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IN SOLID AND STRUCTURAL MECHANICS

Washing Machine Design Optimization Based on Dynamics Modeling

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ABSTRACT

The spinning process in a washing machine is a source of undesired vibrations and noise and may cause discomfort both to the user as well as for the machine itself due to vibrations which have impact on system lifetime, reliability of operation and capacity.

The aim of this thesis is to develop mathematical and computational models of a washing machine of modern type and use these models for dynamics analysis and in optimization routines to make improvements of the machine mechanical design. In the thesis a multibody system model of a commercial front loaded washing machine is presented and used for analysis of the vibration dynamics and system design optimization. The model has been built using a theoretical-experimental methodology consisting of integration of multibody system (MBS) formalism, detailed modeling of machine functional components and experimental data based validation. The complete model of a washing machine is implemented in the commercial MBS environment Adams/View from MSC.Software. Several test rigs for experiments on components and the complete washing machine have been developed. Validation of the developed computational models has shown acceptable agreement both with tub kinematics as well as forces transmitted to the hosting structure.

In this thesis the vibrations of a washing machine have been channeled into kinematic, dynamic and stability cost functions. The defined kinematic cost function deals with the tub motion and can be used to ensure margins to collision between parts inside the machine or constitute a step in the process if increasing the system capacity. The dynamic cost function measures transmitted vertical forces to the hosting structure, forces which are found cause the most noise and vibration impact on the surroundings. A cost function based on the necessary and sufficient criterion for stability of a washing machine in the sense of walking avoidance is also presented. The introduced cost functions are used for multiobjective optimization of washing machines, and several problems have been formulated and solved by changing suspension geometry and component parameters.

To solve multiobjective optimization problems of a washing machine on a set of spinning operational scenarios a multistep approach has been proposed. The approach reduces the number of variables in the considered generic complex multiobjective constrained optimization problem of a washing machine and makes use of engineering knowledge and other requirements on the system, e.g. such as esthetics.

The utilization of several vibration control technologies for washing machines, such as a semi-active suspension system, automatic balancing, etc are also investigated. The solution comprising a magnetorheological damper, validated by incorporation into a washing machine suspension system, giving force propagation amplitude reduction of up to 40% is presented.

KEYWORDS: Washing machine, Suspensions, Modeling, Experiments, Vibration dynamics, Multiobjective Optimization, Automatic balancing, Semi-active control.

PREFACE

This thesis is the result of a project started in April 2006 at the division of Dynamics, department of Applied Mechanics, Chalmers University of Technology.

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Thanks to the colleagues at the department for interesting and ever lively discussions around the coffee table. Especially warm thanks go to the lunch group for their contribution in making my time at Chalmers joyful.

Göteborg, February 2011
Thomas Nygårds

THESIS

This thesis consists of an extended summary and the following appended papers:

Paper A
T. Nygårds, J. Sandgren, V. Berbyuk, A. Bertilsson: Vibration Control of Washing Machine with Magnetorheological Dampers. *Proceedings of the 8th International Conference on Motion and Vibration Control (MOVIC2006*), KAIST, Daejeon, Korea, August 27-30, 2006.

Paper B
T. Nygårds, V. Berbyuk: Dynamics of Washing Machines: MBS Modeling and Experimental Validation. *Proceedings of the Multibody Dynamics 2007, ECCOMAS Thematic Conference*, Politecnico di Milano, Milano, Italy, June 25-28, 2007.

Paper C
T. Nygårds, V. Berbyuk, A. Sahlén: Modeling and Optimization of Washing Machine Vibration Dynamics. *Proceedings of the 9th International Conference on Motion and Vibration Control (MOVIC2008)*, Technische Universität München, Munich, Germany, September 15-18, 2008.

Paper D
T. Nygårds, V. Berbyuk: An Adams/View-Matlab Computational Interface for Clustered Optimization of Washing Machines. *Proceedings of the 22nd Nordic Seminar on computational Mechanics (NSCM-22)*, DCE Technical Memorandum No. 11, ISBN 1901-7278, Aalborg University, Aalborg, Denmark, October 22-23, 2009.

Paper E
T. Nygårds, V. Berbyuk: Pareto Optimization of a Washing Machine Suspension System. *Proceedings of the 2nd International Conference on Engineering Optimization (Engopt2010)*, Instituto Superior Tecnico, Lisbon, Portugal, September 6-9, 2010.

Paper F
T. Nygårds, V. Berbyuk: Multibody Modeling and Vibration Dynamics Analysis of Washing Machines, *Submitted for international publication*.

Paper G
T. Nygårds, V. Berbyuk: Optimization of Washing Machine Kinematics, Dynamics, and Stability During Spinning Using a Multistep Approach, *Submitted for international publication*.

The appended papers were prepared in collaboration with co-authors. The author of this thesis was responsible for the major progress of work, including writing, development of theories, coding of test rig software, and implementation of models and methods into software environment for the Papers B-G. For paper A the majority of the writing was carried out by the thesis author and about 50% of the implementation and simulation work.

Contents

Extended summary

1 Introduction

1.1 History of washing machines

Starting from smacking ones dirty clothes on the river side with a flat stick, a long journey of washing has been undertaken. Today machines handle the process for us. Machines which can be pre-programmed, who determine and use the right amount water and detergent, operates when the user sleeps and deliver clean and dry clothes in the morning will soon be available on the market, if not already so. All the necessary technology is already available.

One of the first mechanizations of the washing process was done with hand operated washing machines in which the clothes were moved around in water using a rod or a crank with connected shovels [1]. When the electric washing machines had their breakthrough, in the early 20th century, the machines used the same technique, comprising a so called agitator which shifted the clothes around inside a tub filled with water and detergent. These early machines only performed washing and left the laundry soaked with water. Later, machines could come with a wringer as an accessory for water extraction. A wringer consists of two rollers mounted together, compressing the laundry passing between them. The wringer was operated most often by hand and the construction demanded that the laundry needed to be feed manually to the rollers, at least until they could get a grip. Electrically operated wringers were also available, being just as dangerous as they may seem. Fingers and even arms were crushed before release mechanisms, which separated the rollers, were introduced.

During the early years of electric washing machines, similarities between washing and other household tasks were seen. Electric motors were expensive new technology in the homes and the mechanic principal components behind the washing machines at the time doubled as butter churners and meat grinders [2]. Machines appeared which instead of moving the load inside a fixed tub could wash with help of a rotating or rocking metal tub. This made it possible to see other similarities. Equipping a centrifuge with ridges of various designs, will make the load tumble or move around, thereby giving it the necessary mechanical treatment which is a one of the vital parts of an effective washing process today.

The washing machine nowadays almost always includes a second operation mode, namely spinning or centrifugation, which became common after the Second World War. Spinning means that the water is extracted radially through (high speed) rotation. To accomplish this, a design using double cylinders is used. An outer cylinder called the tub, or container, in which the water is kept during washing, has a cylinder inside called drum or inner tub. The laundry which is to be washed and spun is put into the drum, which rotates relatively to the tub. To make the extraction work, the drum is perforated with small holes distributed along its mantle. The same holes also serves as the in- and outflow path of the water during washing. Lately, additional to the spinning a third operation mode has started to appear on the market. The similarities between a washing machine and a tumble dryer have made manufacturers to add drying of clothes using hot air as a step in the process after the spinning is completed, but still machines with this additional functionality does not have the same performance nor capacity of washing as the conventional washing machines have. However, it constitutes a promising step forward which reduces the original number of apparatuses from three to one.

The typical load capacity for a consumer machine ranges from 5 to 8 kg for a front load machine with a typical standard size of width and depth 60 cm by 60 cm. But there are some examples of machines having 9 kilo of capacity but still fitting inside this standard size. The source of these particular constraints size is kitchen and bathroom fitting standard dimensions, which may vary between countries. There are also other established standard dimensions.

The capacity rating is, as indicated above, normally given in kilos for consumer washing machines in Europe, but the constraining property of the load is seldom the actual weight of the load, as this can be handled, e.g. with stiffer suspension. In reality it is the volume of the drum that determines the capacity. This volume is then related to weight by a density factor, which at the time of examination was $0.12 kg/dm^3$ taking a mean of the machines available on the market in Sweden according to [3]. The factor used for different models varies up to ±15 %. This means that two machines that have the same volume of the cylinder can have up to 2 kg of difference in rated load. One could argue that the manufacturers choose to rate their machines lower than others to diversify their product range. This could be true to some extent but a more probable explanation is that there has been inflation over time in the density the manufacturers use driven partly by market requirements and partly by improved washing process. A trend towards higher density can be seen if the density is compared with the year of introduction on the market [3]. Professional machines have still not experienced this change in density factor and still require 10 liters of space to wash 1 kg of load.

The primary drive forces within washing machine research at the moment are energy consumption and capacity [4]. Energy consumption measurement and labeling of white goods products are regulated in many parts of the world. In the EU, grading on the scale A-G is being used where A is the best [5]. The scale is becoming old after its 15+ years in use and the top performers now consume up to 20% less energy that the maximum grading. In Australia the grading system is called the Energy rating label [6] and the consumption is graded with stars up to a maximum of six. Energy consumption of a washing machine is strongly coupled to water heating during washing. About 70-85% of the total energy is used for this [6], but if the washing machine is seen as a component of the washing and drying system also its drying process contribution becomes important. The more water that can be extracted through spinning the less must be removed by other, possibly high energy consuming, means such as heating. For example, the Australian energy rating label grading takes the spinning performance into consideration when the grade is determined. This means that increasing the spinning performance will, justifiable, be a drive force towards improving energy consumption rating.

A typical washing machine which has to heat water takes between one and two hours to complete its task and deliver clean load. Nowadays all machines have at least two main operation modes, washing-rinsing and spin drying. The conventional operational modes can be divided into several program steps. The washing mode includes water take-in, water heating, mechanical treatment. After completion, rinsing is performed and after this the spinning takes place. The steps can be repeated several times, giving for example a pre-wash program for extra dirty loads, or extra rinsing steps for anti-allergenic purposes. The spinning operation mode typically comprises load distribution, load characterization, the actual spinning step with purpose to extract water, and sometimes a final step to loosen up the spin-dried load. The load distribution step and load characterization step are normally repeated until the load is sufficiently well distributed inside the drum. The spin step compromises normally the following of a time based spin-speed scheme which is dependent on the user´s selection of desired final (highest) spinning speed.

1.2 Washing machine design concepts

The first type of categorization that should be done when dealing separation between washing machine designs is based on how the rotating drum axis is oriented and later how the clothes are loaded into the machine. There exist two different drum orientation concepts: vertical and horizontal mounting. Vertical mounting means that the drum is mounted via an axle and bearings normally only at the lower gable of the drum. The load is put into the machine at the top gable of the drum which in some cases can be covered with a supporting lid or sometimes left completely open. This concept is the far most common in North America, having a market share of about 65% [7]. These machines exert their mechanical treatment to the load via a screw-like agitator mounted in the center of the drum or by rotating a separate bottom piece creating a whirl in the water to move the load around.

