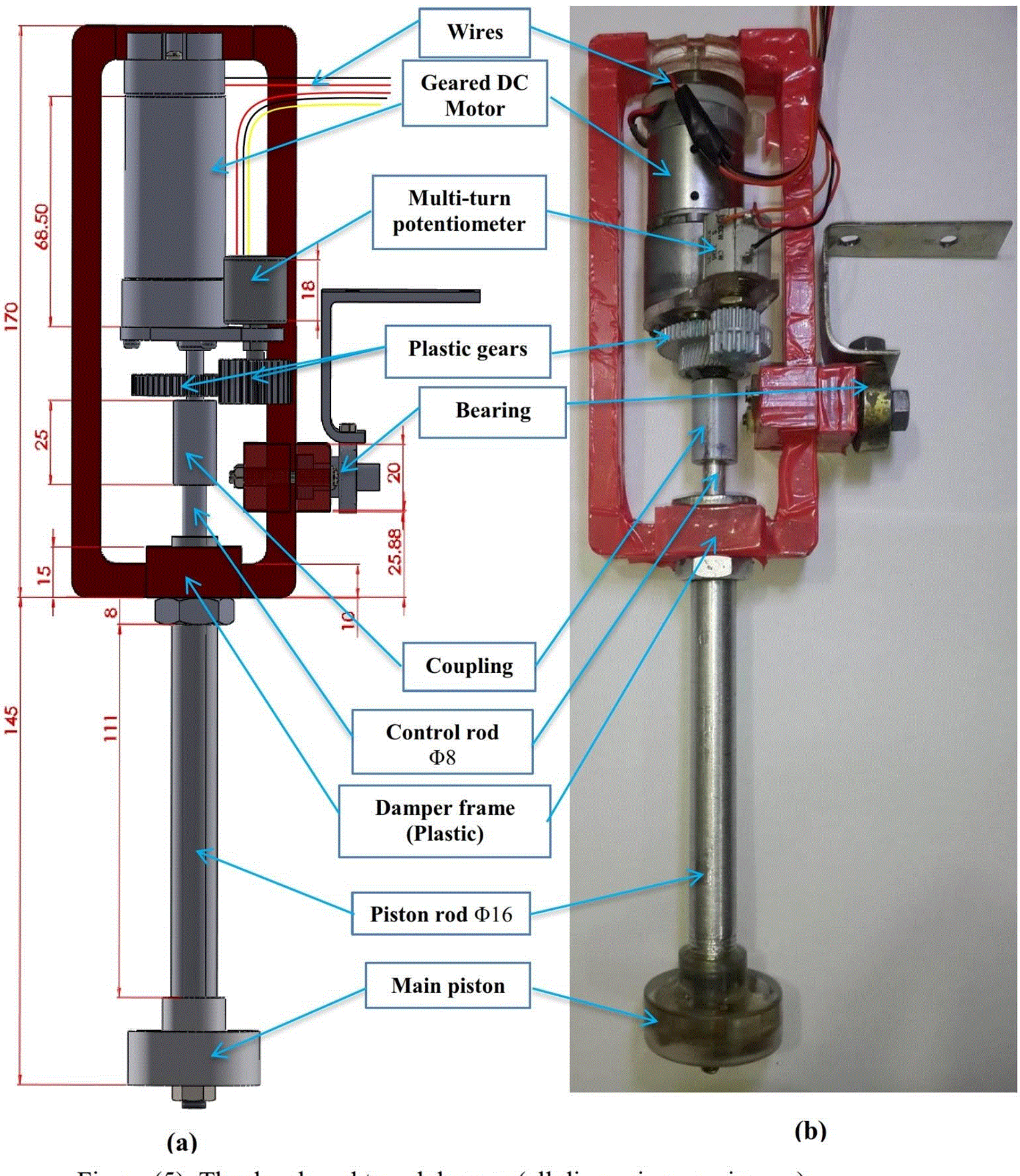


Figure (5): The developed tuned damper (all dimensions are in mm)

