The Effect of Damper Configurations on the Vibration of Horizontal Washing Machines

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Abstract. This paper focuses on constructing a Matlab/Simulink program of vibration model with 2 DOF horizontal washing machine. The key point of this paper is using a strong nonlinear friction force - velocity (F-V) relationship of dampers which is obtained from experiment as an input for the simulation. The suitability and high reliability of the model were proved by comparing simulation and experimental results. The model was used to study the effect of damper configurations on the vibration characteristics of horizontal washing machines and then to propose several suitable damper configurations.

Keywords: Horizontal washing machine · HWM · Nonlinear F-V relation
Nonlinear damper · HWM vibration

1 Introduction

There are several approaches in investigating damper configurations of horizontal washing machines (HMW).

Spelta et al. [1] replaced the passive dampers with devices characterized by an electronically-controlled damping ratio and controlled them according to feedback control strategies to reduce the vibrations measured on the cabinet panels. Nguyen et al. [2] proposed an optimally designed magnetorheological (MR) damper for reducing vibration of a front-loaded washing machine where finite element analysis-based optimization tool was used to obtain optimal geometric dimensions of the dampers where they also considered different types of magnetorheological fluid. The optimization problem for the damper was constructed so that the viscous coefficient of the damper is minimized and the yield stress force of the damper is greater than a required value that attenuates almost the resonant peak of the tub mechanism. Buskiewicz and Pittner [3], presented a new switchable electromechanical damper. A dynamic model of the washing unit was developed to simulate the motion of the system after the damping forces are disengaged. The amplification of the tube vibration is not observed. The model could be developed in order to more precisely describe the damping forces, and all the energy losses. Lu and Pang [4] proposed a new damper on the basis of improving the degree of harmonics (DOH) of dynamic impacting force, which is employed as the critical criterion for damper design. Improving the spectra of impact forces in dampers is a feasible and effective scheme to suppress impact excitation on

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the components generating noise, thus the HWM noise can be reduced correspondingly.

Boyraz and Gündüz [5] presented a simple and easy-to-use 2D dynamical model of an HWM for optimization and control of vibration behavior based on Genetic Algorithms (GA). The author defined the objectives of the optimization without considering any linearity or differentiability constraints. YalçJn and Erol [6] developed a semi-active vibration control method to cope with the dynamic stability problem of an HWM. The changes of bracelets radius located on the dampers resulted in the damping properties of the damper in the suspension system, and thus, the semi-active suspension system absorbs unwanted vibrations and contributes to the dynamic stability of the HWM. Argentini et al. [7] introduced 2 designs for the suspension system: a secondary suspension system, designed to filter out the high frequency force components resulted by the dampers and therefore to mitigate panel vibrations; Substitute the existing damper with a tuned mass damper (TMD), directly fixed to the oscillating group. These solutions are a compromise between cost and efficiency, since a linear oil damper would impact excessively on the final cost of the appliance. The TMD proved to be very effective, but it has not been yet tested on a complete washing. Another required improvement for the TMD is the source of damping, that must be achieved at low-cost, and further studies are necessary in this direction.

Tyan et al. [8] developed a multibody dynamic model for a front-loading type washing machine with two magnetorheological (MR) fluid dampers in details. A multibody simulation package RecurDyn and Simulink are utilized to construct this model, which is used for verifying the bearing model between the tub and drum, and for analyzing the suspension system composed of two springs and MR dampers between the case and basket.

Although there are several researches on design and application of dampers to study the vibration of the suspension system, the optimal design of damper was not fully considered. Our study aims to investigate effect of the damper configuration on the vibration of the suspension system. Since the inclination and the number of dampers affect the most to the vibration, they are used in optimization process as primary input parameters.