The horizontal mounting concept can be split into two different configurations, single-end mounted and both-end mounted drums. Machines with single-end suspended drums usually have two bearings mounted at only one end of the drum whilst both-end suspended drums usually have one bearing at each side of the drum. In the latter concept as well as for the vertical mounted drum concept the laundry is loaded from above, hence they have been given the name top loaded washing machines or TLWM. To enable installation under counters, for stacking of home appliances on top of each other and by tradition, consumers want machines which load the laundry from the front end. Machines with this possibility have analogously been named front loaded washing machines or FLWM. Currently machines with design according to the horizontal single-end mounted concept are the only available which have front loading capability. Work has been and still is being done on other designs, like a both-end suspended drum with a large front bearing through which the load can be put, and thus enabling the front loading capability [8].

From the vibration dynamics point of view, a TLWM horizontal-axis drum has more similarities with a FLWM then with vertical-axis TLWM. This is true because the two first machines' conceptual difference is a 90 degree rotation of the spinning axis in the plane. Both suspension concepts have the gravity acting in their radial direction, i.e. the gravitation vector is orthogonal to the axis of rotation. This makes the load tumble and given mechanical treatment, something that is not the case for the TLWM with vertical-axis drum. The level of mechanical treatment in horizontal-axis washing machines is generally considered as one of their advantages, compared to vertical-axis machines.

A washing machine with a drum suspended in both ends can be built significantly smaller than a single-end suspended washing machine, whilst having the same load capacity. To be easier to place in furnished houses with standard-sized kitchen and bathroom fittings, the saved

space is used to make the machine less wide. Typically, these machines are as deep as single-end suspended machines but are only 75% as wide.
The drum is in almost all cases a cylinder with round mantle but there exists also machines with other geometry of their drums. For example, the American manufacturer Staber [9] has a machine equipped with a six sided drum and an eight sided tub in their product range.

To hold the water during washing, the drum is mounted inside a tub or container. The tub is then suspended in the housing, or outer casing, with some allowance for it to move around. A soft suspension, comprised by springs and dampers, is only needed during spinning. In a machine just performing washing the forces generated from the rotary motion are so low that vibrations are not any major problem even if the tub is mounted rigidly to the housing. To make a good suspension system for spinning it shall be designed to be sufficiently soft in stiffness so that the spinning is performed at over-critical speed. With over-critical it is meant a spin speed greater than the resonance frequencies giving high transmissibility of force. If the suspension system is designed in such a way that the force transmissibility is low for the spinning speed range of the machine, the suspension will perform well in terms of vibration output. But, if the suspension is too soft in terms of stiffness the large range of loads that the machine need to handle might cause large range in vertical equilibrium position of the tub at these load conditions, and also cause the tub to hit the housing during spinning. Different concepts of washing machine suspension systems exist. They can be divided into groups: top-hung suspensions and bottom-mount suspensions (see Figure 1).

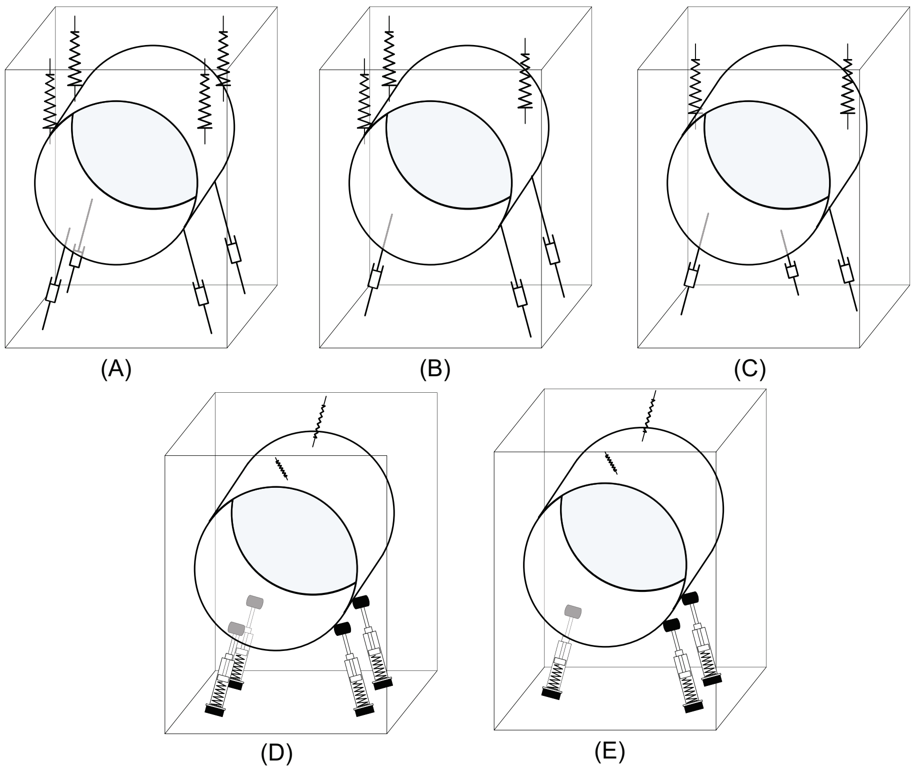

Figure 1*: Typical suspension concepts: top-hung (A-C), bottom-mount (D-E)*

In hanging tub systems two, three or four springs are mounted between the tub and top of the machine housing. Dampers are usually mounted at the bottom of the machine. Bottom mount suspensions support the weight of the tub at between three and four different positions with springs which sometimes are combined with dampers into one unit, a so called strut. Bottom mount suspensions with only two struts has also existed earlier [11] but have been discontinued [10].

The advantage of bottom mounted suspension systems are that the path for propagation of forcers does not pass through the whole housing structure but only the bottom part. This means that the housing does not have to bear the weight of the tub and can thus be made weaker, something which might reduce cost.

The advantage with top-hung concept is the stability it gives to the system. One can compare the top-hung concept to a hanging pendulum system whilst the bottom mount concept compares to an inverse pendulum. The bottom mount system is easiest made stable with some additional structural components providing a stabilizing torque to the tub, typically rubber bushings. In a top-hung system bushing are usually not needed.

To make the drum rotate during washing and spinning different concepts have existed. Early washing machines had manual drive of the drum, but over the years even machines equipped with steam engines and combustion engines have seen the market. Now the only commercially available power is electric motor drive. Most machines have a pulley connected to the axle of the drum being driven via a belt with a motor located at the bottom of the machine. With a motor axis-pulley ratio of about 1:11 the speed of the electric motor can be as high as 20000 rpm as washing machines perform spinning with speeds ranging from 800 up to 1800 rpm. But lately, several manufacturers have started to develop machines with direct drive, i.e. where the motor is mounted directly on the drum axle. This will not only make the design more compact but may also reduce the noise, due to the lower rotational speed of the motor.

1.3 Earlier work

The dynamics of washing machines has been studied over the latest years with increased interest due to stricter consumer requirements on performance and environmental issues, such as water usage and energy consumption. Lowering of energy consumption has, as stated earlier, been driven also by legislation. In the late 80s, before spinning speed control restricted by imbalance became commonly used, problems with walking were common and thus a topic of interest for manufacturers. Walking is the name of the behavior which occurs when the machine moves from its position, due to high unbalance forces during drum rotation. The problem has however raised again as a consequence of a change in common machine installations. Nowadays consumers may want to install their washing machine on wooden floors or on other slippery surfaces.

Washing machine dynamics is not one of the biggest topics in the field of research, but there is relevant literature. In [12] computational a model of a washing machine was used to study the suspension system dynamics with the aim of to optimize its performance with respect to stability and vibration propagation. In [13] measurements on a TLWM was performed and compared with simulations of a 6dof model implemented in the Adams environment [14] with the aim to predict tub kinematics during a transient spinning cycle. A finite element model of a similar TLWM was developed in [15] for prediction of stress and strain of the tub system.

Computational power development under recent years has enabled the use of modeling tools like in [16] where the software Dymola/Modelica is used with focus towards motor control. Commercial multi-body software which reduces the step from CAD environment to a dynamic model has also become possible and several models constructed in this type of environment have been developed. A washing machine model with the same suspension configuration as in [16] was developed in [17] with the future application to serve as input source to an acoustic model for prediction of noise. In [18] a rigid body model of the drum-tub system implemented in Adams is completed with flexible components of the housing and used with good prediction results for feet forces. The focus of the mentioned paper is also to enable use in acoustic simulations. Here for simulation of the side panels, which are big sources of structural noise in some machines according to [19] and [20]. Presented in [21] is an analysis of the tub dynamics of a washing machine with a flexible drum axis conducted using a dynamic model of a suspended tub. The presented mathematical model has 12 degrees of freedom and models flexibility of the drum axis and bearings using a discretized approach. For the purpose of numerical verification, the results for unbalance response are presented and compared with the experimental vibration test. The housing is not taken into consideration in the presented model. In [22] the influence and importance of proper installation of washing machine in terms of feet adjustments is investigated both with models and with experiments. It is shown that proper adjustment will reduce the vibration of the washing machine significantly. In addition, the effect of so called vibration damper pads which are to be placed under the washing machine to reduce vibration propagation into the flooring structure is investigated. Experiments confirm what is presented in [23] and show that even a worsening effect is possible. A conclusion that can be drawn from these publications is that even though is possible that the pads might work for some machines installed under some conditions they can not be recommended in general.

In [24] several models of stiff suspension washing machines are considered with the focus of modeling of imbalance forces and compared on conceptual levels. The topic of the paper is walking of washing machines and it is treated with the establishment of criterias to be fulfilled during design. In [11], these conditions are established for washing machine installation on surfaces with different slopes and the risk of tipping is investigated. Several comparisons of vertical- and horizontal-axis washing machines are presented. Top-hung and bottom-mount suspension systems are modeled in the plane and compared. Different criteria for determination of the margin to walking during the spinning cycle are presented too, extending [24]. In [12] the focus is put on prediction of the tub motion together with stability issues, i.e. translational walking. In [25], which is from the same author, a kinematic and stability (walking) approach to the optimization of the washing machine suspension is presented. The kinematic cost functions are there measured amplitude of lateral, vertical, and orbital motion of point on the back and front end of the tub. Walking referred to as stepping in the paper is addressed with a necessary and sufficient condition (sic) based on “friction resistive force”. The presented criterion does however not take into consideration the sign of the forces in the horizontal plane and may not cover all cases. The developed condition is however used as a constraint during optimization of kinematics with good results. In [26] a formulation denoted “slip margin” is presented, 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. With that formulation the sign of the forces are handled, but only the lateral forces are used in the condition. In [27] the topics of limitation on deflection of tub, walking, and force transmission are discussed and a step by step algorithm for selection of suspension stiffness and damping based on the two first objectives is presented. The criterion is based on experiments on a custom made rigid housing for one particular geometrical configuration of the suspension.