a) SolidWork drawing

b) Photo for the implemented tuned damper

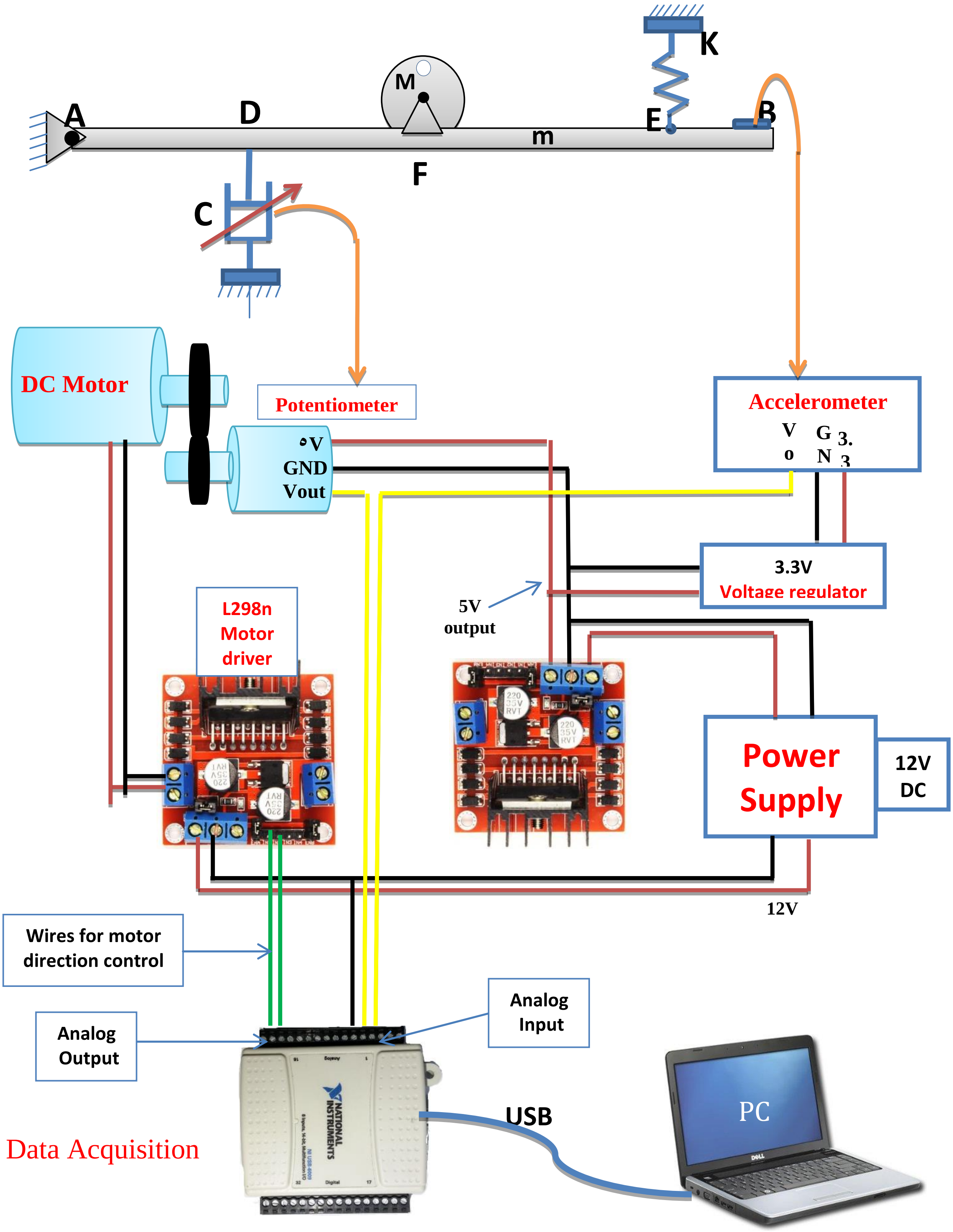


Figure (6): Overall system connection

5. Controller Design

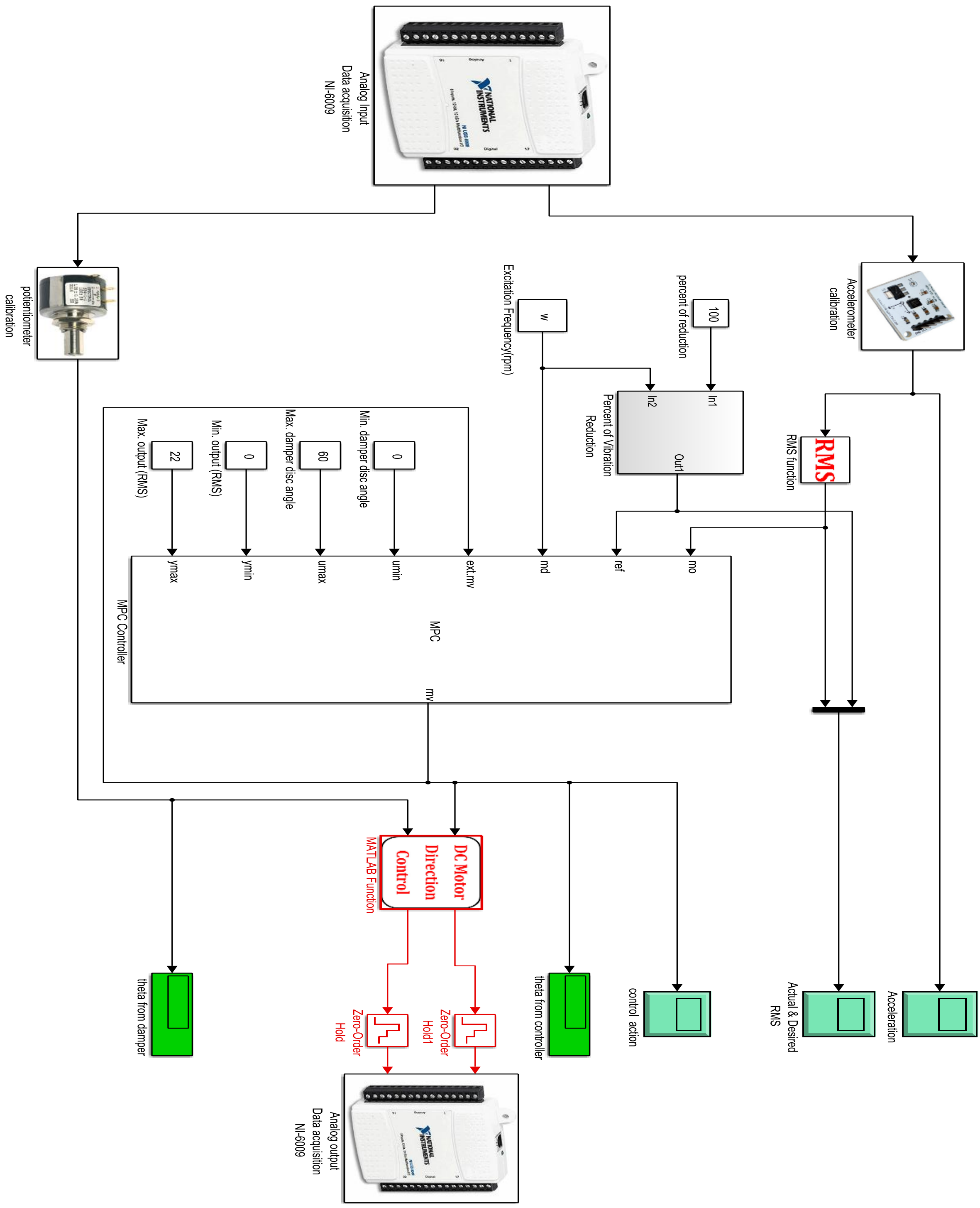
The design of a control system requires a mathematical model of the dynamics of the process, and the model of a dynamic system is difficult to be obtained due to the complexity of the process, because the modeling of variable damping element is very complex. The artificial neural network (ANN) is utilized to identify the dynamic systems, the ANN architecture used here has two inputs (the angular position of the damper disc and the speed of unbalance disc) and one output (RMS value of acceleration). In this work, the Model Predictive controller (MPC) is used because the known disturbance can be taken explicitly into account in predictive control [12]. The model predictive controller toolbox in MATLAB/SIMULINK is utilized in controller design. The controller is designed theoretically based on ANN model, then applied to the experimental system. The designed controller with experimental system is shown in figure (7).

6. Results and Discussion

Before designing a controller, the inputs-output data are recorded from experiments. The RMS value of the obtained acceleration from the experimental work as a function of two inputs which are the damper disc angle and the excitation frequency is plotted and presented as shown in the figure (8). It can be seen from this figure that the largest root mean square (RMS) of acceleration for the system is about (22 m/s^2) and occurred when both of excitation frequency reaches the resonance (9.167 Hz) and the damping force is minimum .The minimum damping force occurs when the damper disc is fully opened (the disc angle = 0 deg.). As the angle of damper disc is increased, the damping force increased. So that, the RMS of acceleration decreased until reaching about (0.4 m/s2) at resonance as shown in the figure (8) , when the damper disc is fully closed. This figure shows that the vibration (root mean square of acceleration) is very affected by damping variation at different excitation frequencies.

