MODELING AND OPTIMIZATION OF WASHING MACHINE VIBRATION DYNAMICS

(article starts on next page)

Proc. of The 9th International Conference on Motion and Vibration Control, September 15 – 18, 2008,Technische Universitaet Muenchen.

MODELING AND OPTIMIZATION OF WASHING MACHINE VIBRATION DYNAMICS

Thomas Nygårds, Viktor Berbyuk and Anders Sahlén

Abstract The paper presents a rigid body model of a front loaded washing machine implemented in MSC.Software/ADAMS. The model has been validated against measurement data to such extent that the model could be used as an efficient virtual instrumentation and graphical system design platform for evaluation of existing and developing suspension concepts of washing machines. A new test rig for experimental study of force-frequency-displacement characteristics of dampers together with measurements performed on a friction damper is presented. Estimation of the parameters of a damper model based in the Bouc-Wen hysteresis is performed using optimization routines. A simulation and optimization environment with the dynamic washing machine model in ADAMS and MATLAB optimization toolbox has been developed for use in a computer cluster. The developed virtual instrumentation and graphical design tool has been used for engineering analysis and optimization of a new 3-strut based suspension concept for washing machines. The analysis of the obtained results has shown the engineering feasibility of the 3-strut based suspension and the solution of the optimal placement of the third strut which minimized the transmitted forces during spinning process was found.

1 Introduction

A washing machine is a product common to almost every one in the developed world. Since the beginning of automation of the washing process the dominating

Thomas Nygårds
Applied Mechanics, Chalmers University of Technology, SE-412 96 Göteborg, Sweden,
e-mail: thomas.nygards@chalmers.se

Viktor Berbyuk
Applied Mechanics, Chalmers University of Technology, SE-412 96 Göteborg, Sweden,
e-mail: viktor.berbyuk@chalmers.se

Anders Sahlén
Research & Pre-development, Asko Cylinda AB, Jung SE-534 82 Vara, Sweden,
e-mail: anders.sahlen@asko.se

method of washing has been to let the clothes tumble around in a cylinder. At first the washing machines only did washing, but later the machines also began to be used to dry the load by spinning. Spinning or high speed rotation of the drum gives rise to vibrations, so the suspension system became increasingly important.

Although the method of washing is still the same, i.e. clothes move in a rotating cylinder, two major drum-mounting concepts exist on the market: vertical and horizontal orientation of the axis of rotation. The vertically oriented drum concept implies that the load is put into the machine from the top; hence a machine using this concept is given the name top loading washing machine. The second concept, i.e. horizontally mounted drum, does not itself imply the point of loading; however front loading is most common.

There is a drive within the industry towards production adaptability and cost reduction. Many manufacturers have gone from the traditional metal tub concept to tubs of polymeric material. Among the cost benefit the production method of tubs in such material gives better possibility for the selection of suspension mounting positions and orientations. This paper gives some insight to how the existing suspension concept can be changed slightly to take a step towards a new suspension system concept for washing machines.

Washing machines have been modeled before, for example in [1] where the dynamics of a portable washing machine was described by the Newton-Euler equations in order to deal with sliding and floor friction problems during the spinning cycle, and in [2] where a model of a washing machine was used to study the suspension system dynamics. The Newton-Euler equations describing the dynamics of a specific full size washing machine were derived by the authors from Chalmers, and were presented in [3]. The ADAMS model of washing machine dynamics, which was previously published in [3,4], has been further developed and validated with experimental data. In present paper the developed model is used both for modeling of vibration dynamics and for optimization of the suspension system of a horizontal mounted drum in a washing machine.

2 ADAMS model of dynamics of front loading washing machine

A front loaded washing machine of the most common type has the drum only suspended by bearings in the rear. The design implies the use a sturdy rear gable to support the high torques and bearing forces which arise during spinning. The nowadays less costly way of accomplishing is to use much material in the back gable. To align the drum horizontally cast iron or high density concrete is added as counterweight on the front. The side effect of the addition of these extra masses is of course the lower vibration as the suspended weight increases.

