VIBRATION CONTROL OF WASHING MACHINE WITH MAGNETORHEOLOGICAL DAMPERS

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Abstract
Current environmental awareness demands the improvement of washing machine efficiency and increasing of quality of washing processes. Developing of efficient and high speed spinning washing machines raises the problem of designing advanced suspension systems to control vibration and noise in real time. In the paper a horizontal-axis washing machine is under the study. The results of modelling of dynamics of washing machine and designing of smart suspension system comprising magnetorheological dampers are presented. The commercial software MSC.ADAMS/View and MATLAB/Simulink have been used for 3D-modelling of dynamics of washing machine. Both, suspension with passive dampers and suspension with magnetorheological dampers are implemented into the developing mathematical models. Full scale test rig of washing machine was used for validating the models. The models have been used for study of the dynamics of washing machines, designing and optimization of smart suspension system. Analysis of the obtained results has proved a potential of utilization of magnetorheological dampers for suspension systems of washing machines.

Introduction
The development of high speed spinning washing machines is a great challenge. In the water extraction process, the drum starts rotation and this gives rise to significant centrifugal imbalance forces and imbalanced rotation of the laundry mass. This results in vibration and shaking. By elimination of such vibrations it will be possible to design more silent washing machines for higher wash loads within the same housing dimensions.

The components used in a washing machine suspension are springs, dampers and rubber bushings. Damping nowadays is passive in commercially available household machines and dimensioned to handle worst case scenario (resonances) and the washing cycle at once. The fundamental constraint is of course to prevent the drum from hitting the outer housing. This is mainly done with the rubber bushings, but also to some extent with the dampers and springs if they are placed wisely. All these components will however affect the transmission of forces out of the machine and onto the floor.

A whole washing cycle consists of washing, rinsing and spinning. The spinning is less than 5% of the total program time, but is the part that creates the most disturbing vibrations. The dimensioning has however to take into account the whole program cycle. This means that passive components will bring big drawbacks, i.e. most often greater spinning speed lead to higher level of vibrations.

Minimizing the output vibrations from washing machines is an old problem and researchers have worked with different solutions ranging from counterweight balancing (E. Papadopoulos and I. Papadimitriou, 2001) to smart damping. If the desire is to lower the overall vibration as much as possible at every stage of the
washing cycle, then variable damping is the key for suspension improvement. Although some results have been presented, for example by M. J. Charzan and J. D. Carlson, (2001) who implemented smart damping in practice, it has not yet been seen in commercially available machines. In this paper we present results showing the potential of washing machines with magnetorheological (MR) dampers based suspension. We also investigate the effect of different suspension strut mounting angles on the force transmission to the floor and on the drum movement.

Test Setup and Experiments

The aim of the experiments was to measure force transmission onto the floor from the washing machine feet, to observe the magnitude of the drum movement and to identify the damper characteristics.

Equipment used to perform the experiment was: a washing machine with motor controller [type FT46, Asko Cylinda AB], four force sensors [type 201B04, PCB Piezotronics], measuring system [type Pulse 3560B, Brüel & Kjaer], an accelerometer [type 4507, Brüel & Kjaer], and a data storage unit [Standard PC]. The drum was loaded with a small weight simulating the imbalanced clothes. The inner drum was spun using the washing machine's own motor controlled by an external controller. Under each foot a force sensor was placed making it possible to measure how the forces individually changed. This was essential as the machine tilted slightly when passing through resonances. An accelerometer was used to measure the drum’s vertical acceleration. In the performed experiment this signal was also used to extract angular velocity of the drum.

The setting of the present test rig allowed the rotation speed to be varied easily. The imbalance could also be varied by adding steel plates with predetermined weight to the drum. Individual force measurements from the feet together with drum movement served as output from the developed test rig. In Figure 1 an example of force output from the washing machine feet during spin up is shown.

![Figure 1. Force output from the washing machine feet.](image)

A test rig for damper measurements has also been developed. The rig was built around an electric motor with a crank. Position sensors on the crank were used to extract the velocity of the damper piston. The model of the passive friction damper was build using the reaction force $F_d$, and piston velocity $\dot{x}$, according to:

$$F_d = A \arctan(B \dot{x}) + C\dot{x}$$  \hspace{1cm} (1)