The experimental collected data are about 1500 set of inputs and output. These data are divided into training (80%), validation (10%) and (10%) for testing set. After that, a structure for the neural network is proposed, starting with small number of neurons in the hidden layer and then increasing the neurons until reaching the best performance. The best architecture of the neural network that obtained by training was a one hidden layer of eight neurons with log-sigmoid activation function and one output layer of linear function. The MSE change during each epoch in the same training process is shown in figure (9).

Figure (7): MATLAB/SIMULINK program for experimental System



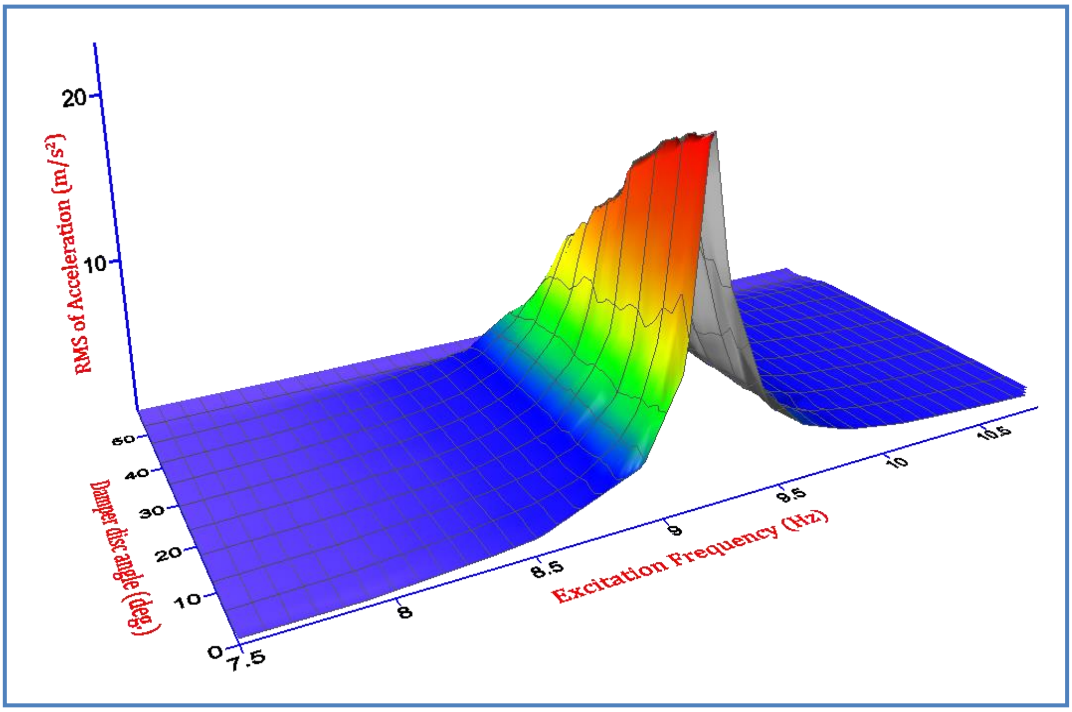


Figure (8): RMS of acceleration versus damper disc angle and excitation frequency