Different outer tub suspension concepts exist on the market but the dominating is top suspended machines where the tub “hangs” in springs. Dampers are normally mounted at the bottom of the tub. The concept under the study in this paper is used by fewer manufacturers and has one key difference to the majority of other machines. The concept is based on that the tub stands on struts mounted between the bottom of the tub and the bottom plate. Another difference from other machines is that no bellow exists as the load lid is attached to the tub and thus moves during spinning. This makes it possible to use a normal rubber seal instead of the bellow. For the modeler this makes the life easier as a nonlinear part disappears.

In production machines, using the above described concept, the struts are four to the number (see fig. 1). In this paper the possibility removing one of these struts is investigated (see fig 2). Designing a suspension system with only three suspension points gives a statically determined system. This will facilitate measurement of displacement for use in for example load estimation. In the three strut model the sole strut has been placed on the left side. This sole strut is referred to as left center strut (LC strut).

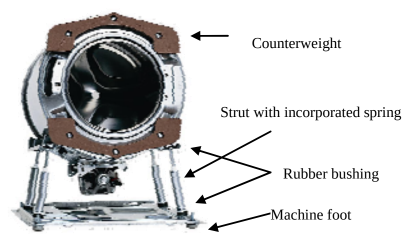

Fig. 1 The four-strut concept

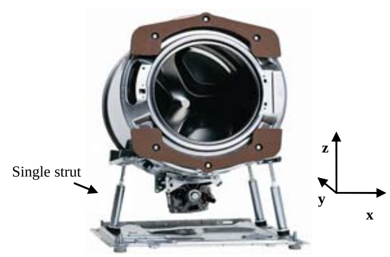

Fig. 2 The three-strut concept

Each strut consists of a friction damper with incorporated spring with attached rubber bushings on each end to provide the stabilizing torque to the tub. The friction damper is a relatively cheap type of damper with a sponge element soaked in oil sliding on the inside of the cylinder. A simple Coulomb-friction-based model of the damper was suggested earlier in [4]. Models of this type are common but in the case [4] it did not describe the frequency and amplitude dependent behavior to satisfying extent. To make the damper component model behave better over the whole washing machine operating range another model is used. The current proposed model is a friction model based on the Bouc-Wen hysteresis model [5] in parallel with a spring and a viscous damper according to equation (1).

$$\begin{cases} F^d = k\max(0, x_0 - x) + c_d\dot{x} + \alpha z \\ \dot{z} = A\dot{x} - \gamma\dot{x}z|z|^{n-1} - \beta\dot{x}|z|^n \end{cases} \tag{1}$$

where k is the spring constant, x_0-x the deflection from the unidirectional spring's unloaded length x_0, c_d is the viscous damper constant, α is the coefficient of the evolutionary variable z. The parameters A, γ, β and n determine the properties of Bouc-Wen hysteresis are explained in more detail for example in [6]. The other suspension parameters like stiffness and damping coefficients of upper and lower rubber bushings have been described earlier in [4].

The twelve degree of freedom dynamic model of the washing machine is built in MSC. Software/ADAMS taking into account CAD drawings giving accurate mass and geometry data for the simulations. Production and research drawings were made available by ASKO Cylinda AB. As many of the model's components are linear, standard components in ADAMS can be used to describe their dynamics. For example linear springs are used for main suspension springs and bushings, for machine feet, etc. The rubber bushings at the ends of the struts are implemented using a Gforce component to describe the torques. The nonlinear damper model differential equation is implemented together with a Sforce component. A complex rotational speed input can be described with the akispline function (Akima-spline) to make it C^1 continuous meanwhile having low overshoots and oscillations. This procedure has shown improving the calculation stability and speed to some extent. Simpler rotational speed schemes like ramps are set as input to the pulley directly.

Parameters which have been made easy to set are: imbalance, position in y-direction of the left side strut, and all the component model parameters. The model input parameters are provided by a file which is loaded into the program at start of simulation. The file contains a total of approx 80 parameters controlling, for example: stiffness and damping parameters for all struts individually, load, simulation time, damper model and more. In the case of the optimization it is this file which carries the parameters uses as input for a function evaluation.