One promising technology for vibration reduction is counterbalancing. Several strategies to counterbalance an imbalanced load have been proposed. Some are active solutions which require external sensing and control to position counteracting solid masses. Simulations on such devices have been performed in one plane in [26] and two planes [28]. The paper [26] presents a simplified 3D dynamic model of a horizontal-axis portable washing machine with a balancing device. Other solutions to remove static imbalance involve translation of the spinning axis relative to the drum [29], [30] and [31] by using different mechanisms to control its eccentricity. Using water, which is easily accessible in washing machines, to actively position the total center of mass has also been discussed in [29]. There are also passive solutions to the problem of counterbalancing. In a passive solution no external control stimuli to position the counterweights is needed, instead the positioning is done with the circulatory forces coming from rotation of a counterweight. Passive counterbalancing can be done with a liquid filled ring called the Leblanc balancer. In [32] a mathematical model of such a hydraulic balancer was derived and implemented in a model of a vertical-axis washing machine dynamic analysis of an automatic washing machine during spin drying mode. The model of the hydraulic balancer was validated by the experimental results at steady state conditions. The technology is also discussed in [11]. In [33] a similar derivation of the technology is presented. Passive balancing, also called automatic balancing can also be done with solid masses like in [11] where a concept based on an elastically suspended solid ring which can move in the cross axis plane radially is presented. But typically, the masses consist of balls of stainless steel [34], [35] and [36] rolling in a housing or with pendulums or sliders like in [37].

The use of semi-active damping in washing machine would be beneficial to its vibratory performance. A prototype magnetorheological damper designed for washing machines was actually manufactured by Lord Corporation [38] and put to the market for manufacturers to test. In [39] this damper was used in a front loaded washing machine to test different mounting configurations and control strategies, comparing them to the conventional and undamped configurations.

Researchers are trying to incorporate feedback control of the washing process with respect to the amount of dirt using different sensors, for example so called electronic tongues. This could lead to shortening of the washing process, meaning time consumption and energy saving, reduction of detergent use, and as a result of this also reduction of wear of clothes [40]. Wear during washing constitutes about 20% and if tumble drying also is added an additional 10% the total clothes wear can be assigned to the process of keeping clothes clean [4].

Sensors and hydrodynamics modeling have been used to extract information about the local flow velocities in the drum washing to monitor effective cleaning [41].

Other methods for sensing properties more related to the washing machine dynamics during operation have also been investigated. Estimation of the state of the load has been performed by for example [42] where artificial neural networks were used to detect the amount of imbalance, location of the imbalance in the drum and the amount of balanced load, with inputs coming from a load cell under the machine, an accelerometer and two laser distance sensors. In [44], the angular position and mass of the imbalance by direct relation to the orbital displacement of the tub.

Detection of the current state of load (balanced and imbalanced), can also be done by measuring the motor power consumption and variation of the spin speed during a revolution [10], [43]. This is one of the methods used in production machines today. To make it work the drum must be rotated according to certain rotational speed schemes. This imbalance control could for example be improved with deflection based sensors which are used today for indirect measurement of the weight of the load. The information of this sensor is normally fed to the user giving for example advises on which amount of detergent is needed for washing and to help the washing machine decide the amount of water needed.

The addition of more sensors to the machine will enable for future multi usage of the sensor information and drive the development of the washing machine towards more an even more intelligent mechatronic product.

1.4 Purpose and scope of the project

The general purpose with the project is to contribute to the process of research and development of washing machines in topics related to vibration dynamics, by evaluating existing, adapted or new suspension concepts and corresponding control strategies. Here, specifically by reducing the vibration output of a front loaded washing machine to its surroundings whilst keeping or increasing load capacity and maintaining or increasing stability. The purpose is also to develop a mathematical and computational model of a modern front loaded washing machine based on multibody system formalism and measurements of functional components to be used as a tool for analysis and optimization. Also included, is development of criteria for classification of vibration related performance for washing machines during spinning, taken into account different loads and spinning operational scenarios. The strategy is to use the developed models and tools for system dynamics analysis and in efficient multiobjective optimization routines. In this way, improvements of the washing machine mechanical design can be suggested.

The project focus is on vibration problems coupled to spinning with a non-tumbling load. The scope of the project covers bottom mount washing machines of horizontal-axis type and frontloading capability designed for consumer/household/community usage. However, many of the developed vibration criteria might also be applicable to other types of washing machines and rotating machinery with soft suspensions.

2 Modeling and analysis of washing machine dynamics

2.1 Mechanical model

Washing machines discussed in the context of this thesis consist of the following main components: a horizontally mounted drum containing the load, a tub keeping the water inside the system, a motor, a suspension system, and housing.

The drum is in all front loaded washing machines of any spread of importance, as earlier stated suspended/mounted with two bearings of some sort in its back-end. As the single-end concept supports the forces at only one side, the cylinder end must be stiff and strong to support the torques produced by drum weight and load forces. An easy and native way to make a part stiffer is to make it thicker, like for example in the current machine, with of a cast iron. The thicker cylinder end or the tub results in an increased weight of the back end of the whole suspended tub system. Counterweight masses are therefore added to make the center of mass of the total suspended mass become near to the center of the drum in its longitudinal direction. Counterweights are often made from cast iron or special high density concrete [8] and are placed in the front of the tub in the upper and lower ends. The result is of course an increased total weight of the suspended tub-structure. However in force driven vibration problems this is not bad. With increased weight of the affected system the resulting accelerations becomes smaller. Some manufacturers actually add additional counterweights on top of the tub and also inside the housing to improve vibration performance.

The specific washing machine model which is under the study in this thesis is a front loaded bottom mount suspension machine without bellow seal. The bellow seal is usually a rubber cylinder which tightens the gap between the moving tub and standing front hatch in machines with a front hatch fixed to the housing. Here, in the studied concept the front hatch is fixed to the tub meaning it vibrates with the suspended body and does not allow propagation of forces between the tub and the housing. Only an ordinary rubber seal (gasket) is needed for restriction of water flow. A rubber bellow connected between housing and tub is quite common in household machines from other manufacturers. This configuration is more frequent in multiple household machines like the Wascator series from Electrolux or in pure industrial machines.

On the axis of the drum of the studied machine a pulley is mounted. To this pulley, the torque which drives the drum is provided from an electric motor via a belt transmission. The motor is bolted on the tub via a support structural plate named the cradle. It is between the cradle and the housing that the suspension is mounted. The suspension consists, in principle, of several struts with a spring and a damper incorporated in the same unit, see Figure 2. Small springs are connected between the top of the tub and the housing and are therefore strictly also a part of the suspension. Nevertheless, it is from the cradle, through the struts, and to the housing that the principal path for suspension forces goes, making it a bottom mount suspension. In each end of the respective strut a rubber isolator/bushing is mounted. In the upper end it is made asymmetric and in the lower end round symmetric. The damper is a low-cost friction damper widely used in washing machines, which basically consists of cylinder in which a friction element slides. The friction element in its turn consists of a reel and a sponge. The sponge is compressed against the cylinder by the reel and causes friction forces. The amount of friction damping is decided by the diameter of the reel, thus giving a variation of the pre-stress compressing the sponge.

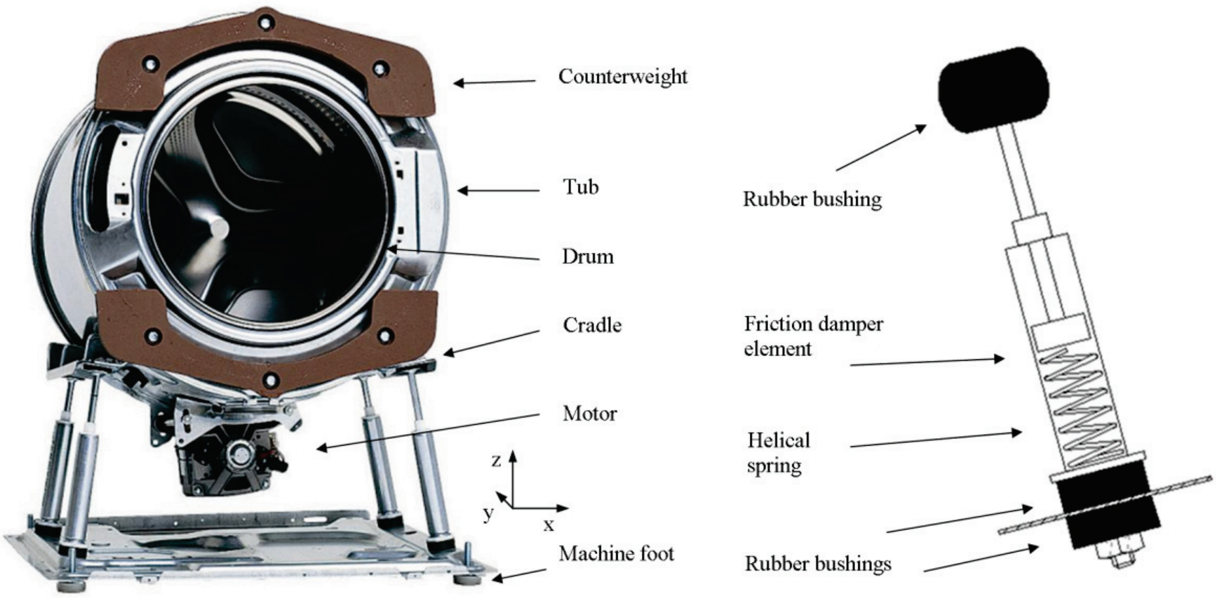

Figure 2: *The inside of one of the studied washing machines, together with a detailed view of a strut.*

A piston which can rotate around its axis of operation push the friction element downwards compressing the spring, and the spring hence provides a force in the opposite direction. The inside of the tube is greased to provide as little wear of the friction element as possible. Dampers degrade with time and may lose up a significant part of its original damping capability when the life of a washing machine has ended. The round-symmetric bushing at the lower end of the strut is clamped to the bottom plate between the strut and a mounting bracket and fixed with a nut. In the upper end it is inserted into a cup and held in place by the gravity forces of the tub. The washing machine has four system isolator feet of rubber which can be adjusted in height to make the machine leveled on an uneven base or floor, in this thesis also denoted hosting structure.

In addition to the above described components, rubber hose connections for water from inlet valves and to drainage pump are connected to tub. Inside the housing at safe distance from the moving tub various electronic components for control of operation, such as sensors and actuators for water control are placed. The housing and these components together with main circuit board, motor control board, detergent box, wires etc. are not shown in Figure 2 for reasons of clarity.

2.2 Mathematical model

It is assumed that the mechanical system modeling the washing machine comprises a set of inertial functional components (IFCs), stiffness functional components (SFCs), damping functional components (DFCs), and stiffness-damping functional components (SDFCs). All the functional components are joined by couplings derived from the physical washing machine design.

To the group of IFCs belong all parts of the washing machine which exhibit inertia properties. In our consideration these are: the drum, the tub, the unbalanced load, the housing, the motor rotor, the suspension strut pistons, and the suspension strut cylinders

To the group of SFCs belong the strut springs and the top stabilizing springs. Functional components which exhibit mainly damping properties are called DFCs, and in the washing machine the strut friction dampers belong to this group. There are also components in the current washing machine design which have both stiffness and damping of significant amount, SDFCs. This group comprises the lower and upper bushings of the washing machine as well as the rubber feet. See Paper F for details on how the functional components are modeled.