2 Mathematical Model of the Damper

Dampers used in HWMs are non-Coulomb friction. The piston mounted in a sponge (see Fig. 1) is soaked in special glue to ensure moving resistance. HWMs use friction dampers as a low-cost solution to reduce vibration. For an ideal friction damper, the friction coefficient is constant, but experimental results indicate that it depends on velocity. So, there were some mathematical models of damping friction used in modeling and simulation for a suspension system. The Coulomb friction force is described in the model as $F_D = F_0$ sign(V), in which F_0 is the magnitude of the dry friction force and V is the relative velocity of the piston [9, 11, 12]. Since the damping force is dependent on the relative velocity of the piston, a viscous component is added so that $F_D = F_0$ sign(V) + C.V, in which c is the coefficient of viscous friction [11]. Or for the sake of simplicity, many researchers used model F_D = C.V, as [10, 5].

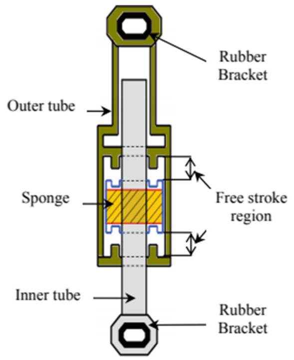


Fig. 1. A structure of a damper

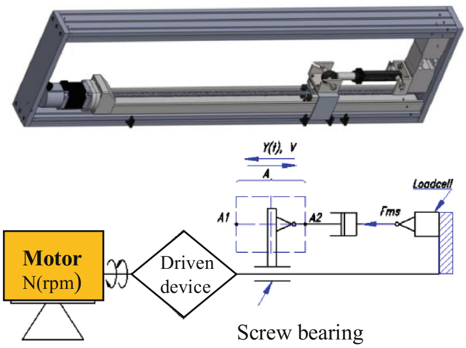


Fig. 2. The damping test system schema

In this study, to determine the characteristics of a damper, a damping test was performed as shown in Fig. 2. Figure 3 shows force-velocity curve according to experimental data. Its nonlinear regression model is shown in Fig. 4. This reflects the F-V relationship of dampers in HWMs.

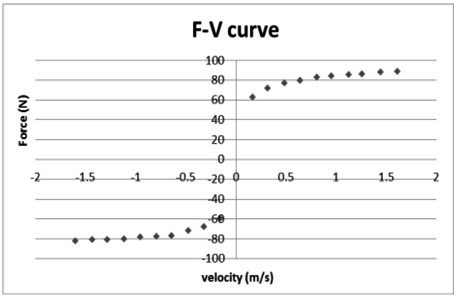


Fig. 3. Experimental F-V curve

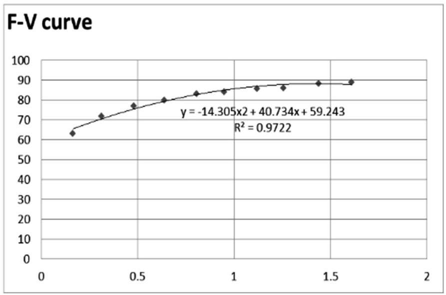


Fig. 4. Nonlinear regression model of F-V experimental data

The nonlinear F-V relation of dampers derived from experimental data shown above is:

$$F_D = C_0 V^2 + C_1 V + C_2 \quad (1)$$

where $C_0 = -14.305$, $C_1 = 40.734$ and $C_2 = 59.243$.

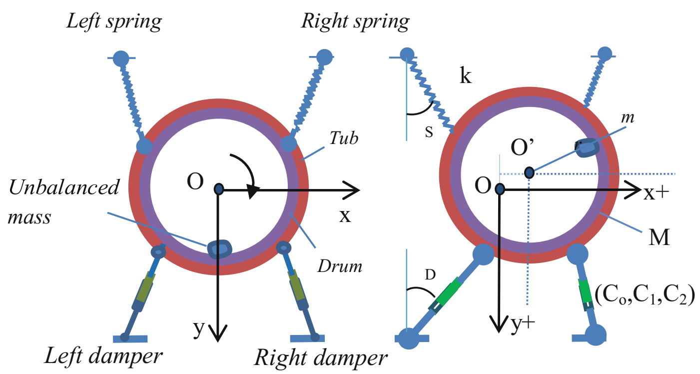


Fig. 5. Physical model of an HWM

Table 1. Nomenclature

Symbol	Definition	Parameters of LG WD 8990TDS
M	Drum total mass (kg)	32.5
m	The unbalanced mass (kg)	0.62
k	Stiffness coefficients of springs (N/m)	5500
θ_s	The inclination of the springs (^{o})	10
θ_D	The inclination of the dampers (^{o})	33
ω	The angular speed of the laundry rotation (rad/s)	

3 Mathematical Model of the Suspension System

The governing equation of a 2D dynamical model of the HWM (Fig. 5) with strong nonlinear damping force as shown in Eq. (1) is presented via Eqs. (2) and (3) (Table 1).