The calculation time - simulation time ratio of the model is about 25 on a computer with a Pentium 4 3.4GHz CPU. This means that the simulation of a spinning process with duration 60 seconds takes about 25 minutes of calculation time.

3 Experimental setup and measurements

At the department of Applied Mechanics at Chalmers a whole machine test rig for washing machines has been developed. This was described in detail in [4]. A few additions to this rig have been made since then. On the control side a program has been developed for control of the motor according to any specified angular speed relation that the motor can handle. The motor electronics is now also used as an additional source for rotational speed of the drum via its built-in tachometer.

A rig made for classification of damper properties has been developed in cooperation between ASKO Cylinda AB and Chalmers University of Technology (see fig 3 and 4). It is based on a standard test rig used for classification of production dampers, but made a bit more rigid and adapted with a frequency converter to cover a frequency range of 0-30 Hz. The possible stroke setting is 0 to 50mm. Force and displacement are measured with a load cell and a LVDT, respectively. Data acquisition is made with a PC equipped with a NI PCI-MIO-16E-4 card which is running LabVIEW 8.5.

In total 144 experiments using 4 different dampers have been conducted. Some of the data which was collected is presented in figure 5. Here the dependency of the friction damper force on frequency and amplitude can be observed. The somewhat extra dynamics of the curve for 15Hz, 4.6 mm which can be seen in the right plot comes from the test rig itself.

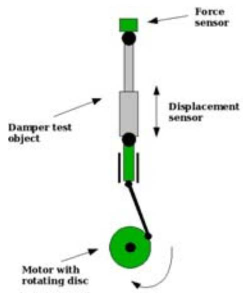

Fig. 3 Functional sketch of damper rig.

Fig. 4 The developed test rig

This oscillation was one of the reasons for an update of the force sensing mechanism which has been performed afterwards. The data used in this paper was however deemed reliable.

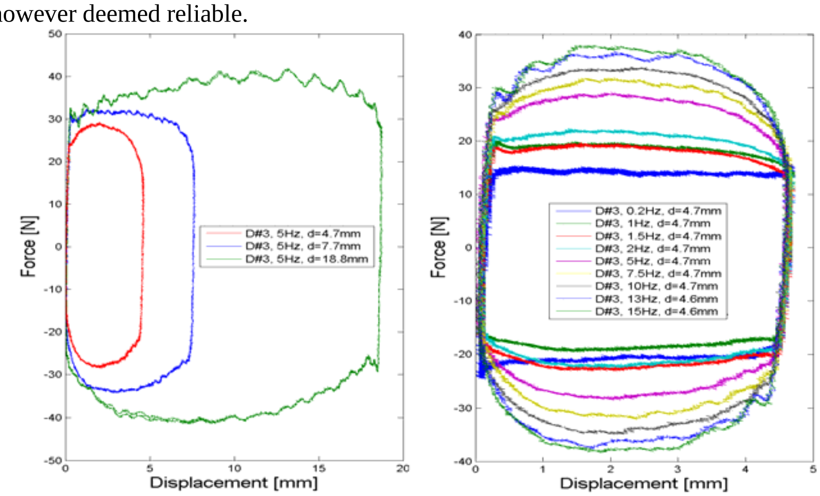

Fig. 5 Force-amplitude and frequency dependencies for one of the friction dampers

Using the obtained experimental data the parameters of the damper model were determined by the optimization toolbox in MATLAB. Two different dampers with different amount of damping were tested in the rig. The datasets for each damper were passed though the optimization algorithm separately but still the hysteresis parameters became relatively similar, (A=3, γ=9.9, β=-2, n=1). These hysteresis parameters generated a relatively high value of parameter $\alpha \approx$100 and a low value of $c_d \approx$0.01 making the washing machine system dynamics unstable at low velocities. To treat this the internal hysteresis parameters were set fixed (*A=30*, γ=3, β= -2, *n*=1) and the optimization was run again. Now only the viscosity coefficient, and the coefficient of the evolutionary variable c_d and α were varied. To reduce the complexity of the future strut optimization problem a polynomial relation was constructed between the optimization result for the parameters c_d and α. The relation was determined with the curve fitting toolbox in MATLAB to equation (2), which together with equation (1) described 3 of the tested dampers well.

$$\alpha = 39.08c_d - 2.955 \tag{2}$$

Now, as only one parameter determines the damper model, not only the optimization problem will become easier, but also the physical realization of a found damper parameter.