Analysis of the description of a washing machine mechanical model, system functional components and imposed constraints leads to a conclusion that the washing machine is modeled as a controlled multibody system (MBS) comprising 13 rigid bodies with constraints removing 66 degrees of freedom, giving the system totally 12 degrees of freedom. In state space representation the dynamics of the considered washing machine is described by the system of differential equations and initial conditions

$$\dot{\mathbf{x}} = \mathbf{f}(\mathbf{x}, t, \mathbf{p}, \mathbf{d}, \mathbf{u}) \tag{1}$$

$$\mathbf{x}(0) = \mathbf{x}_0 \tag{2}$$

Here $\mathbf{x}=[x_1,x_2,x_3,\ldots,x_{24}]^T$ is the 24-dimensional vector of state variables $\mathbf{p}=[p_1,p_2,p_3,\ldots,p_{Np}]^T$ is the N_p-dimensional vector of given system structural parameters, $\mathbf{d}=[d_1,d_2,d_3,\ldots,d_{Nd}]^T$ the N_d-dimensional vector of washing machine design parameters (design variables), and $\mathbf{u}=[u_1,u_2,u_3,\ldots,u_{Nu}]^T$ is the N_u-dimensional vector of external control stimuli, e.g. forces and torques acting on the system, spinning speed, etc. The output of the computational model is a vector of dimension N_y defined by $\mathbf{y}=\mathbf{g}(\mathbf{x},t,\mathbf{p},\mathbf{d},\mathbf{u})$. The vector function $\mathbf{f}$ is specified internally inside the numerical environment through the implementation of the computational model. Here the vector function $\mathbf{g}$ also is specified according to standard results sets of the numerical environment with additional user specific measure functions. The initial state $\mathbf{x}_0$ is given in terms of initial positions and initial velocities. The static equilibrium positions are not necessarily the same as the one described by the positions of $\mathbf{x}_0$. For most usages of the washing machine model implemented in the environment the calculation starts at static equilibrium, so therefore the preloads of the force components have been designed to give equilibrium positions that is as close the positions in $\mathbf{x}_0$ as possible.

Different parameters can, during optimization, be considered as components of vectors **d** and **p**. For instance, the components of vector **d** can be stiffness of springs in struts, rubber bushing stiffness and damping, strut position, etc. Inertia of the drum and housing together with other parameters which should be considered given at all times comprise the vector **p**.

2.3 Computer implementation

By using the developed mathematical models of functional components (Paper F) the computational model of a washing machine is implemented in the commercial MBS software

Adams/View from MSC.Software [14]. One reason for the selection of this computational environment is the visual feedback of the kinematics and dynamics of the model the software provides. By using this capability model debugging is greatly facilitated. The other reason is compatibility with drawings produced in CAD-Software. The IFCs of the washing machine model are determined based on CAD drawings which are taken from production and research machines. Mass and inertia can in this way be automatically calculated given the density of the material of the part in question. Even geometries can describing volumes can be parameterized. In the model this is done to describe different load scenarios in the washing machine. This gives the model a flexible and at the same time robust definition of mass and inertia properties. By robust, it is meant that there is a visual update of the model if a part with different geometry is used in the model. Also specification and checking of locations of joints or other constraints is facilitated as their exact location and orientation are indicated in the model with icons. The computational model is built in a modularized way meaning for example that multiple definitions of several parts can be stored in a database and selected based upon the version of washing machine model that is to be studied. Examples of this are different counterweight designs which have a big impact on inertia properties of the system, or different detergent boxes which have impact on collision detection values.

Adams/View handles the solving of the initial value problem (1) and (2), i.e. calculation of state variables and output vector. Inside the same software the results can be animated to visually inspect the motion of bodies and motion signature characteristics of a particular suspension system.

3 Verification and validation of the models

The model of the washing machine is built around functional components of structural parts of the washing machine. It has been the aim that as many as possible of the functional component models parameters should be identified separately if possible. To accomplish this several test rigs have been constructed. The parameters of the DFC modeling the strut damper were estimated using optimization routines, i.e. formulation of the parameter identification problem with the objective to minimize the discrepancy between measured data and modeled data. With the purpose of acquisition of the measurement data a strut test rig was built based on a standard production damper test device. The test rig provides kinematic excitation and measures force response of the test subject. It is capable of exciting a strut of washing machine size with sinusoidal motion with amplitude of up to 25mm. The frequency range that can be used to test a typical washing machine strut is 0.2 Hz-30Hz. Lower excitation speeds are possible for test subjects with lower damping and stiffness.

In Paper C the rig is presented together with raw measurement data. A description of six suggested models, the statement of the parameter identification problem and resulting parameters are available in Paper F.

Dynamic experiments have also been performed on the machine rubber foot using low amplitude kinematic excitations. Experimental data was used for model parameter identification for the SDFC modeling each machine foot in the vertical direction. To acquire measurement data to be used for the estimation of the model error an available test rig was used and adapted to fit the operational conditions of the foot with respect to frequency, amplitude, and prestress (corresponding to the weight of the machine).

Stiffness parameters for the SDFCs modeling the rubber bushings were taken from static experiments, together with parameters for SFC modeling strut springs and top stabilizing springs.

A description of the suggested models, one linear and one non-linear, the statement of the parameter identification problem and resulting parameters are available in Paper F.

The washing machine model is, as stated, implemented in the MBS software Adams/View. In the manual of this software a verification procedure is given. It should be executed to verify that the numerical results of a model implemented in the environment produces trustworthy results corresponding, within a tolerance, to analytical results. Apart from resolving the source of obvious issues like warnings and assess these, the strategy is to sequentially reduce the integrator tolerance of the simulation until no difference is noticed. The recommended approach is to use steps of one order of magnitude at a time until convergence is reached. The convergence criterion is fulfilled according to the manual when two consecutive simulations with different tolerances give no noticeable discrepancy of the results. After convergence, the highest tolerance of the two should be used to enable as time efficient solving as possible. In Paper F this procedure is executed and the results are shown.

4 Washing machine design optimization

4.1 Spinning operational scenarios

Multiobjective optimization is one of the key subjects of this thesis. When using multiple performance measures it is not always so that all scenarios are critical to all performance measures. Often when evaluating a system on with respect to different performance measures, different scenarios become critical with respect to each performance. Spinning at high speed generally cause more vibration noise whilst the risk of oscillatory walking is highest under 500rpm. In addition to this the washing machine is versatile in the sense of load capacity and critical events can arise during different conditions. The imbalance of the washing machine is hard to control and its center of mass may end up at different positions inside the drum. To ensure a robust construction different scenarios could be tested. In the field of engineering the results from studying a worst case is often used as a criterion for dimensioning. The washing machine suspension design approach utilized in this thesis is also based on this concept, and the presented optimization method is dependent on it.

In Paper G, the term critical Spinning Operation Scenarios (SOS) is introduced to synthesize and define load configurations and corresponding rotation schemes of the drum, and constitute a critical case worth studying. Different spinning operational scenarios are defined and used as conditions upon which performance shall be evaluated for different optimization problems.

Mathematical definitions of a SOS together with restrictions that can be put on different scenarios to be valid are given in Paper G.

4.2 Criteria for evaluation of vibration related performance

To differentiate washing machine designs that lead to good performance from those who lead to bad performance measures that are able to catch washing machine specific features are needed. Below three criteria are discussed and the motivation for their use together with references to definitions in the appended papers are given. Over the time the criteria have evolved and therefore slight differences between definitions in the papers exist.

Collision avoidance between internal parts

As stated before, a direction within washing machine development and research is capacity increase. Capacity increase leads to the need of a big drum which in its turn needs a bigger tub to contain the water. If it is not possible to change the size of the housing due to dimension standardizations, the space between tub and the housing with its internal parts will have to be smaller. Collision between parts inside the machine can cause a shock and make the machine jump, or generate noise. Collisions should be avoided due to reasons of machine durability but not least due that the user will experience collisions as a sign of bad quality [10], [45].

So, to be on the safe side the intuitive desire when it comes to tub kinematics is to have as little motion of the whole tub as possible. As collisions can occur at various locations it would be ideal to measure distance all over the tub surface with respect to the housing and its components. This is not easy to do in a physical machine and in the computational model it is not easy either. This statement is confirmed by the fact that even though it can be done inside the computational environment used, i.e. Adams/View, it is a very time consuming process.

When searching for a less time consuming method one should remember that motions in all directions are not necessarily critical to keep low, i.e. in some directions the motion might be small regardless of imbalance, and in some directions there are empty spaces which cannot be used to increase capacity but maybe could "be used" for motion. Additionally, the distance between some of the moving objects might be small but relatively constant regardless of imbalance. To deal with this problem a number of critical points $\mathbf{P}_k$, and respective critical directions $\boldsymbol{v}_k$, were defined on the tub of the washing machine. Each point is also associated with a maximum magnitude of motion $X_k^{\max}$, according to results of a combination of measurements on a physical machine and in drawings, depending on the state of the model to be studied.

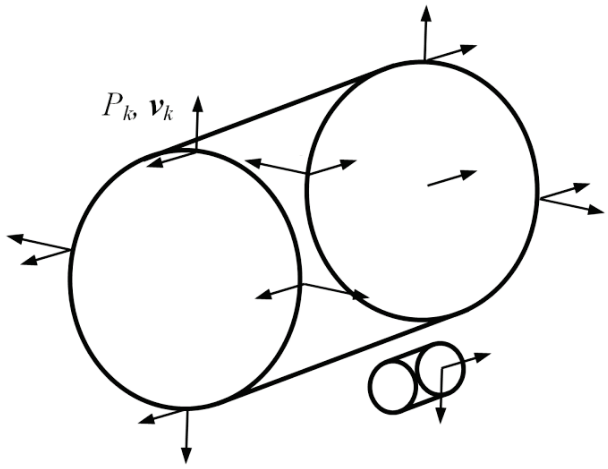

Figure 3: *Typical definition of points and directions for a generic washing machine with a motor mounted under the tub*

Taking into account the above verbal statements, the kinematic objective to be minimized for a prescribed SOS_j of a washing machine is defined as

$$F_{Kj} = \max_k \left(\max_t \left(X_{kj}(t) - X_k^{\max} \right) \right),\ k = 1,2,...,N_k,\ t \in [0, T_j],\ j = 1,2,...,N_{SOS} \qquad (3)$$

Typically for a good suspension and admissible SOS_j, no collision occurs and $F_{Kj} < 0$ for all j=1,2,3… N_{sos}. If at least for one critical point P_{k*} and at least for one SOS with index j^* $F_{Kj*}(P_{k*}) > 0$, then collision between the moving parts of a washing machine will happen, if no safety margin is being used.

Vibration isolation

Vibration output causing noise and vibration of its surroundings is maybe the first that comes to the mind of the user of a washing machine when vibration is discussed. One of the challenges when it comes to washing machines is to define how the impacting vibrations should be measured. In Europe today, the only official vibration output related performance classifications performed are various sound pressure level measurements. The consumer of a washing machine often has these values at hand when comparing machines before a purchase. The sound levels are not necessarily related to how much vibration the washing machine exerts on its surroundings. Much noise can come from the motor and side panels can be poorly designed and acting like loudspeakers. A few attempts have been made to classify the amount of vibrations that a washing machine causes on the surroundings directly due to contact. Examples of this are a method for testing developed for Consumers Union in USA [23] and the method previously used by Swedish Consumer Agency [46]. Both methods are based on measurement of accelerations at specified points in its surroundings and include a standardized floor (not the same between the methods). The problem with vibration impact measurement on a floor or another external object is that the properties of the external object influence on the measurement value, i.e. the mass and stiffness parameters must guarantee equal dynamic response between all floors used for measurements for comparisons between test labs. The difficulties have caused problems in ensuring repeatability [19] and are perhaps one of the reasons to that a standard international measurement method not yet is established.