The motion equation can be written as follows:

$$\begin{aligned} &M\ddot{x} + 3[C_0\dot{x}^2 \sin^3\theta_D + C_1\dot{x}\sin^2\theta_D + C_2\sin\theta_D] \\ &\quad - sign(x.y)[C_0\dot{y}^2\sin\theta_D\cos^2\theta_D + C_1\dot{y}\sin\theta_D\cos\theta_D + C_2\sin\theta_D] \\ &\quad + 2kx\sin^2\theta_S = m\omega^2 r\ \cos(\omega t) \end{aligned} \tag{2}$$

$$\begin{aligned} &M\ddot{y} + 3[C_0\dot{y}^2 \cos^3\theta_D + C_1\dot{y}\cos^2\theta_D + C_2\cos\theta_D] \\ &\quad - sign(x.y)[C_0\dot{y}^2\sin^2\theta_D\cos\theta_D + C_1\dot{y}\sin\theta_D\cos\theta_D + C_2\cos\theta_D] \\ &\quad + 2kx\cos^2\theta_s = m\omega^2 r\ \sin(\omega t) \end{aligned} \tag{3}$$

The dynamic models (2) and (3) are solved in Matlab/Simulink environment in the schema as depicted in Fig. 6. Where, "n_Function" is built to generate specific harmonic to simulate the drum's spin speed equations; "Ft" function is used to generate harmonic forces to simulate the centrifugal force caused by unbalanced mass. The "Main function" calculates the dynamic forces of springs and dampers in x, y directions and the dissipated energy function of damper (Ed) and,

$$E_d = \int_{t_0}^{t} [F_{LD}(t)x_{LD}(t) + F_{RD}(t)x_{RD}(t)]dt$$

Where F_{LD}, F_{RD} and x_{LD}, x_{RD} are dynamic forces and displacements of the left and right dampers respectively.

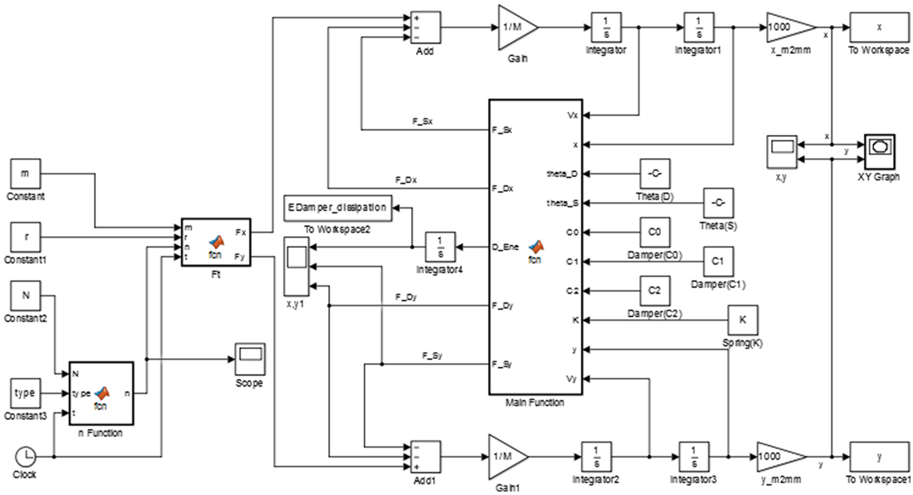
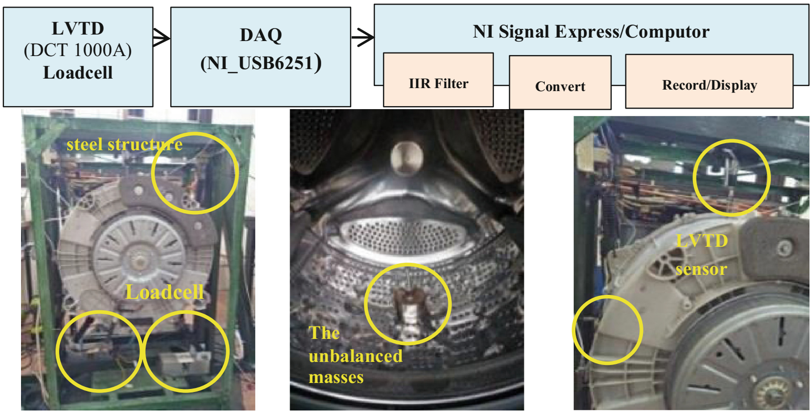