4 Simulation and validation of model

The real spinning cycle contains various direction changes and flat parts, i.e. constant speeds. But to make the simulation output as little complex as possible a simple low gradient ramp was used as input for simulation and as input to the motor during measurement. As excitation a 0.3 kg steel bar was mounted to the inside of the drum simulating a fixed load evenly distributed over the depth of the drum.

Figure 6 shows measured forces amplitude over time as a green contour together with time history of simulated forces in blue. The black line shows the rotational speed of the drum. The notation LB stands for left back, RF for right front etc.

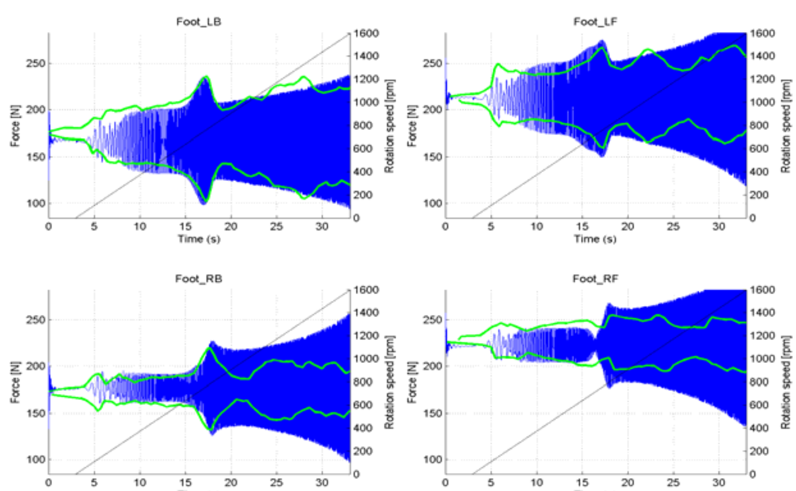

Fig. 6 Measurement data together with simulation data.

It can be seen from the comparison that the resonant behavior between 16s and 19s (600-900 rpm) is captured by the model. During simulations this resonance has been showing high sensitivity to the stiffness of the machine feet. The resonance position in frequency was most sensitive to the stiffness in vertical direction. The shape of the resonance was most sensitive to the relation of the vertical-lateral stiffness. The machine feet damping parameter was the single most important parameter for determination of transmitted forces. A small error in weight distribution of the machine is also present due to lack of modeling of small components inside, e.g. electronics. After 1000 rpm the modeled forces differ from the measured more than acceptable and are therefore a subject for further research.

5 Optimization

One of the intentions of the optimization work of this paper is to show the possibility of reducing the number of struts while keeping the same or less vibration output. To use a methodology of consequent step-by-step optimization to speed up the finding of a solution can be advantageous if the problem is large and complex. This methodology is used to find improvement possibilities for the washing machine suspension system. The structural parameters are put into different groups (see table 1), meaning that during the optimization only the parameters of the particular group will be varied meanwhile the parameters not belonging to the group remain constant.

Table 1 Example of grouping of parameters

	Group 1	Group 2	Group 3	…	Group N
Parameters of third strut to optimize	LC strut y-position	Strut damping Strut stiffness	Lower bushing stiffness Lower bushing damping		LC strut y-position Strut damping Strut stiffness …

5.1 Statement of optimization problem

To prevent the tub and on-mounted components from hitting the housing the displacement of the tub must at no instance of time exceed given constraints on movement. The constraints imposed are defined at 9 points on the tub which ASKO Cylinda AB has determined to be critical in terms of collision risk during spinning or washing. The input rotational speed during the optimization was ramp climbing from 0 to 1200 rpm in 23 seconds, rotating clockwise. Two different load cases were also desired to be evaluated: one case with 0.5kg imbalance in the front end at the periphery of the drum and one case with additional 0.5kg imbalance at the back end at the opposing periphery.