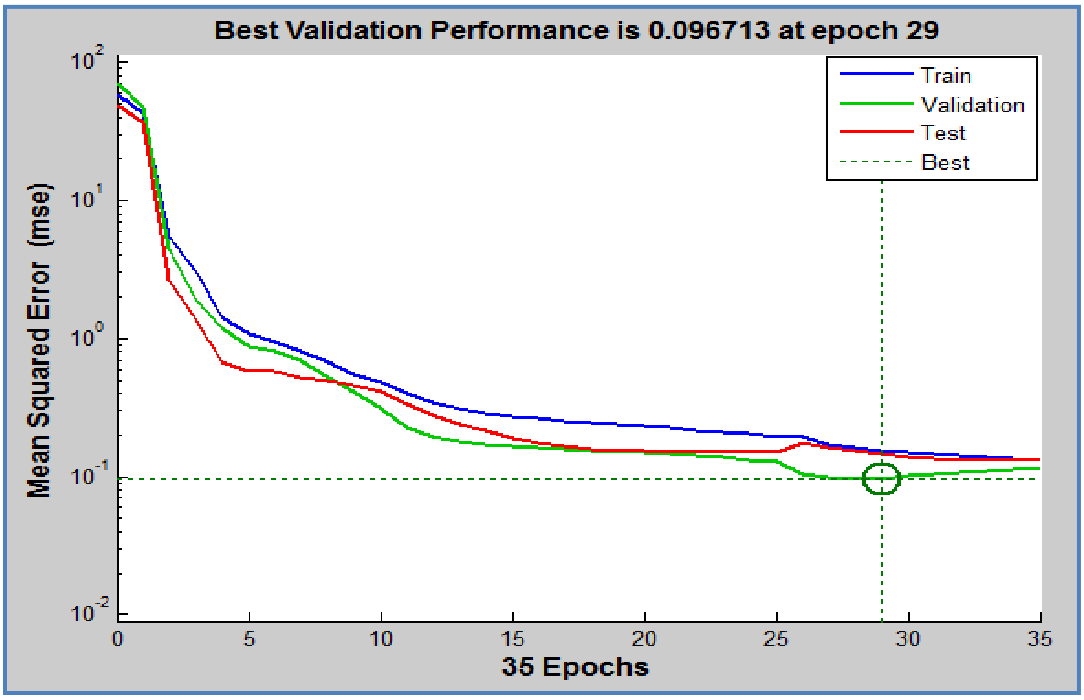


Figure (9): Neural network training performance

The experimental response is plotted at the resonance frequency (9.167 Hz) and at different excitation frequencies near the resonance to show the performance of a control system in this important region. The experimental response (the acceleration and its RMS) at a different percent of vibration reduction is shown in figures (10) to (12). From these figures, there is an oscillation about the desired RMS, which occurs at low damping, this oscillation is due to the following main effects:

- The rotational speed of damper motor during vibration produce turbulent viscous fluid flow through the slot of damper disc, which mainly occurs when the angle of damper disc is widely opened (lower damping force) and the oscillation occurs.
- The time delay occurs in hardware (electronics parts) and software (MATLAB simulation) of implemented component.

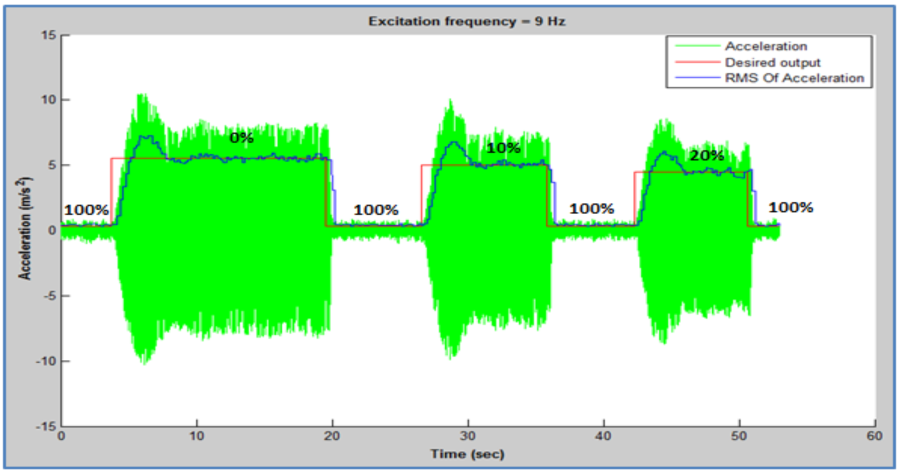


Figure (10): Experimental response acceleration and RMS at excitation frequency 9Hz (540rpm)

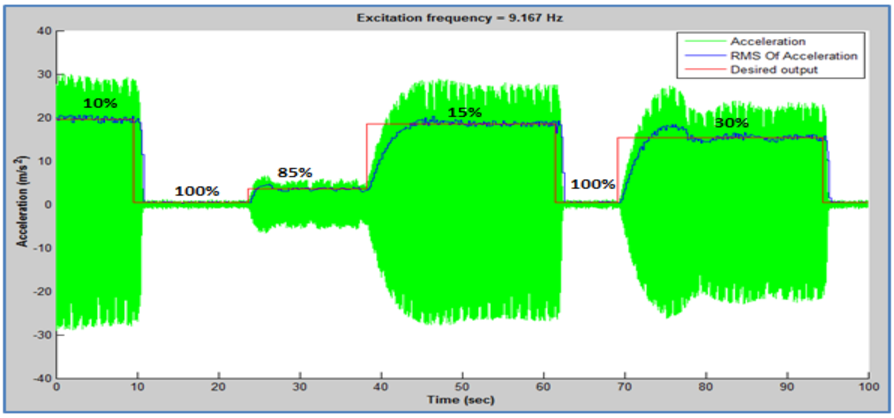


Figure (11): Experimental response (acceleration and its RMS) in time domain at resonance frequency 9.167 Hz (550 rpm)