In optimization, incorporation of a method based on measurement of surroundings in a computer model to be used for optimization might be unnecessary complicated. If the dynamics of a floor also must be calculated more computational resources are needed, apart from the risk of that the incorporated floor model does not represent the actual floor to a sufficient extent. Also the aspect of over-fitting of the washing machine components to a particular floor must be considered in such a method.

To skip the complexity of floor modeling the transmitted force to a fixed base is measured directly. In Paper B and Paper F a whole machine test rig is presented. In this test rig it is possible to measure the transmitted force in vertical direction. To enable the use of same cost function for measurement and simulations only the vertical direction of the force data was used. Experimental studies using different SOS have shown that the vertical direction is the dominating when it comes to washing machine transmitted forces, both in absolute values as well as in magnitude of oscillations. Second in magnitude is the lateral force when it comes to front loaders.

Different performance measures have been defined over time to quantify the vertical force. These functionals have been used as objectives in optimization or as performance measure for dynamic analysis in Papers C-G. The following functional is the most generally defined and also used in Paper G, which is the latest.

$$F_{Dj} = \sum_{i=1}^{n_f} \sqrt{\frac{1}{t_{fj} - t_{0j}} \int_{t_{0j}}^{t_{fj}} \left(F_{zij}(t) - F_{zij}^{0} \right)^2 dt}, \ j = 1,2,...,N_{SOS} \tag{4}$$

Here F_{Dj} is the sum of the RMS values of the vertical components of $\mathbf{R}_{ij}(t)$, $F_{zij}(t) = \left\langle \mathbf{R}_{ij}(t), \mathbf{e}_z \right\rangle$, at the feet of the machine i=1,2,…,n_f. The resultant vector of the reaction forces $\mathbf{R}_{ij}(t)$ acting at the center of the i^{th} foot. It is obtained, in this thesis, by the Adams model of dynamics of a washing machine, within the assumption that the contact between feet and hosting structure is ideal, i.e. a point contact. The reaction $\mathbf{R}_{ij}(t)$, is evaluated during the timeframe $t=[t_{0j}, t_{fj}]$ of SOS_j, j=1,2,…N_{SOS}. F_{zij}^{0} is the vertical component of the reaction force at static equilibrium of the machine.

Oscillatory walking

Walking is the term used for an unstable behavior of the washing machine during operation. The behavior occurs when one or more of the feet of the machine is/are shifted from its/their installed position. Walking has been defined in [24] to have three modes: Translational slip, rotational slip and tip. The tip mode is of very little concern in washing machine suspension stability today that uses electronic imbalance control of high quality, so the remaining concerns are sliding in the plane.

The cause of walking behavior can be one or more of the following:

- Bad installation of the machine
- Installation on a slippery surface
- High imbalance during spinning
- Bad suspension

A typical bad installation comprises poorly adjusted feet of the machine, which means that the weight of the machine is not distributed evenly. This may cause the machine feet to lose the friction grip more easily during spinning. Another example is installation of the machine on an inclining hosting structure, leading to bad leveling of the machine [24].

The surface on which the machine stands is of great importance as one of the possible conditions for sliding to occur is when the friction force is too low to keep a foot in place relative to its underlying surface. Additionally, it can be a problem if an object (drip-pan, carpet etc.), placed under the machine, slip relatively to the foot or to the hosting structure. This underlying object can produce a more slippery contact towards the machine or towards the hosting structure than the machine foot contact would be directly towards the hosting structure. From a vibrations point of view is not recommended that low friction objects are installed under the machine, especially not drip-pans made of hard plastic. An object increasing the friction would however be beneficial to the walking performance.

A washing machine comes with specifications on installation requirements and with operation instructions, but still it is spinning of an imbalanced load that is the cause of the forces which will cause the machine to walk. If the distribution algorithm of the machine behaves poorly and the machine fails to detect a too high imbalance the machine may walk even though all requirements are fulfilled by the user.

A badly designed suspension system with respect to walking is such that in which the lateral forces become unnecessary high, or such that the feet also must provide an unnecessary high resisting torque to prevent the machine from rotating. A somewhat classical, but always unexpected demonstration is achieved if the user forgets to remove the transit bolts before operation. The resulting stiff suspension will lead to machine walk like there was no tomorrow.

The objective stated in equation (4) has the purpose of minimizing the vibration output by minimizing the transmitted vertical forces between the machine and the hosting structure. The vertical force is related to walking of washing machines but walking is not addressed with the above objective. It might even have the opposite effect if the vertical forces are minimized with the drawback of increased lateral forces. When the machine walks, one of more feet move relatively to the machine's hosting structure in the horizontal plane. To quantify the performance of a machine in terms of walking two approaches have been defined and used. One, presented in Paper E relies on local counteraction, i.e. that the resulting friction force precisely counteracts the applied force to the foot, see Figure 4. With this definition both translation and rotational sliding is targeted as the resulting torque around each point in the plane caused by each foot is zero. The drawback is that the approach puts high requirements on each foot even though walking can be prevented with more loose conditions.

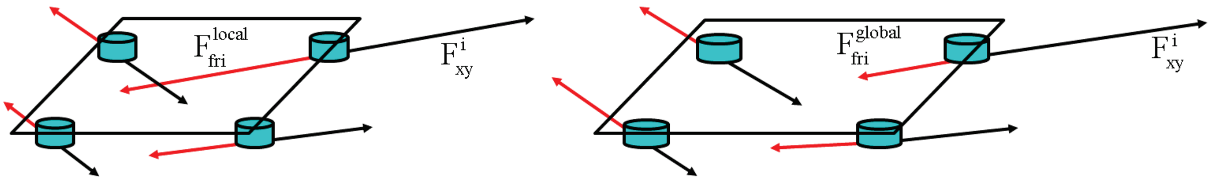

Figure 4: *Local and global counteraction with friction*

In Paper G an additional approach is presented. It is based on the assumption that only global prevention is required. This means that the forces that originate from the washing machine only must be counteracted globally with the friction force. With global counteraction it is meant that the sum of the forces in the plane originating from the washing machine will equal the sum of the friction forces. This definition gives freedom to the origin of the friction, i.e. it is not important from which foot the force originates. Noted must be that if the friction forces are distributed badly it might give rise to a resulting torque. To avoid this, a requirement on the resulting torque must also be fulfilled.

The second approach, based on global counteraction will result in much lower values of the friction that will keep the machine from slipping. This approach, which is more computationally intensive, is believed to give values closer to the true and gives the necessary and sufficient conditions with respect to prevention of walking. A comparison of the two criteria is given in Paper G together with applications of them.

4.3 Statement of the optimization problem

The general formulation of the washing machine optimization problem can be written as follows

PROBLEM A

Determine the vector of structural design parameters $\mathbf{d}_*$ and corresponding state vectors $\mathbf{x}_*(t)$ which satisfy the vector variational equation

$$\min_{\mathbf{d}}\left(\mathbf{F}\left[SOS_j,\mathbf{d},\mathbf{x}(t)\right]\right)=\mathbf{F}\left[SOS_j,\mathbf{d}_*,\mathbf{x}_*(t)\right],\ j=1,2,\ldots,N_{SOS} \tag{5}$$

subject to the differential equations of motion (1) initial conditions (2), restrictions and constraints on spinning operational scenarios, see Paper F. Here $\mathbf{F}$, is the N_F - dimensional vector of cost functions. The evaluation is to be done on a set of N_{SOS} given spinning operational scenarios.

If N_F=1 the solution to Problem A is an optima described by a point. If N_F>1 a Pareto front or a Pareto surface will be the solution. Coupled to the Pareto front or surface is a set of vectors $\mathbf{D}_*=\{\mathbf{d}_*\}$ of structural parameters $\mathbf{d}_*$ which each is a vector of optimal parameters for a point on the Pareto front.

There are different solution approaches to the above defined problem. A classical approach [47] is to optimize on just one of the objectives and treat all other objectives as a constraints bounded by some allowable range ε_j. If the Pareto solution is searched for then this approach is repeated with different ranges ε_j. A bi-objective optimization can in this way be transformed to a constrained single objective optimization. If the relative importance of the cost functions subject to optimization is known a priori, weighing techniques can be used to reduce the dimension of the solution and the complexity of the problem. To determine the Pareto solution is computationally intensive but several methods suitable for calculation in parallel are available. Many are population-based genetic algorithm methods in which a set of solution candidates should be evaluated in each step. One such a method implemented in the "gamultiobj.m" implemented in the optimization toolbox of Matlab.

4.4 Algorithms for solving optimization problems

Engineering optimization problems are not seldom complex and time consuming to solve, especially if the problem is large-scale, nonlinear and of multicriteria type. The more parameters which are allowed to vary the harder the problem becomes to solve. However, more free parameters will usually improve the solution. Therefore it is good to optimize a system with respect to as many parameters as possible. If the cost functions are time consuming to calculate, such as when they are based on large-scale nonlinear dynamics simulations, the optimization problem might also take a long time to solve. Additionally, in many systems and in washing machines in particular, it could be necessary to evaluate the performance under different conditions. Problems of this type are normally addressed with a method according to Figure 5, which is an approach that can require much computational power to execute within a reasonable timeframe. This approach was used in Papers C-E.

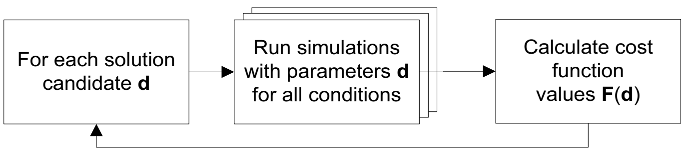

Figure 5: *The standard approach when solving optimization problems on multiple conditions, so called spinning operational scenarios, SOS*

All parameters are equally treated by the optimizer in the standard approach and will lead to a large search space for the optimal value. If engineering knowledge is added to the optimization problem solving algorithm then some smart selections of parameters to avoid conventional optimization of these parameter values can be done. It is assumed that in the washing machine there are values of parameters which can be found by solving other, auxiliary problems and that the values found will also be close to the values of the global optimum or at least can be a part of a set of parameters which produce similar performance to the performance of the global optimum. So, by finding values of some parameters by solving simpler auxiliary problems the search space can be greatly reduced. In the current washing machine application, the approach to reduce complexity of the optimization process is of the seen in Figure 6.

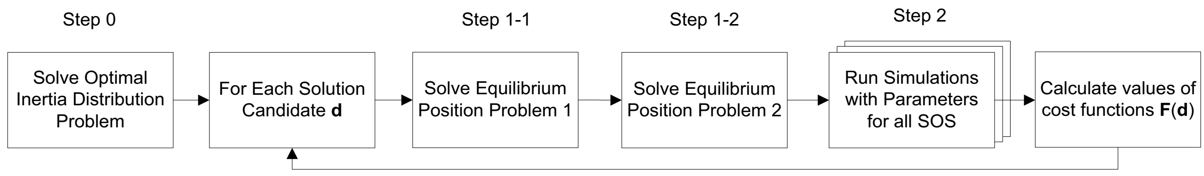

Figure 6: *The multi-step method for optimization of washing machine vibration dynamics*

The suggested method is executed as follows. If inertia functional components (like counterweights) are subject for optimization then the values of these parameters should be found by executing step 0. After execution of step 0 the found values are supposed to give optimal inertia distribution upon given restrictions of the parameter values. The found values are then taken as a input for the rest of the optimization steps.