Fig. 6. Simulink diagram for the HWM vibration mathematical model

4 Experimental Validation

Experiments were carried out for determining the vibration of the HWM to validate results obtained from the numerical analysis. Figure 7 shows the experiment on LG WD 8990TDS model with cabinet removal and all the other parts of the machine were installed on a stable steel structure. The unbalanced mass of 620 g was attached into the inside drum as shown in Fig. 7b. Two Linear Variable Differential Transformer (LVTD) sensors were used to measure the horizontal and vertical displacement on the middle of tub (see Fig. 7c); 02 loadcells modelled MT 1041-100 measure dynamic forces exerted in the springs and 03 loadcells modelled MT 1260-50 measure dynamic forces on the dampers (see Fig. 7a); DC LVTD transducers, RDP DCT 1000A models (stroke ±25 mm, output ±5 V, linearity error of full scale ±0.5%) are installed (see Fig. 7c). DAQ, NI-USB 6251, is used for acquisition analog data from the sensors, and these data are then stored and processed in computer. NI-USB 6251 model is a high speed and high resolution DAQ with 16 analog input (16bit), 1.25 ms/s single channel.

Experimental and simulation results for linear and nonlinear frictions are shown in Table 2.



(a) (b) (c)

Fig. 7. The experimental system

Table 2. Simulation and experimental displacement amplitudes at the steady-state spin

	Measurement				Simulation $F_f = c.V$				Simulation $F_f = c_o.V^2 + c_1.V + c_2$			
	600 rpm		800 rpm		600 rpm		800 rpm		600 rpm		800 rpm	
	x	y	x	y	x	y	x	y	x	y	x	y
Amp (mm)	4.61	5.89	4.48	5.36	4.57	4.85	4.58	4.73	5.04	6.45	4.93	5.78
Abs. Err.					0.03	1.04	0.09	0.63	0.53	0.55	0.54	0.41
%Err.					0.7	17.7	2.1	11.7	9.32	9.34	9.87	7.65

Applying the experimental responses of dampers into vibration model of an HWM gives excellent results (errors between simulation and experimental results in x and y directions are less than 10%). The fact that responses of dampers are considered to be linearly proportional with velocity is not really suitable. Thus, the nonlinear damper model should be used instead of linear ones. In this paper, the 2D vibration model using nonlinear dampers is used to evaluate the effect of the inclination of the dampers and number of dampers to vibration of the HWM.

5 Results and Discussion

The main cause of HWMs' vibration is the unbalanced mass forced into the inner side of the tub. It is obvious that the higher the squeeze velocity is, the better the capacity of removing water from clothes is and the greater the vibration energy is. The suspension system should meet 2 requirements: high stiffness and high vibration absorption capacity. In this paper, we introduce a new approach by using Dissipated energy function (E_d) as an important factor to investigate the HWM's vibration. Figure 8 shows the variation of dissipated energy at different angles of the damper, specifically the angle θ_D varies from 10^o to 45^o, corresponding to N = 600 rpm and 800 rpm at spinning state. For example, with N = 600 rpm, at 10^{th} second, E_d = 8.16 J at $\theta_D = 10^o$, 7.38 J at $\theta_D = 33^o$, 8.18 J at $\theta_D = 40^o$ and with N = 800 rpm, at the 10^{th} second, E_d = 7.20 J at $\theta_D = 10^o$, 6.85 J $\theta_D = 33^o$ and 11.6 J at $\theta_D = 40^o$.