The general optimization problem for the three strut washing machine model can be stated with equation (3) subject to the dynamics of the ADAMS model under the kinematics constraints of equation (4), with the restrictions imposed on the parameters subject to optimization according to equation (5):

$$\Gamma = \sum_{c=1}^{n_C} \sum_{i=1}^{4} \left(\frac{1}{T} \int_0^T \left(F_{zi}^c(t)^2 \right) dt \right) \tag{3}$$

$$\Delta \mathbf{X}_p(t) \le \Delta \mathbf{X}_p^{\max},\ p = 1,2,3...9, \forall t \in [0,T] \tag{4}$$

$$\begin{aligned} &K_{\min}^{lx} \le K^{lx} \le K_{\max}^{lx}, C_{\min}^{lx} \le C^{lx} \le C_{\max}^{lx}, K_{\min}^{ly} \le K^{ly} \le K_{\max}^{ly}, C_{\min}^{ly} \le C^{ly} \le C_{\max}^{ly}, \\ &k_{\min} \le k \le k_{\max}, c_{d\min} \le c_d \le c_{d\max}, 0 \le y_{LC}^* \le 1 \end{aligned} \tag{5}$$

where Γ is the desired function to be minimized under time t ranging from 0 to T for n_C load cases, F_{zi} is the vertical reaction force at foot i, $\Delta\mathbf{X}$ the displacement at point p on the tub, K^{lx},K^{ly}, the lower bushing stiffness, C^{lx},C^{ly}, the lower bushing damping, k the main spring stiffness, c_d the damping parameter of the damper model, and y^*_{LC} the relative position of the left side centre strut. The parameters which were not desired to be optimized in accordance with the described methodology were considered given and set fixed together with the rest of the structural parameters according to equation (6).

$$\xi = \xi_g = \{k_{LF}, k_{LB}, k_{RF}, k_{RB}, c_{dLF}, c_{dLB}, ...\} \tag{6}$$

5.2 Optimization algorithm and results

To solve the optimization problem for the washing machine the algorithm has been developed and implemented in MATLAB with the function “fmincon” as optimization solver. As the calculation of force output and movement of the tub is done in the ADAMS model a communication interface was developed. The communication interface made possible to use an arbitrary number of computers in a cluster for solving parallel function evaluations, for example different load cases or ramps with the same suspension setting. All data from individual simulations are saved on a hard drive for later inspection. Viewing the progress of the optimization as well pausing or stopping the algorithm can be done remotely from an external computer.

The figure 7 shows the cost function (3) as a function of the LC strut position in *y*-direction. From the figure it can be seen that a minima for the vibration output according to (3) can be found for a *y*-position, meanwhile keeping the drum movement within desired limits. Figure 8 describes the negative margin of the peak motion at some of the most interesting points, $\boldsymbol{X}_p$ for different positions of the strut.