The constants $A$, $B$ and $C$ were identified using obtained experimental data together with the Curve Fitting Toolbox in MATLAB. In Figure 2 the measured data is presented together with the fitted model.
The dynamics of the MR fluid sponge damper was proposed by Y. Shen et al., (2005) as a damper with a Bouc-Wen hysteretic model in parallel:

$$ F_{d_{MR}} = c_d \dot{x} + \alpha z $$

(2)

where $c_d(I)$ is the current dependent damper parameter and $\alpha(I)$ is the current dependent parameter of the evolutionary variable $z$ causing a hysteretic behaviour. The Bouc-Wen model's evolutionary variable $z$ is given by

$$ \dot{z} = A \dot{x} - \gamma |z|^{n-1} [z - \beta \dot{x}] z^n $$

(3)

The $c_d(I)$, $\alpha(I)$, $\beta$, $\gamma$, $A$ and $n$ were determined by Y. Shen et al. (2005) as:

$$ c_d(I) = -9.37 I^4 + 10.22 I^3 - 4.33 I^2 + 0.89 I + 0.02 $$

$$ \alpha(I) = 72.8 I^3 - 42.88 I^2 + 14.83 I + 0.29 $$

$$ [\beta, \gamma, A, n] = [-7.0773, 10.6140, 36.2095, 1] $$

(4)

**Modeling**

**MATLAB/Simulink Model**

Each of the four suspension struts can be divided into its core parts namely a spring, a damper and a rubber mounting. The spring’s force $F_s$ was modeled using the following relation

$$ F_s = k \max(0, \delta) $$

(5)

where $k$ is the spring constant and $\delta$ is the deflection from its unloaded length. The $\max(0, \delta)$ comes from the liftoff property of the spring mounting, allowing only one directional force from the spring.

The rubber mounting was modeled as a rotational spring together with a rotational angle velocity proportional damper. The spring causes a negative torque when the strut is bent from its equilibrium angle in the positive direction and the damper causes a negative torque when the strut is moved with a positive angular velocity. The total torque $M'$ was represented as follows:

$$ M' = F' l = -k_\theta \dot{\theta} - b_\theta \dot{\theta} $$

(6)

where $l$ is strut length and $\theta$ is the deflection from equilibrium angle of the strut. The constants $k_\theta$ and $b_\theta$ in (6) were determined with data from experiments performed at ASKO Cylinda AB. The proposed model of rubber mounting is an extension of what was suggested by Türkay et al. (1998).

The imbalance centrifugal force components can be described as
\[ F_y^u = -m_u r \hat{\beta}^2 \sin \beta - m_u r \hat{\beta} \cos \beta, \quad F_z^u = -m_u r \hat{\beta}^2 \cos \beta + m_u r \hat{\beta} \sin \beta \]  

(7)

where \( \beta \) is the angular position of the imbalance \( m_u \) in the drum having radius \( r \).

Using the above relations (1)-(7) the equations describing the translational motion of the center of gravity of the drum and its rotational motion can be written in the following way:

\[
(M + m_u) \ddot{x} = \sum_{k=1}^{4} F_{x_k}^{u_i} + \sum_{k=1}^{4} F_{x_k}^{u_i} + \sum_{k=1}^{4} F_{x_k}^{d_k}
\]

\[
(M + m_u) \ddot{y} = F_{y}^{u} + \sum_{k=1}^{4} F_{y_k}^{u_i} + \sum_{k=1}^{4} F_{y_k}^{u_i} + \sum_{k=1}^{4} F_{y_k}^{d_k}
\]

\[
(M + m_u) \ddot{z} = F_{z}^{u} + \sum_{k=1}^{4} F_{z_k}^{u_i} + \sum_{k=1}^{4} F_{z_k}^{u_i} + \sum_{k=1}^{4} F_{z_k}^{d_k}
\]

(8)

\[
I_x \dddot{\theta}_x - (I_y - I_z) \dddot{\theta}_y \dot{\theta}_x \dddot{\theta}_z = \sum_{k=1}^{4} (F_{x_k}^{u_i} y_k - F_{y_k}^{u_i} z_k) + \sum_{k=1}^{4} (F_{y_k}^{u_i} y_k - F_{y_k}^{u_i} z_k) + \sum_{k=1}^{4} (F_{z_k}^{d_k} y_k - F_{y_k}^{d_k} z_k)
\]

\[
I_y \dddot{\theta}_y - (I_z - I_x) \dddot{\theta}_z \dot{\theta}_y \dddot{\theta}_x = \sum_{k=1}^{4} (F_{x_k}^{u_i} z_k - F_{z_k}^{u_i} x_k) + \sum_{k=1}^{4} (F_{z_k}^{u_i} z_k - F_{z_k}^{u_i} x_k) + \sum_{k=1}^{4} (F_{z_k}^{d_k} z_k - F_{z_k}^{d_k} x_k)
\]

\[
I_z \dddot{\theta}_z - (I_x - I_y) \dddot{\theta}_x \dot{\theta}_z \dddot{\theta}_y = \sum_{k=1}^{4} (F_{y_k}^{u_i} x_k - F_{y_k}^{u_i} y_k) + \sum_{k=1}^{4} (F_{y_k}^{u_i} x_k - F_{y_k}^{u_i} y_k) + \sum_{k=1}^{4} (F_{y_k}^{d_k} x_k - F_{y_k}^{d_k} y_k)
\]

(9)

The equations (8) and (9) were implemented in the MATLAB/Simulink model of the washing machine.