Based on the dissipated energy factor, the inclination can be chosen to be small or large. However, Figs. 9 and 10 show that changing the inclination affects the stability of the system rather than the value of dissipated energy. The ramp-up of displacement along x-axis is obtained with any θ_D less than 30^o and larger than 35^o. The orbit of the tub motion (illustrated by the orbit of the center of mass) is no longer stable.

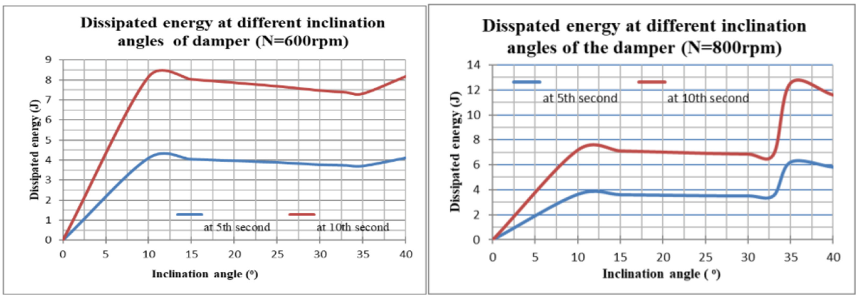


Fig. 8. Dissipated energy vs the inclination of dampers at N = 600 rpm and 800 rpm.

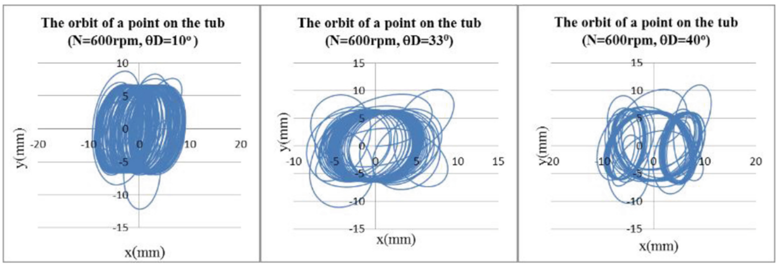


Fig. 9. The orbits of the tub at different inclinations of dampers at N = 600 rpm

The numerical simulation results for 10 s are represented by graphing x and y displacement corresponding to the orbit of the drum oscillation. Each of the orbits is depicted by a curve. The coincidence of these orbits expresses the stability of the system. With $\theta_D = 10^o$, Fig. 9 and Fig. 10 show that the orbit is unstable in x direction while with $\theta_D = 40^o$ the orbit is unstable in both x and y directions.

Thus, to keep the system stable and mild, the inclination of dampers should be chosen from 30^o to 33^o and this is currently used in real-life production.

A more important parameter affecting E_d is the number of dampers in the suspension system. Some damper configurations are taken into consideration as follows (R, L stand for right and left dampers, respectively).

At different rotational speeds, there are some possible configurations as: (1) two dampers (1R1L), (2) three dampers (1R2L), (3) four dampers (2R2L), (4) six dampers (3R3L), six dampers (2R4L) and seven dampers (2R5L). As the modeling results shown in the Fig. 11, the number of dampers is related to the dissipation energy. The more dampers are used, the larger dissipation energy the system has. However, the orbit of tub at different configurations is considered as shown in Figs. 12 and 13, the coincidence of these curves is no longer decided by numerous numbers of dampers. Thus, suitable damper configurations are: 1R1L, 1R2L, 2R2L and 3R3L.

