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Vibration Analysis of a Horizontal Washing Machine, Part IV: Optimal Damped Vibration Absorber

Galal Ali Hassaan
Department of Mechanical Design & Production, Faculty of Engineering,
Cairo University, Egypt

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Abstract:

This is the 4th research paper investigating the vibration of a horizontal washing machine. In this paper a dynamic vibration absorber is attached to the machine drum. The dynamic system is modeled and the dynamic absorber is assumed to have a known mass and damping coefficient. Only the absorber stiffness is tuned using MATLAB optimization toolbox. The drum vibration velocity is used an objective function to attain a value compatible with ISO standard10816. The isolation efficiency is used as a functional constraint to succeed in isolating the large unbalanced rotating force of the laundry during the spinning cycle of the washing machine. The simulation results are amazing. The proposed approach can reduce the vibration velocity to less than 0.7 mm/s RMS and increase the isolation efficiency to greater than 99.7 %.

Keywords — **Horizontal washing machine, vibration control, dynamic vibration absorber, absorber tuning.**

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I. INTRODUCTION

Vibration of washing machines during the spinning cycle is something undesirable since it has high levels usually beyond the levels recommended by international standards. The author investigated the vibration of horizontal washing machines in a series of research papers. In the present paper the author applies a dynamic vibration absorber to limit the vibration velocity of the drum and increase the isolation efficiency of the machine suspension to higher levels to decrease noise generated during the spinning cycle.

Miller (2003) investigated the nonlinear mechanical properties of absorbers for potential application in adaptive-passive tuned vibration absorbers. He used an adaptive-passive tuned vibration absorber capable of varying the absorber resonant frequency in the range 45 to 211 Hz [1]. Liu and Liu (2005) revisited a classical problem about optimum damped vibration absorber. They have applied Brock's approach to different type of damped vibration absorber named model B (when the main system is damped and the absorber damper is connected to the ground) and found its optimal parameters [2]. Thompson (2007) considered a mass-spring absorber system to attenuate structural waves in beams. He investigated the parameters controlling the behaviour of the absorber and developed a formula allowing optimization of its performance [3].

Krenk and Hogsberg (2008) presented a design procedure for tuned mass absorbers mounted on structures with structural damping. They found an accurate explicit approximation for the optimal damping parameter of the absorber and the resulting damping ratio of the response [4]. Najafi, Ashory and Jamshidi (2009) suggested different models for the vibration absorbers to assign the optimal one for SDOF vibration suppression. They used genetic algorithm to optimize the best absorber model [5]. Lin and Coppola (2010) studied the optimal design of the damped dynamic vibration absorber for damped primary systems. They applied two numerical approaches by solving a set of nonlinear equations by the Chebyshev equioscillating theorem and by minimizing a compound objective function subject to a set of constraints [6]. Brown and Singh (2011) formulated a minimax problem to determine the parameters of vibration absorber by minimizing the maximum motion of the primary mass over the exciting frequency domain. They could minimize the main mass displacement magnitude to a value lower than methods available in the literature [7].

Fang, Wang and Wang (2012) investigated a minimax design of damped dynamic vibration absorber for a damped primary system to minimize the vibration amplitude. They illustrated the advantage of their proposed method through numerical simulations [8]. Huang and Lin (2014) designed a dynamic vibration absorber called periodic vibration absorber for mechanical systems subjected to periodic excitation. Their results showed that the periodic vibration absorber could be a very effective device for vibration reduction of mechanical systems subjected to periodic excitation [9].

Kamran, Rezazadeh and Ghaffari (2015) investigated reducing the unwanted vibration in machine tools. They proposed a algorithm to achieve the optimal parameters of the vibration absorber. They evaluated the effectiveness of the proposed algorithm and the designed vibration absorber through comparing the vibration amplitude of the machine tool in the presence and absence of the absorber [10]. Abdelhafiz and Hassaan (2015) investigated using an adaptive tuned vibration absorber to maximize the vibration attenuation of the main vibrating system. Their tuning condition was used to track the exciting frequency. They showed that the frequency response of the main system could be reduced by 10 % [11].