An optimization algorithm is then assigned to solve the problem of finding preload and stiffness values for the machine main load bearing suspension springs. The found values of the parameters are determined by solving the equilibrium position problem twice, i.e. step 1-1 and step 1-2. Using the found parameters together with the remaining free variables as given values the simulation of the washing machine model can be started on all SOS. Upon simulation completion values of cost functions and constraints can be calculated based on the all the available results. This approach was used to solve some of the applied optimization problems described in Paper G.

4.5 Computational environment for optimization

The objective functions which are subject for minimization when Problem A is solved use the response from the simulated computational model describing the washing machine dynamics. As the model is implemented in MSC.Adams/View a Matlab-Adams\View communication interface was constructed. It enables a function in Matlab to start simulations and calculations in parallel and evaluate cost functions as the simulations are completed. For multiobjective optimization the interface is called by “gamultiobj.m” with a generation of solution candidates. The interface is compatible with other optimizer functions as well but it reaches its best performance when it is called with a set of solution candidates. In this way the feature of parallel calculation is enabled. The cluster is capable of evaluation of performance on a set of conditions. In the washing machine context they are called spinning operational scenarios, SOS. Also it is possible to make some parameters of some operational scenarios dependent on the result of other scenarios. In this way the method presented earlier is implemented. The steps 1-1 and 1-2 are defined to be executed by the cluster before the simulation according to step 2 is executed. In this way found spring properties are included in the simulation of the model.

In Paper G a flowchart of the working principle of the cluster controller is given together with details on how the cluster is connected to a user selected optimizing algorithm. Other details of the cluster and how it could be used are available in Paper D.

4.6 Applications of the design optimization method

The described methodologies have been used to formulate and solve several optimization problems related to washing machines. The problems have been formulated with the intention to find values and geometrical locations of functional components. In Paper C the position of a strut with given properties is investigated and optimized. The problem is of constrained single objective type with restrictions on its parameters. The used objective function is related to the vertically transmitted force and the constraints are imposed by the dynamic model of the washing machine and also kinematic displacement at several, by the manufacturer defined, critical points. Results from the optimization were used to construct a physical prototype of the washing machine by the industrial partner.

In Paper E several parameters of a given washing machine suspension type were subject to analysis and optimization. The machine in question, which was of new increased capacity tub design, had a four strut bottom mount suspension system and the study was executed with the intention to give a starting point to the engineers for further experimental studies. The optimization problem was of bi-objective type where the Pareto front of the objectives was sought. In the formulation of the problem the dynamic (force transmission) and kinematic (collision avoidance) objective functionals were used. Apposed were the dynamic constraints of the computational washing machine Adams model. Based on the results originating from this study a new machine has been put into production.

The focus aim with in Paper G is to present the full multi-step optimization approach. It was here used to find the parameters of a novel machine with even higher capacity and higher tolerance on possible imbalance loads that would be accepted during high speed spinning. The reasons for allowing higher levels of imbalance are to improve the number of successful spinning cycles compared to failed and improve the average result of spinning. The study of machine was conducted at an early stage where still much freedom on geometrical placement of struts and component properties were still available. Symmetric strut properties and placement around the tub center of mass was desired. To determine a good center of mass suitable for the whole load range of the washing machine also the counterweights were set as a subject for design. The properties of the counterweights were determined using step 0 of the method and a sensitivity study shown convergence, see Paper G. The preload and stiffness of the strut springs were also included in the set of variables subject for optimization. As the strut springs have a great impact on the equilibrium positions of the drum at empty and fully loaded machine respectively they were extracted from the set of variables available to the standard optimizer. The values of these parameters were instead selected to be determined during steps 1-1 and 1-2, first spring preload and later spring stiffness, for each solution candidate composed by the remaining parameters. The formulated optimization problem A3 was stated as a multiobjective optimization problem with dynamic constraints and restrictions on parameters. The objective functions were to be calculated based on multiple SOS. The dynamic cost function according to (4) was used to evaluate the results of all dynamic simulations of each solution candidate and then weighted together to form the dynamic objective value. The kinematic cost function (3) was used together with a set of general directions, as the washing machine subject to the study was still only on the virtual stage and not built physically. The worst case of all dynamic simulations of a solution candidate was selected to provide its value

to form the kinematic objective value. For the first time, a walking constraint based results of on the dynamic simulations was used for optimization. The necessary and sufficient formulation of walking avoidance was used to calculate the corresponding friction coefficient. The worst value calculated for all SOS of a solution candidate was selected as the solutions candidates walking performance. A constraint was formed by requiring that this value had to be smaller than a given friction coefficient. The given coefficient had been determined earlier by experiments to μ_g=0.17. Also constraints on tub positions at full and empty loads were imposed, but solved easily with step 1 of the multi-step method.
A Pareto front and the corresponding Pareto sets which constituted the solution to the multi objective problem were determined by using the developed computational cluster. The results were studied with respect to sensitivity of parameter values. The idea was to provide smooth functions describing the values of the parameters which would give the Pareto optimal performance. The estimated washing machine design configurations were evaluated and their performance was plotted along the Pareto front. It was shown that for this example the fitted polynomials could provide sufficiently good estimations of necessary variable values for almost the whole front.

5 Technologies for vibration control

5.1 Conventional damping

Dampers for washing machines are usually of passive friction type, where the force is generated from a prestressed sponge or fabric sliding inside a cylinder or housing, see Figure 7. The passive friction damper is a low cost solution, but performs decently. A high force response is achieved from using friction damping even though the relative velocity between the piston and damper cylinder is low, meanwhile the force is kept relatively low at high relative velocity.

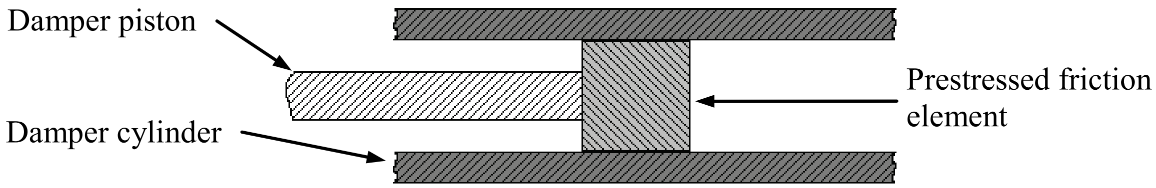

Figure 7: Sketch of a typical friction damper

For an ideal friction damper the force would be constant, but experiments have shown that it is wise to assume that some velocity dependence is present (Figure 8), even though this is undesired from the designer's point of view.

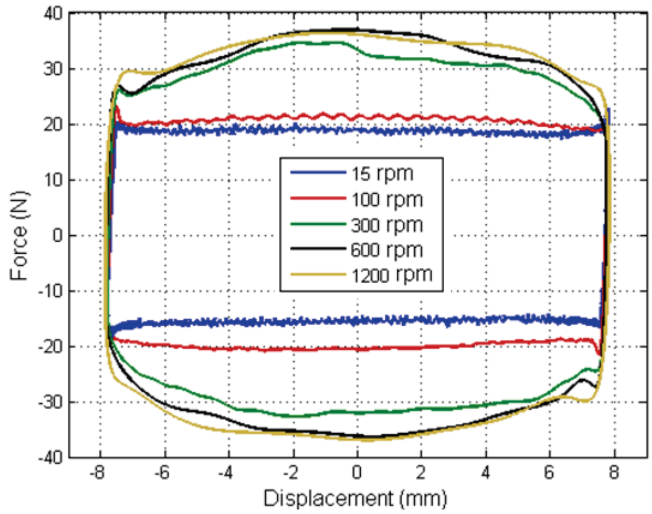

Figure 8: *Typical displacement force relation of a washing machine friction damper at different spinning speeds*

Ideal viscous dampers (e.g. increasing with velocity) will not perform well in washing machines as the damping force in this type of dampers increases with the frequency of excitation, even though the damping is only needed at low frequencies. Ideally, a damper with high damping at low frequency and low damping at high frequency would be used. Such dampers which suits washing machine cost and size limitations have not been made available yet. Thus, the friction damper is today still the most common choice due to cost and to the acceptably low dependency on excitation frequency and velocity.

5.2 Free-stroke dampers

One other approach to reduce the damping when spinning at high rotational speed is to introduce displacement depending damping. Examples of this are the so called free-stroke dampers, 2-stage dampers, or gap-dampers. When spinning over a certain spinning speed the displacement of the tub is normally small. The idea with this kind of damper is exploit this behavior and to let the piston move freely inside the damping cylinder without generating damping force, under these conditions.

Dampers of different types exist on the market today, for example the dampers provided by washing machine component manufacturers Suspa [48] and Aweco [49]. Technologies from these manufacturers and have been implemented in washing machines from for example Bosch-Siemens and LG. The design of the dampers of this type available on the market is a bit more complex than the conventional damper design hence more expensive to manufacture. The performance of a machine with a well tuned suspension with gap damping outperforms a conventional damper based system in terms of force propagation according to [20] and [50]. So it can be well motivated to incorporate such a damper if the dynamic criterion (4) is given high priority. In Figure 9, a sketch is shown of a free-stroke friction damper similar to one of the, on the market, available dampers.

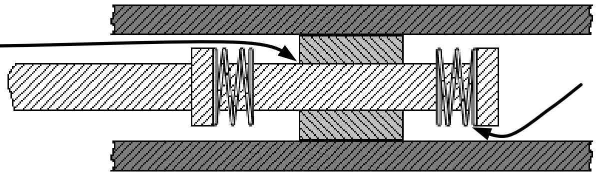

Figure 9: *Sketch of a free-stroke damper*

Additionally, within the department of the author, several experiments have been done on low cost prototype gap damper to study the performance of a machine equipped with a gap damper device. The results from the study confirms that a high potential of force propagation reduction exist if the size of the gap is tuned wisely. A reduction of up to 40% was found possible with the used low cost prototype, which if manufactured would comprise no extra parts compared to the conventional damper. Characteristics of the prototype damper developed at the department can be seen in Figure 10. Note that the data is taken from gap damper of another type and design than the device seen in Figure 9. A study of the developed damper is published in [50].

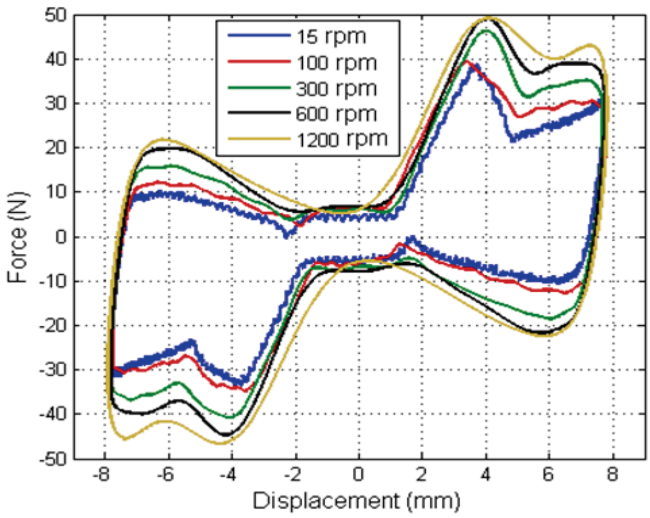

Figure 10: *Displacement force relation of a prototype gap damper at different spinning speeds*

5.3 Semi-active damper control

In washing machines the demands on a suspension are limitation of movement of the tub, which require high damping when passing resonances, together with low vibration output levels, which require low transmissibility at high speeds. Additionally the prevention of walking must be accomplished.