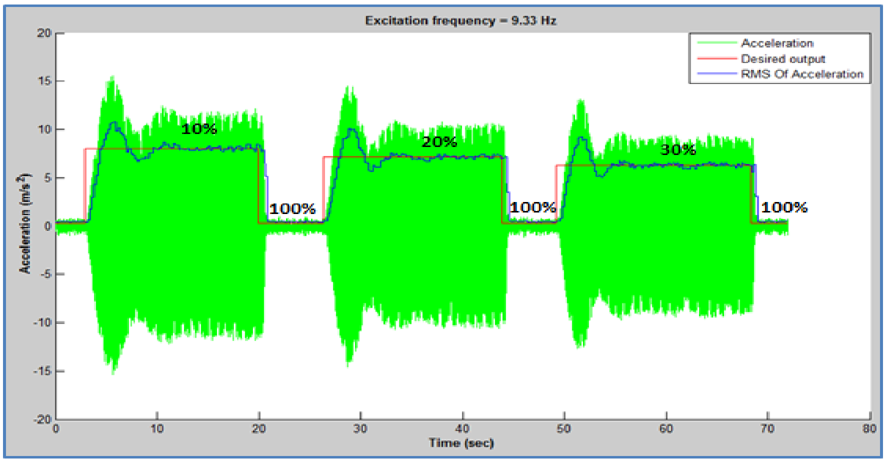


Figure (12): Experimental response (acceleration and its RMS) in time domain when the excitation frequency is 9.33 Hz (560 rpm)

The time required for reducing vibration to a minimum level is very important [13], which is mainly dependent on the performance of the control system. The time of vibration reduction at different frequencies that liberated and magnified from figures (10) to (12) for one reduction at each selected frequency are shown in the figures (13 a , b ,c) in which, the time of vibration reduction from higher RMS value of acceleration to a minimum value depending on the range of reduction.

For verification purposed, the results of the derived model (theoretical) are compared with the experimental results without a controller. By using the logarithmic decrement method based on plotted data of acceleration with the time which is obtained from experimental system shown in figure (14), the damping ratio for a set of damper disc angles can be calculated and the results are presented in the table (1). The maximum damping ratio that can be calculated using the logarithmic decrement is (0.1832) where the angle of damper disc is (52 deg.). Above this angle, the damper leads to a higher damping force, which provides a difficulty in calculating the damping ratio due to the invisible oscillation. A good relation between the damper disc angle and damping ratio using neural network fitting. By this way the inputs of system model are the damper disc angle and the excitation frequency, while the output is the roots mean square of the acceleration. So that, for verification purpose, the experimental and theoretical results are plotted at different angles of damper disc (0, 10,20,30,40 and 50) deg. As shown in figure (15).

Table (1): Data of damper disc angle with damping ratio

Damper disc Angle (deg.)	*Damping Ratio*
0	0.00733
5	0.0077
10	0.0082
14	0.00853
17	0.0096
20	0.01
24	0.01075
27	0.01127
31	0.0136
35	0.0146
40	0.01997
44	0.03208
48	0.0787
50	0.13261
52	0.1832