II. THE WASHING MACHINE PHYSICAL MODEL

A physical model of the horizontal washing machine is shown in Fig.1 [12], [13]:

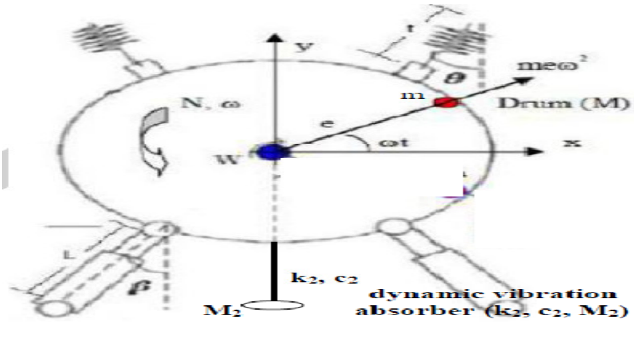

Fig.1 Physical model of the horizontal washing machine [12], [13].

The drum mass is M, the laundry mass is m and its eccentricity id e. The suspension stiffness is k , damping coefficient is c , spring inclination is Θ and damper inclination is β with the vertical axis. The rotor speed is N (rev/min) and ω (rad/s).

A dynamic vibration absorber consisting of a cantilever of stiffness k_2, damping coefficient c_2 and a lumped mass of mass M_2 is attached to the drum at its bottom as shown in Fig.1.

III. VIBRATING SYSTEM MATHEMATICAL MODEL

The mathematical model of the system is constructed as follows:

1. The washing machine drum is the main vibrating system of motion x (assumed as a SDOF system [14]). It has a mass M and its suspension of parameters k and c.
2. The absorber of lumped mass M_2 and stiffness k_2 is attached to the drum. Its mass vibrates with a dynamic motion x_2.
3. The system is a two degree of freedom one excited by the inertia force of the laundry ($me\omega^2$).
4. It is assumed that the suspension and absorber parameters have linear characteristics.
5. The mathematical model is derived by drawing the free body diagram of each mass and applying Newton's second law of motion. This step yield two differential equations in x and x_2.
6. The two differential equations are written in matrix form as follows:

$$\mathbf{M}x'' + \mathbf{C}x' + \mathbf{K}x = \mathbf{F}_0\, e^{j\omega t} \qquad (1)$$

Where: $\mathbf{M}$ = mass matrix = $\begin{bmatrix} M & 0 \\ 0 & M_2 \end{bmatrix}$

$\mathbf{C}$ = damping matrix = $\begin{bmatrix} c'+c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}$

$\mathbf{K}$ = stiffness matrix = $\begin{bmatrix} k'+k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}$

$\mathbf{X}$ = peak amplitude vector = $\begin{bmatrix} X \\ X_2 \end{bmatrix}$

$\mathbf{F}_0$ = exciting force vector amplitude

$= \begin{bmatrix} me\omega^2 \\ 0 \end{bmatrix}$

The main system has an equivalent stiffness, k' and damping coefficient, c' given by [12]:

$k' = 2k(\sin\theta)^2$

and $c' = 2c(\sin\beta)^2$

The steady state solution of Eq.1 is [15]:

$$x = \underline{\mathbf{X}}\, e^{j\omega t} \qquad (2)$$

where $\underline{\mathbf{X}}$ is the amplitude phasor (complex value). Combining Eqs 1 and 2 gives:

$$(\mathbf{K} - \omega^2\mathbf{M} + j\omega\mathbf{C})\underline{\mathbf{X}} = \mathbf{F}_0 \qquad (3)$$

Eq.3 give the phasor of the vibration amplitudes as:

$$\underline{\mathbf{X}} = (\mathbf{K} - \omega^2\mathbf{M} + j\omega\mathbf{C})^{-1}\, \mathbf{F}_0 \qquad (4)$$

According to Miller, Eq.4 fives the vibration amplitude of the main mass, X as [1]:

$$X = me\omega^2 \sqrt{\{N_1 / (D_1 + D_2)\}} \qquad (5)$$

Where:

$N_1 = (k_2 - M_2\, \omega^2)^2 + (c_2\, \omega)^2$

$D_1 = [(k' - M\, \omega^2)(k_2 - M_2\, \omega^2) - k_2 M_2\, \omega^2 - c' c_2 \omega^2]^2$

$D_2 = \omega^2\{[k_1 - (M_1+M_2)\omega^2 + c'(k_2 - M_2\, \omega^2)\}^2$

The vibration velocity of the washing machine has an RMS amplitude given by:

$$V = 0.707\omega X \quad \text{mm/s} \qquad (6)$$

The isolation efficiency of the washing machine isolators, η is [16]:

$$\eta = 100(1 - TR) \qquad (7)$$

Where TR is the transmissibility of the drum vibration given by:

$$TR = F_{t0} / (me\omega^2) \qquad (8)$$

Where $F_{t0} = X \sqrt{\{k'^2 + (\omega c')^2\}}$

IV. OPTIMIZING THE VIBRATION ABSORBER

The dynamic vibration is optimized using the following approach:

1. The number of absorber parameters are reduced from three to parameters to only one parameter.
2. The mass of the absorber is set to 2 kg and its structural damping coefficient is set to 10 Ns/m.
3. The left parameter is the absorber stiffness which can be controlled by the cantilever length , cross-sectional area and material.
4. The algorithm used depends of assigning the absorber stiffness k_2 to minimize an objective function subject to a functional constraint and a stiffness boundaries.
5. The MATLAB command '*fmincon*' is used for this purpose [17].
6. The objective function used is the vibration velocity of the machine drum. It is required to minimize the drum vibration velocity.
7. The functional constraint used is the isolation efficiency. It is required to keep the vibration velocity ≥ 98 %.
8. The vibration absorber stiffness is bounded as: $500 \le k_2 \le 5000$ N/m.
9. This optimization procedure is applied for spinning machine N in the range $400 \le N \le 1200$ rev/min.
10. The results of the application of this optimization procedure are given in Table 1 for a machine drum of 30 kg mass and suspension of 5000 N/m isolator stiffness and 130 Ns/m damping coefficient.

TABLE I
OPTIMIZING THE VIBRATION ABSORBER

Spinning speed (rev/min)	Optimal system performance			
	k_2 (N/m)	X (mm)	V (mm/s RMS)	η (%)
400	1000	0.0223	0.6605	99.6867
500	1000	0.0160	0.5925	99.8356
600	1000	0.0126	0.5609	99.8979
700	1000	0.0105	0.5434	99.9300
800	1000	0.0090	0.5326	99.9488
900	1000	0.0079	0.5254	99.9608
1000	1000	0.0070	0.5203	99.9690
1100	1000	0.0063	0.5167	99.9748
1200	1000	0.0058	0.5139	99.9791

11. The optimal vibration peak amplitude of the drum as function of the machine spinning speed is shown graphically in Fig.2 compared with that without vibration absorber.

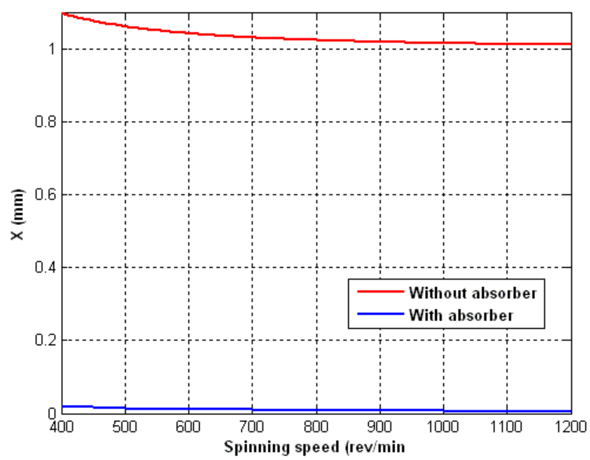

Fig. 1 Optimal vibration amplitude of the washing machine drum.

12. The effect of the spinning speed on the optimal drum vibration velocity in mm/s RMS is shown in Fig.3 for the cases with using a dynamic vibration absorber and without vibration absorber. Good vibration velocity limit is shown according to ISO 10816 [18].