A passive suspension solution is often as in the case of simple friction dampers a low-cost compromise between opposing requirements on a vibrating system, a compromise which produces acceptable performance of all criteria. Without the compromise and limitation of an actuator with constant parameters an active or semi active solution is possible.

Two possible concepts of active/semi-active suspension of washing machines are close at hand:

1. Active feet,
   i.e. active/semi active components mounted between the housing and ground.
2. Active tub suspension,
   i.e. active/semi-active components mounted between tub and housing.

Currently, the focus has been set on the latter concept. As can be seen in Figure 2 several functional components constitute the tub suspension. Simulations and experiments have shown that the bushings are of significant importance for the movement of the tub. Incorporation of bushings of active type could be interesting to test, but is considered complex and expensive. The cost of additional more advanced components is often the reason for not using non-passive solutions. Instead, the translational passive damper is a better candidate to exchange for a damper of semi-active or active type.

In paper A, the potential of a semi-active suspension solution of this type is investigated with help of models the washing machine. The suggested solution showed a big potential for reduction of force output. An experimental study of the use of semi-active damping in washing machines has therefore been executed. The washing machine used was a three-strut washing machine of type Cylinda FT58. It has two struts with equal properties on the right side facing the front of the machine. On the right side a strut, which has had the damping element removed, been placed. In parallel with this a magnetorheological, MR, damper of type RD1097 from Lord Corporation [38] is fitted, see Figure 11.

The MR damper is mounted with the intention to limit the amount of torque interaction at its reaction points. The reason for this is that the bushings which are incorporated into the damper are quite stiff and there exists a risk of breaking the damper by bending it excessively. Bending is likely to occur when dynamic imbalance is present and tub motion in *y*-direction is expected.

With the selected suspension configuration requirements on forces for low vibration output demand that the damping should be low when the system is not in a state of resonance. This will reduce the force propagated through the suspension. Demands on high damping exist in state of resonances with large relative movement between tub and housing, to limit tub movement. The idea used is that if the tub system will remain stabile even though the damping is turned off, the force propagated through the suspension will be less. This strategy was used in Paper A.

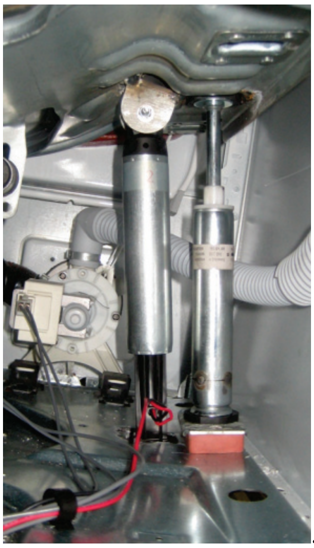

Figure 11: *The semi-active damper mounting seen from the back of the machine*

It has been stated earlier that critical maximum motion occurs at low speed when passing resonances of the tub system, see also Paper F. After passing the critical resonance the tub stays relatively centered, if the angular acceleration of the drum is constant. This shows potential for a switching solution for the damping, i.e. that the damping should be turned off when the drum is rotating faster that a certain speed. This approach is confirmed in [39] where the vibration performance above critical spinning speed is best with no dampers attached to the system.

A scheme for the control voltage u_d of the semi-active damper was defined according to the following equation.

$$u_d = \begin{cases} U_{on} & \text{if } \omega < \omega_s \\ U_{off} & \text{if } \omega \geq \omega_s \end{cases} \tag{6}$$

Here U_{on}=U_g V, U_{off}=0 V, ω is the current rotational speed of the drum and ω_s is a selected switching speed. The value of U_g corresponds to the amount of damping applied to the system and it is clear that it will affect the dynamics of the washing machine. The value must be tuned for good performance. This can be seen for example in Figure 12 and in Figure 13 where the displacement of the tub, and vertical forces measured at the machine feet are plotted for different values of U_g. In Figure 12 only the first part of the data is shown for greater clarity. At t>25 s only minor changes of the motion is registered and the tub trajectory stays practically unchanged. A change in rotational acceleration of spin speed has an effect, but the effect has been determined to be small if the change is smooth.

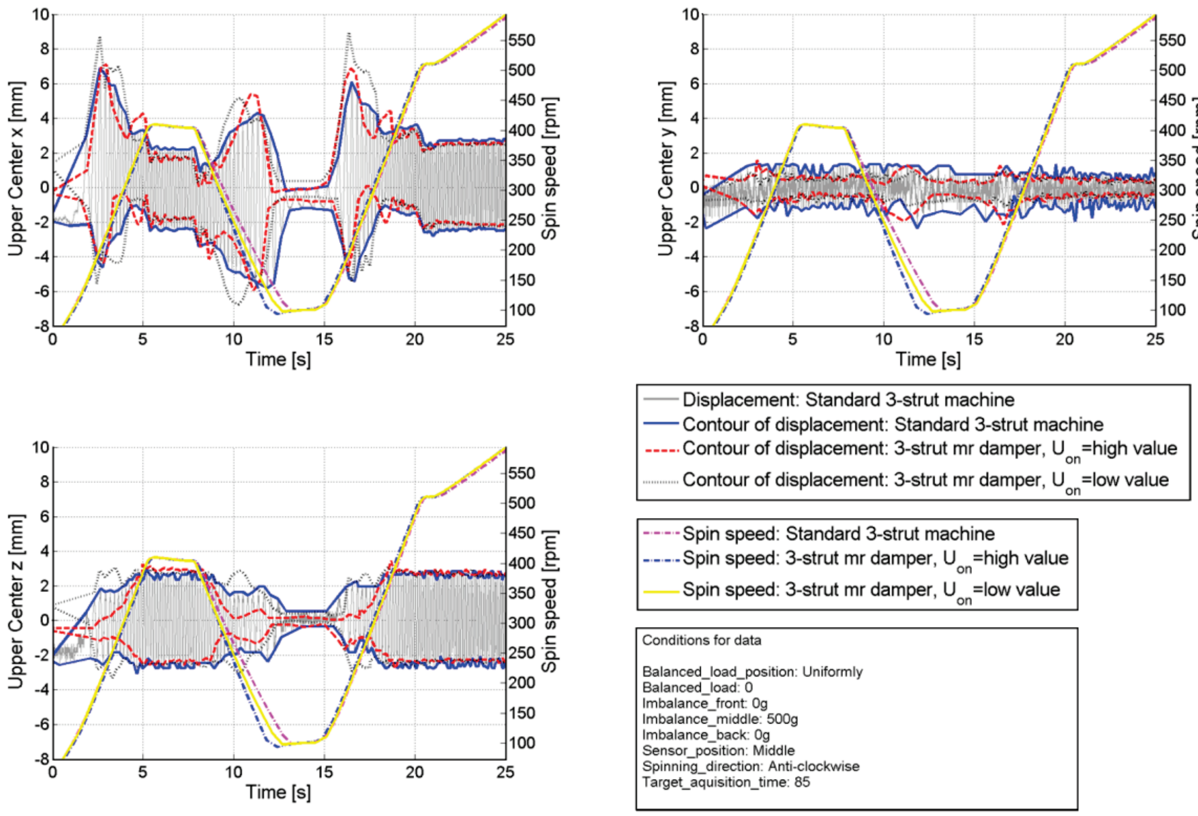

Figure 12: *Displacement in x-, y- and z- directions for passive and semi-active suspensions*

The imbalanced load used in the experiments is spread evenly along the extension of the drum such that it only should create static imbalance. Static imbalance causes very little displacement in the *y*-direction and therefore reduces the complexity of the movement for easier analysis of the performance of the MR damper. In the *z*-direction higher damping clearly reduces the movement, which could be explained by the mounting orientation of the damper. The peak to peak movement in this direction is however not much different to the case of with standard passive damper equipped machine as the extremas occur when $u_d = U_{off}$. Hence, for a range of high control voltages the peak to peak values are unaffected. This can be seen also in Paper A. There exists however a risk that higher imbalances together with low damping could produce more movement in *z*-direction when $u_d = U_{on}$, as indicated by the movement peaks in *z*-direction at t=3 s, t=10.5 s and t=17 s.

The damping also affects the movement in *x*-direction, even though the orientation of the damper implies that only small forces in this direction should be generated by the damper. To conclude it could be said that with the low amount of damping, the peak to peak movement is 24% larger than with the passive suspension. For the high damping case, the peak to peak movement is practically the same as with passive suspension. The difference is only 2.5%, or less than 0.5 mm, which could be considered to be within measurement margin. The larger values of peak to peak motion in the *x*-direction compared to the *z*-direction lead to the conclusion that the damping may be more beneficial for motion limitation if more of it is applied in the lateral direction. This conclusion could also be drawn from the simulations in Paper A, where a more inclined damper seems to be beneficial. One drawback could be the effect on walking performance.

A comparison of force output is shown in Figure 13. It shows that the amount of damping highly affects the amount of force propagated out of the machine and onto the floor. As the damper is mounted on the left side of the machines, the force reduction when the damper is switched off is greatest there, but some reduction is also visible at the other side at certain rotational speeds. Reduction of force output is achieved at almost all spin speeds for left side of the machine, where the amplitude is reduced at most speeds with as much as 40-50%. An increase in force output can be seen at speeds below $\omega<\omega_s$ when U_{on} is high, i.e. when much damping is applied. The timeframe in which damping would be added is however only a few percent of the total spinning sequence, which could last over 10 minutes. It could maybe be considered acceptable to have an increased force output for a short time if the output force is lowered for a longer time. Also, the extreme values for the forces at rotational speeds below ω_s are lower than the extreme values at rotational speeds above ω_s.