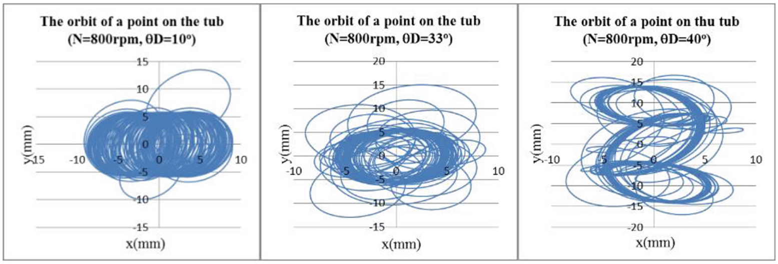


Fig. 10. The orbits of the tub at different inclinations of dampers at N = 800 rpm

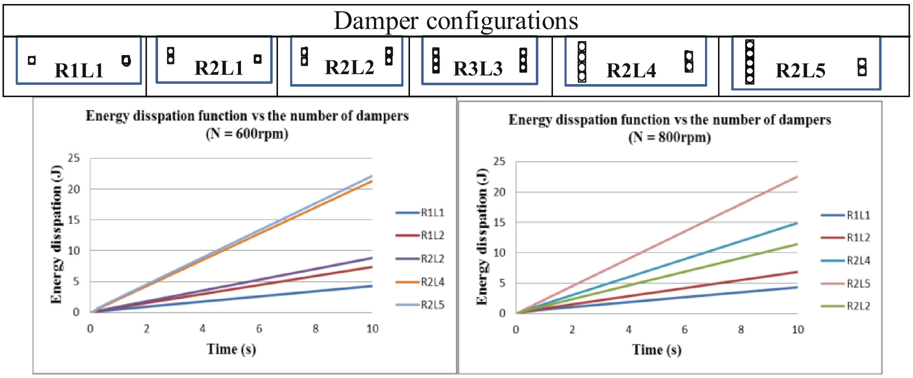


Fig. 11. Dissipated energy vs the number of dampers at N = 600 rpm and 800 rpm

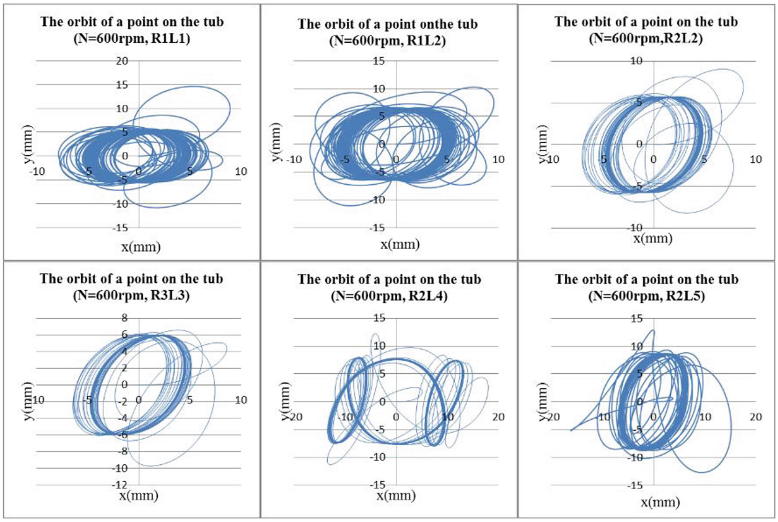


Fig. 12. The orbit of a point on the tub vs number of different dampers at N = 600 rpm

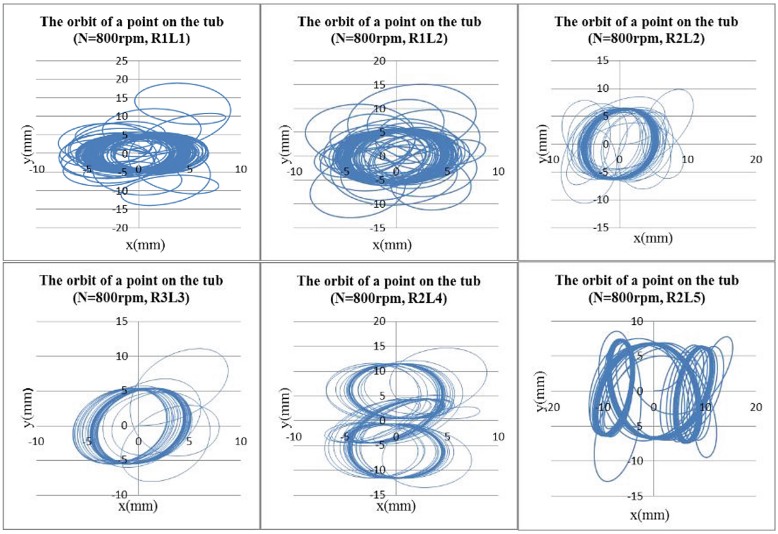


Fig. 13. The orbit of a point on the tub vs number of different dampers at N = 800 rpm

To effectively reduce vibration as well as cost consumption, three dampers (1R2L as in the original design of producer) are more appropriate, in which the position of the single damper on the left or right depends on the tub's rotational direction.