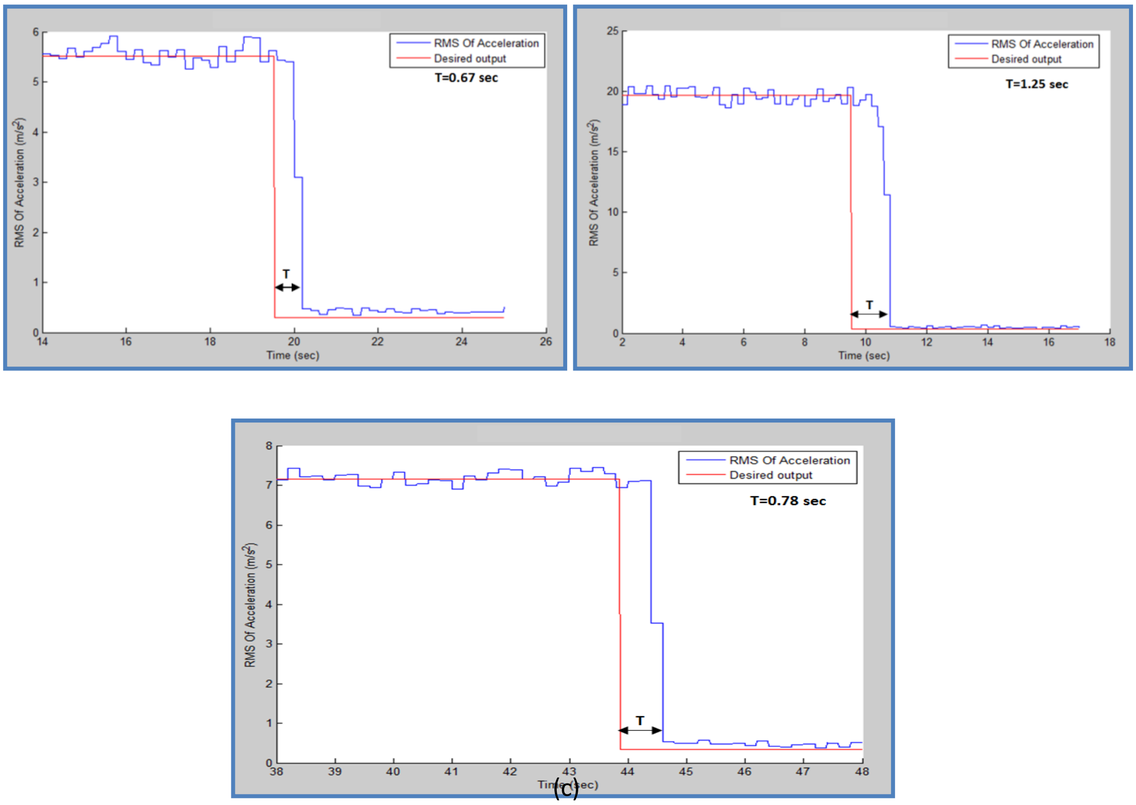
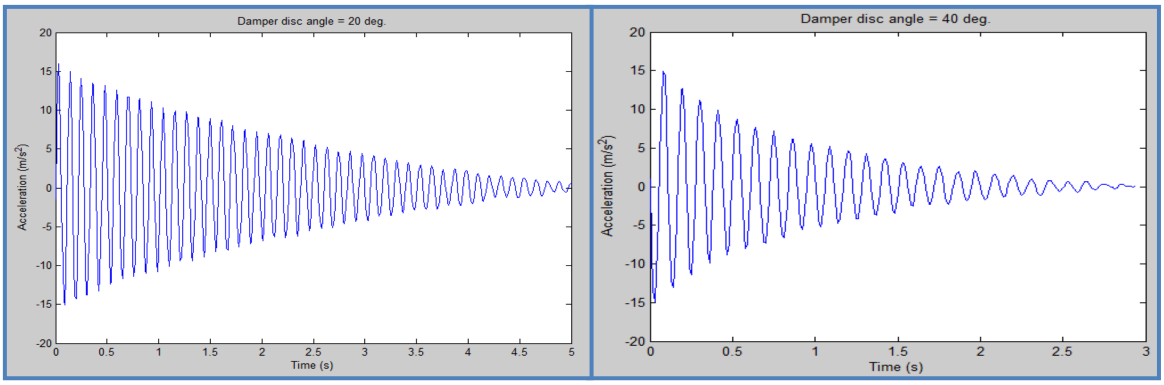


Figure (13): The time of vibration reduction to a minimum level when the excitation frequency:

a) 9 Hz (540 rpm)
b) 9.167 Hz (550 rpm)
c) 9.33 Hz (560 rpm)



Figures (14): The decay in measured acceleration wherethe angles of damper disc (20, 40 egree)