**ADAMS model**

MSC.ADAMS, an advanced tool for simulation of complex mechanical systems, was used for washing machine modeling and dynamic analysis. The ADAMS model was developed using design drawings provided by ASKO Cylinda AB. Hence it can be said that the geometric characteristics of the model are very accurate. The passive damper dynamics (1) has been implemented into the model. Experimentation with developed MATLAB/Simulink and ADAMS models has shown that the ADAMS model will be useful for further optimization of suspension structure when the search space of the design variables has been narrowed down by the MATLAB/Simulink model. The further developed ADAMS model is also believed to have big potential for instance for being able to take into account deformation of structural components. The model of the ASKO Cylinda washing machine developed in ADAMS is shown in Figure 3. Here visualization of some parts has been turned off for clarity.
Results

The developed MATLAB/Simulink and ADAMS models were used for dynamic analysis of peak movement of the drum, transmission of forces and energy absorbed by the dampers of the suspension systems during spinning of the washing machine. The peak movement was determined by the cost functions

\[
J_{NP} = \max(\max(P^k)) - \min(\min(P^k)), k \in [1,2,3,4] \\
J_{NC} = \max(\max(P^k)) - \min(\min(P^k)), k \in [5,6,7,8]
\]

(10)

where the points \(P^k, k = 1,2,\ldots,8\) are chosen as depicted in Figure 4. The following cost functions were used to evaluate the transmission of force and damper energy absorption:

\[
J_f = \frac{1}{N} \sum_{n=1}^{N} |F_z(n)|, \quad J_e = \sum_{n=1}^{N} |F^d_e(n)| \|l(n) - l(n-1)|
\]

(11)

where \(N\) is the total number of samples, \(F_z(n)\) is the sum of forces in vertical direction from all four struts at sample \(n\), \(l(n)\) is the strut length and \(|l(n) - l(n-1)|\) is the distance the piston has moved since sample \(n-1\).

Some results of modeling and dynamic analysis of washing machines with passive suspension and with MR damper suspension are presented in Figure 5 and 6 respectively.

For the passive suspension in addition to variation of strut mounting angle the scaling coefficient \(c_{scaling}\) was used, giving the following relation for damper force \(F^d_e\):

\[
F^d_e = c_{scaling} F^d
\]

(12)

where \(F^d\) is determined by (1). For the active suspension the control current \(I(t)\) was applied from start of spinning of the drum until the resonances were passed. Then the current \(I(t)\) was linearly lowered down to zero for the time corresponding to a speed increase of 100 rpm. Comparative analysis of the simulation results depicted in Figure 5 and 6 shows that approximately the same minimization of drum movement can be achieved using optimized passive suspension or active suspension with MR dampers.
The forces can however be lowered up to 66% using MR dampers compare with transmitted forces during optimized passive suspension performance. For the two suspensions there exists a value for the mounting angle of the strut \( \beta \approx 50^\circ \) which is the most advantageous when the desire is to lower movement in both vertical and horizontal direction.

**Conclusions**

In this paper a theoretical-experimental methodology for dynamic analysis of vibration of washing machines with passive and active suspensions has been presented. The methodology is based on development of MATLAB/Simulink and MSC.ADAMS models of washing machines in parallel. A test rig for horizontal-axis washing machine was built. Validation of obtained results of dynamic analysis of vibrations of the washing machine was done for both developed models by experiments. The proposed methodology also gave the advantage of continuous model-to-model validation during the development process.

Numerical simulations show that strut mounting angle plays an important role in quality of performance of suspension system in washing machines.

The use of magnetorheological fluid based variable damping for suspensions has in this paper been shown promising and hence it is a subject worth further research. The MR damper technology is believed to be good for development of active suspension systems due to its rather easy controllability and also due to expected MR based component availability on the market tomorrow as stated by J. David Carlson (2005). If MR technology ever will be used in washing machines can not be said at this stage as there are still many other aspects to weigh in than spinning which was the focus in this paper. It is however believed that MR technology is one of the most promising ways for active vibration control.

Future work will be focused on extension of washing machine component modeling, on further test rig development and on practical experiments.

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**References**