6 Conclusion and Future Work

In this work, a mathematical model of friction force as a function of velocity and the dynamic system of the suspension system with nonlinear friction forces is presented. The model is validated by using simulation program and experimental vibration data collected from an HWM. It then evaluates the effect of damper configurations (inclination and number of dampers) to vibration of the suspension system. The numerical simulation results for different dampers show that the optimization inclination is from 30^o to 35^o and the suitable dampers number are (1) two dampers, (1L1R or vice versa), (2) three dampers (1R2L), (3) four dampers (2R2L), six dampers (3R3L) in which the first option is the best.

Although the calculation error is acceptable, the 2D model is still limited. It will be meaningful to evaluate the effect of eccentric mass at different positions to the vibration of the suspension system and optimize the parameters of the system as well as offer more suitable options to reduce vibration of the suspension. Therefore, the construction of a 3D model for the HWM is really necessary.

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References

1. Spelta, C., Savaresi, S., Fraternale, G., Gaudiano, N.: Vibration reduction in a washing machine via damping control. In: Proceedings of the 17th World Congress, The International Federation of Automatic Control, Seoul, Korea, 6–11 July (2008)
2. Nguyen, Q.H., Nguyen, N.D., Choi, S.B.: Optimal design and performance evaluation of a flow-mode MR damper for front-loaded washing machines. Asia Pac. J. Comput. Eng. **1**, 3 (2014)
3. Buskiewicz, J., Pittner, G.: Reduction in vibration of a washing machine by means of a disengaging damper. Mechatronics **33**, 1–15 (2015)
4. Lu, G.-Q., Pang, D.-M: New damper design for noise control of washing machine, © JVE International LTD, Kaunas, Lithuania. ISSN Print 2335-2124, ISSN Online 2424-4635
5. Boyraz, P., Gündüz, M.: Dynamic modeling of a horizontal washing machine and optimization of vibration characteristics using genetic algorithms. Mechatronics **23**, 581–593 (2013)
6. YalçJn, B.C., Erol, H.: Semiactive vibration control for horizontal axis washing machine. Shock Vibr. **2015**, 10 (2015)
7. Argentini, T., Belloli, M., Robustelli, F.C., Martegani, L., Fraternale, G.: Innovative designs for the suspension system of horizontal axis washing machines: secondary suspensions and tuned mass dampers. In: Proceedings of the ASME 2013 International Mechanical Engineering Congress and Exposition, IMECE 2013, San Diego, California, USA, 15–21 November 2013 (2013)

8. Tyan, F., Chao, C.-T., Tum, S.-H.: Modeling and vibration control of a drum-type washing machine via MR fluid dampers. In: Proceedings of 2009 CACS International Automatic Control Conference, National Taipei University of Technology, Taipei, Taiwan, 27–29 November (2009)
9. Türkay, O.S., Sümer, I.T., Tuğcu, A.K., Kiray, B.: Modeling and experimental assessment of suspension dynamics of a horizontal – axis washing machine. J. Vibr. Acoust. **120**, 534–543 (1998)
10. Lim, H.-T., Jeong, W.-B., Kim, K.-J.: Dynamic modeling and analysis of drum-type washing machine. Int. J. Precis. Eng. Manuf. **11**, 407–417 (2010)
11. Conrad, D.C., Soedel, W.: On the problem of oscillatory walk of automatic washing machines. J. Sound Vibr. **188**, 301–314 (1995)
12. Chen, H.-W., Zhang, Q.-J., Wu, X.-Q.: Stability and dynamic analyses of a horizontal axis washing machine with a ball balancer. Mech. Mach. Theory **87**, 131–149 (2015)