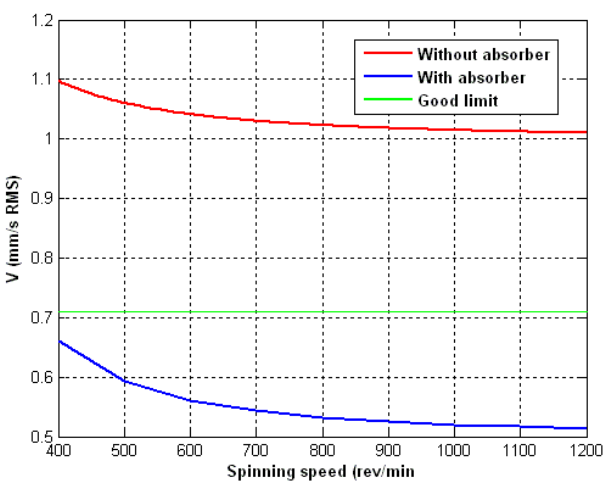

Fig. 3 Optimal vibration velocity of the washing machine drum.

13. The effect of the spinning speed on the optimal isolation efficiency is shown in Fig.4 for the cases with using a dynamic vibration absorber and without vibration absorber. It shows also the desired minimum isolation efficiency.

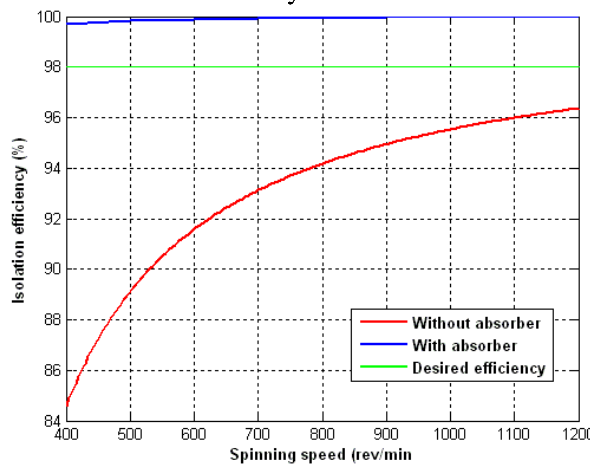

Fig. 4 Optimal isolation efficiency of the washing machine suspension.

V. CONCLUSIONS

- In order to improve the design of the drum-suspension system of a horizontal washing machine, the parameters are optimized for minimum vibration velocity and maximum isolation efficiency.
- The MATLAB toolbox was used for this purpose.
- Only on design parameter was used which is the isolator stiffness.
- The objective function used was the vibration velocity of the drum in mm/s RMS to be compatible with the ISO requirements.
- The functional constraint used was the isolation efficiency such that it has to be above 98 %.
- A value of 1000 N/m isolator stiffness was found reasonable from point of view of vibration velocity and isolation stiffness.
- The effect of the spinning speed of the washing machine was investigated.
- The optimization approach was very effective since it could reduce the vibration amplitude to less than 0.006 mm at 1200 rev/min spinning speed. It could reduce the drum vibration velocity to less than 0.7 mm/s RMS for spinning speeds ≥ 400 rev/min.

- It could increase the isolator efficiency to above 99.68 % for spinning speeds ≥ 400 rev/min.

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BIOGRAPHY

Galal Ali Hassaan

- Emeritus Professor of System Dynamics and Automatic Control.
- Has got his B.Sc. and M.Sc. from Cairo University in 1970 and 1974.
- Has got his Ph.D. in 1979 from Bradford University, UK under the supervision of Late Prof. John Parnaby.
- Now with the Faculty of Engineering, Cairo University, EGYPT.
- Research on Automatic Control, Mechanical Vibrations , Mechanism Synthesis and History of Mechanical Engineering.
- Published 10's of research papers in international journals and conferences.
- Author of books on Experimental Systems Control, Experimental Vibrations and Evolution of Mechanical Engineering.
- Chief Justice of the International Journal of Computer Techniques.
- Member of the Editorial Board of some international journals including IJCT.
- Reviewer in some international journals.
- Scholars interested in the authors publications can visit:

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