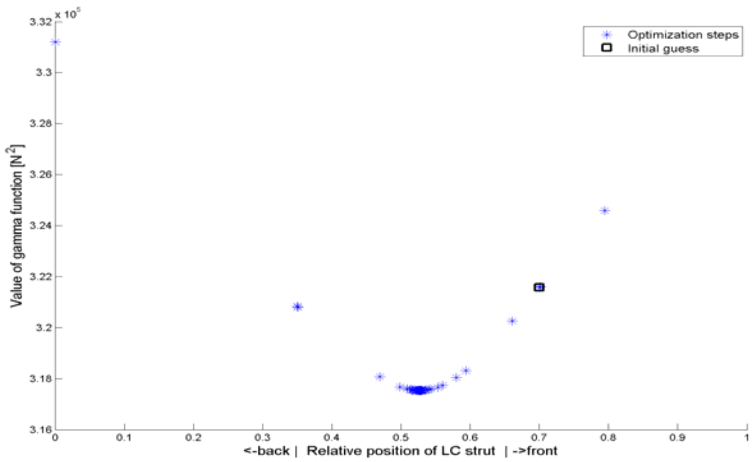

Fig. 7 Result of the optimization

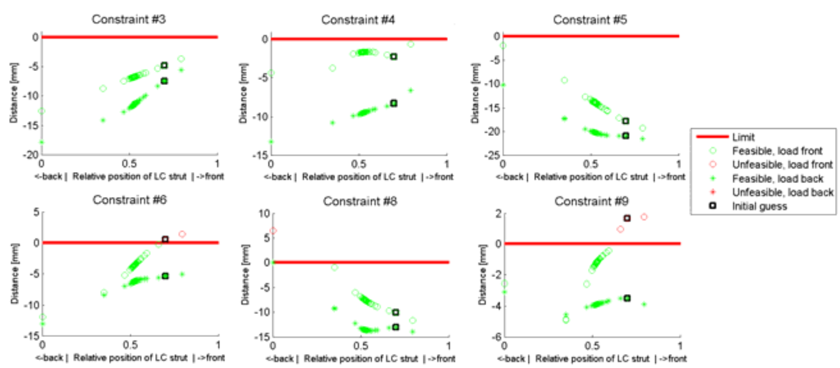

Fig. 8 Constraints values/negative movement margin as function of *y*-position

The result showed sensitivity to the load cases but for the majority of the load case combinations tested the most critical constraint was number 9 which is a point on the front of the tub. It can therefore be concluded that much effort must be put on finding worst-case representative load cases for simulation input to maximize the use of the optimization result.

6 Conclusions

In this paper the developed ADAMS model of washing machine and ADAMS-Matlab interface for clustered simulation and optimization have been described together with experimental setup proving agreement to a satisfying extent.

The frequency and amplitude dependency of the friction damper has been modeled with help of the Bouc-Wen hysteresis model. The parameters of the Bouc-Wen hysteresis part of the damper model were experienced to have big influence on the stability of the whole system. At low piston velocity the stability is a problem and will be a topic for further investigation.

It has been shown that the three strut concept of suspension fulfills the constraints which are imposed on the kinematics of the tub. The optimal placement in y direction of the single strut of the suspension has been found and presented.

Acknowledgments
The authors would like to thank Jan Möller, Research engineer at the department of Applied Mechanics, Chalmers for his help with the test rigs, Peder Bengtsson and Patrik Jansson both at ASKO for their help with simulations and measurements.

References

1. E. Papadopoulos, I. Papadimitriou, 2001: Modeling, Design and Control of a Portable Washing Machine during the Spinning Cycle, Proceedings of the 2001 IEEE/ASME International Conference on Advanced Intelligent Mechatronics Systems (AIM 2001) pp. 899-904
2. O. S. Türkay, et al.: Modeling and Experimental Assessment of Suspension Dynamics of a Horizontal-Axis Washing Machine, ASME, J. of Vibration and Acoustics, 120, 534-543, 1998.
3. T. Nygårds, J. Sandgren; V. Berbyuk, A. Bertilsson: Vibration Control of Washing Machine with Magnetorheological Dampers. Proc. The 8th International Conference on Motion and Vibration Control (MOVIC 2006), August 27-30, 2006, KAIST, Daejeon, Korea, pp. 480-485.
4. T. Nygårds, V. Berbyuk: Dynamics Of Washing Machines: MBS modeling and experimental validation, Proceedings of the Multibody Dynamics 2007, ECCOMAS Thematic Conference, Milano, Italy, 25–28 June 2007.
5. Y. Shen, et al.: Analytical and Experimental Study of the Response of a Suspension system with a Magnetorheological Damper, J. of Intel. Material Systems and Structures, 16, 135-147, 2005.
6. F. Ikhouane, V. Manosa and J. Rodellar: Dynamic properties of the hysteretic Bouc-Wen model, Systems & Control LettersVolume 56, Issue 3, , March 2007, Pages 197-205.