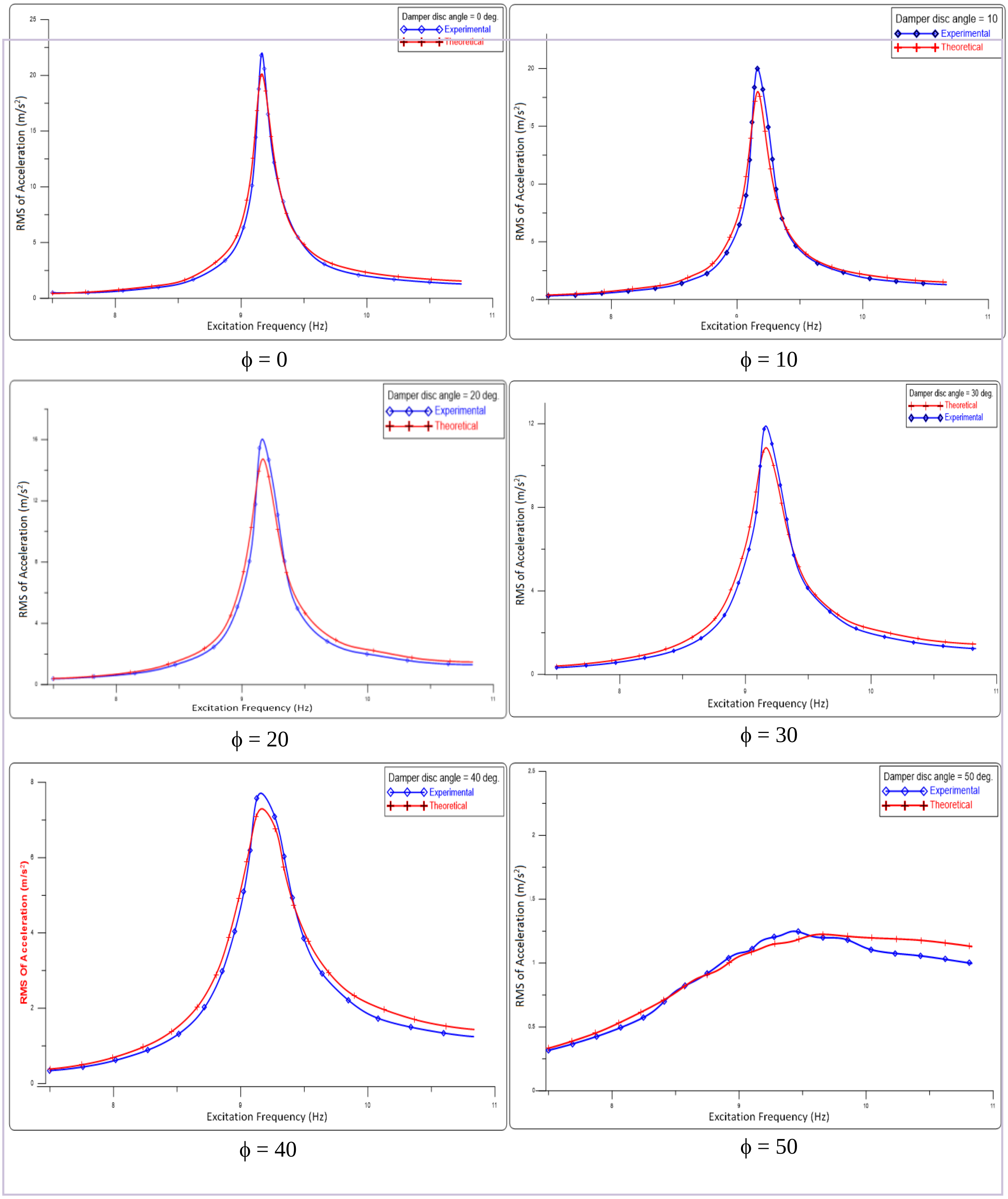


Figure (15): Experimental and Theoretical RMS over a range of excitation frequencies at different angles of damper disc

7. Conclusions

1-The adoption of tuned damper is a good solution to control the system vibration, where the root mean square (RMS) of acceleration of the system can be reduced from (22 m/s^2) to (0.4 m/s^2) at the resonance frequency (9.167 Hz) by varying the angle of tuned damper (0-60) degree.

2- The model predictive controller with proposed semi-active variable damper is significantly reduced the RMS of acceleration at resonance region (peak), in which the maximum time required for reducing vibration of the system from maximum RMS to a minimum level is 1.39 second, which occurs at resonance frequency.

3- The RMS results of experimental and theoretical is relatively converge at different angle of damper disc and the total range of percentage error between experimental and theoretical RMS of acceleration at resonance are (4.9-10) % ,which is acceptable.

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