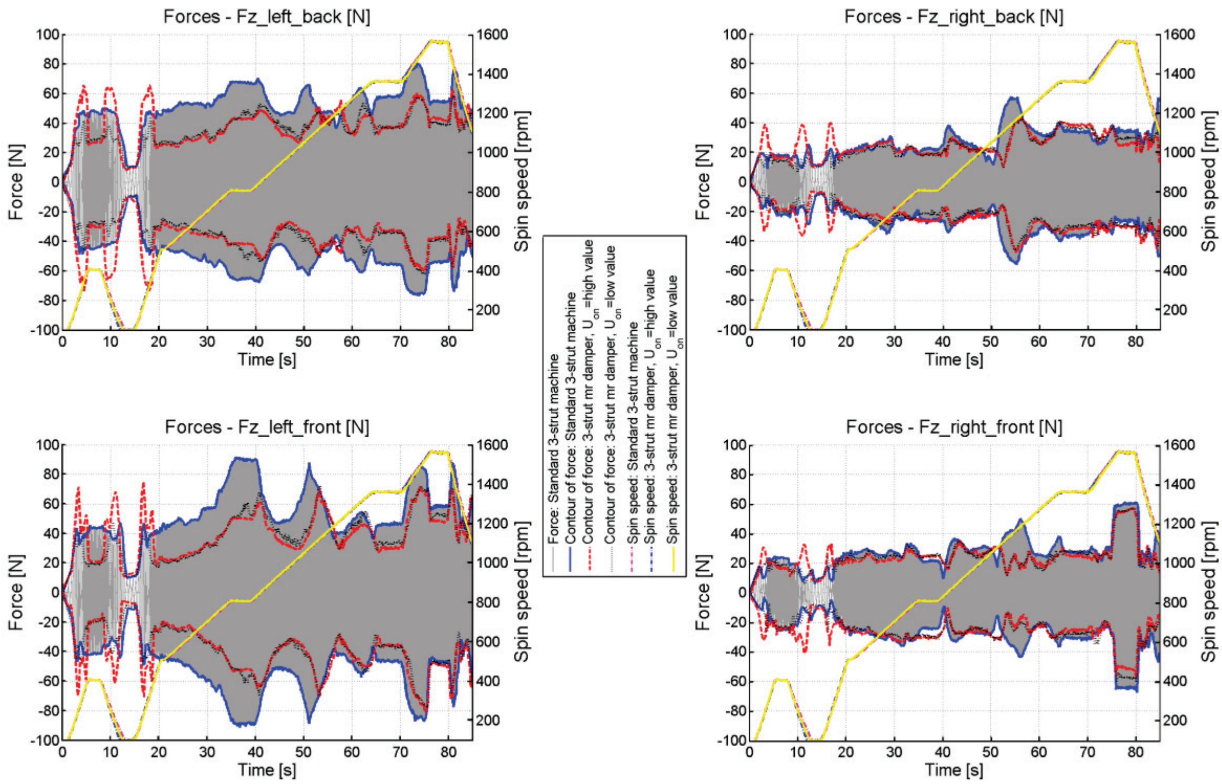

Figure 13: *Vertical output force for passive and semi-active suspensions*

The function Γ described in paper C gives the 47% reduction for damping with the "low value" and 41% reduction of vibrations with "high value". The direction of rotation gives the bigger amplitudes on the left side of the machine so clearly it is advantageous to control the damping on this side. The same reduction is not believed to be achieved if the damper would to be mounted on the right side, if the spinning direction is kept anti-clockwise. Furthermore, it has been shown that one damper is enough for efficient semi-active vibration control with the model of damper used [39] if force propagation is considered. In the referenced paper it is even shown than one single side mounted damper can perform better, in terms of vibration propagation to the housing, than two dampers. It is however unclear in the paper what effect a single slide mounted damper has on margins to collision, and to walking compared to a suspension with several MR-dampers.

5.4 Counterbalancing

The main part of vibrations in washing machines is caused by a poorly distributed wash load around the circumference of the drum. Other reasons can be poor centricity or other asymmetries of the manufactured parts. The imbalance can be of different types; static and dynamic. Static imbalance means that the position of the combined center of mass of the rotating system (drum and imbalance) is not located on the axis of rotation of the drum, but that one principal axis of inertia remains parallel to it, see Figure 14a. Pure dynamic imbalance does not change the position of center of mass of the system but will result in that none of the principal axis of inertia no longer is parallel to the spinning axis, see Figure 14b.

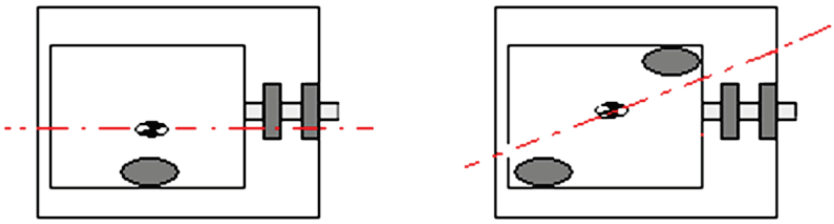

Figure 14: *Sketches showing center of mass and inertia axes at: a) static imbalance and b) dynamic couple imbalance*

In reality the most probable case is that both static and dynamic imbalances are present at the same time. Together they produce a two plane imbalance, also referred to as a static-dynamic imbalance. Pure dynamic imbalance is also referred to as dynamic couple imbalance.

The principle of counterbalancing is exactly what the name indicates, namely placement of additional bodies in such a way that the combined center of mass and one principal inertia axis of the rotating bodies is located on and coincides with the spinning axis respectively. If only static imbalance is present, then counterbalancing is necessary in only one plane. The plane being the same as the imbalance is present in, which has a normal parallel to the spinning axis. Counterbalancing in two planes is necessary to counteract a general static-dynamic imbalance.

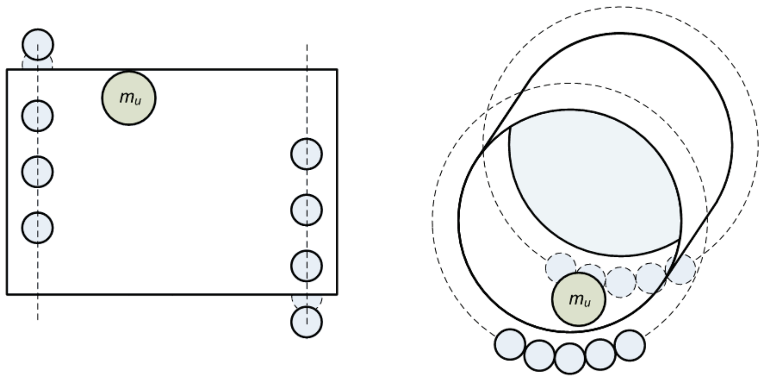

Figure 15: *Placement of balance rings in a washing machine in two planes*

The washing machine MBS model was extended by incorporation of two passive automatic balancing rings according to Figure 15, similar to the ring used in for example [34]. In the figure the locations or the balancing balls are displayed at two different locations. On the left side the balance balls are spread out to show a dynamic (not equilibrium) condition and on the right at rest during static equilibrium conditions. In Paper F results from simulations on variation of the parameters driving the balls around their trajectory are presented. The results show that it is possible to approach an elimination of the imbalance at overcritical speeds, whilst maintaining equal motion magnitude of the tub center of mass during passing of resonant frequencies.

All the above described technologies, i.e. constant of displacement dependent friction damping, semi active control or counterbalancing, can be used in washing machines to counteract the effect of the rotating imbalanced mass causing vibrations and large motion of the tub during the spinning process. Even though some of the technologies are powerful as they are a combination of several might be even more successful in terms of vibration related performance in washing machines. The combination of a free stroke damper, with relatively small free-stroke gap and high damping outside the gap, together with a two plane automatic balancing ring could for example take vibration levels both in terms of the kinematic and dynamic cost functionals (3) and (4) down to really low levels. Left is however the stability issue (walking) which still might be a problem if not taken into consideration properly in the suspension design process.

6 Summary of appended papers

Paper A

The results of modeling of dynamics of a washing machine and designing of a smart suspension system comprising magnetorheological dampers are presented. The commercial software Adams/View and Matlab/Simulink have been used for 3D-modeling of dynamics of a washing machine. Both a suspension with passive dampers and a suspension with magnetorheological dampers have been implemented into mathematical models. The models have been used for study of the dynamics of washing machines, design and optimization of a smart suspension system. Analysis of the obtained results has proved a potential of utilization of magnetorheological dampers for suspension systems of washing machines.

Paper B

The commercial multibody system software Adams was used to develop a model from design drawings. A test rig comprising sensors measuring transmitted force, as well as accelerations and movement of the drum has been built and is described in detail. In the paper results from both experimental and modeled data are presented. The conditions for the performed experiments and simulations were spinning from 0 to 1400 rpm with an imbalance of 0.3 attached to the drum wall. Comparison of obtained results of simulations and measurements show good agreement of tub movement and agreement on level of force output under the tested conditions. Resonances have been found, modeled and their sources have been identified. The biggest resonance has been clearly recognized and identified to be at between 10.5 Hz and 12.5 Hz giving dynamic force amplitude of 40% of the force at static conditions.

Paper C

The paper presents a rigid body model of a front loaded washing machine implemented in 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.

Paper D

The paper describes an Adams/View-Matlab environment for parallel/clustered calculations running on ordinary workstations. This inter-software communication environment was developed for sensitivity analysis and bi-objective optimization of washing machine performance by using a dynamic model built in the commercial multi-body software MSC.Software Adams\View. Together with statistics of performance of the system, results of a bi-objective optimization of a selection of structural parameters are presented.

Paper E

This paper focuses on several aspects of vibration dynamics in washing machines: the capacity maximization through the study of tub movement, the vibration output from the machine to the surroundings, and the “walking” tendency of the system. Three objective functions related to kinematics and dynamics of washing machines have been defined and a numerical algorithm has been created to solve Pareto optimization problems. The algorithm is a genetic algorithm built around Matlab’s subroutine “gamultiobj.m” and executed on an in-house developed computer cluster with possibility of parallel computing of Adams/View models. The results are presented as optimized parameter values of suspension functional components, in this case bushings with respective Pareto fronts. The focus has been set on delivering couplings between parameter values and performance trade-offs in terms of objective functions to facilitate parameter tuning. The obtained optimization results have successively been used in the development of a novel washing machine which went into production after the summer 2010.

Paper F

In this paper a model of a front loading washing machine of domestic type is presented. The model has been built using a theoretical-experimental methodology based on separate modeling of functional components for incorporation into a commercial numerical environment. The implementation of the models has been done in a modular way to enable easy switching between suspension configurations and production model types. Numerical studies are conducted to show the impact of different suspension parameters on dynamic and kinematic features of the washing machine. Experimental data are compared with output data from various models of individual suspension components based on time history of force and to energy dissipation per cycle. The different models of the strut friction damper component show relative force prediction errors down to 8.5% and relative energy dissipation errors down to 5%. Different constructed models of the washing machine feet are also compared with measurement data showing prediction errors down to 5.6% and 13 % respectively. The complete model of the washing machine has then been assembled, based on models with identified parameters describing functional components, and validated giving good agreement on tub motion and on foot vertical force at spinning speeds below 500rpm. As an application of the model, it has been extended with an automatic balancing device to investigate the potential of the counterbalancing technology in two planes for vibration output reduction at supercritical speeds. Simulations of the two plane balancing device show a great potential in eliminating resulting the unbalanced load at supercritical speeds whilst passing of resonances can be done with preserved levels of tub motion compared to the standard machine.

Paper G

In this paper a multi-step approach for optimization of washing machines structural components with respect to different cost functions is presented. In order to deal with the multiple conflicting operational conditions the term critical Spinning Operation Scenarios (SOS) is introduced as a general synthesis for definition of load configurations and corresponding rotation schemes of the drum. Different spinning operational scenarios are defined and used as conditions upon which performance shall be evaluated.

The dynamic vibration characteristics of the washing machine are channeled into three principal cost functions measuring collision margin between internal parts, force propagation, and the risk of walking.

The different cost functions are used as objectives and constraints in three different applied examples to optimize suspension designs for new and existing washing machines, showing the working principle of the multi-step approach. Some of the results derived from the specific solutions to these optimization problems have been used in practical development of new washing machines and other results are going to be the starting point for further studies. The results of sensitivity studies on the achieved results with the focus on facilitation of usage of the optimization outcome for practical engineering usage are presented. For effective numerical computation of the respective cost functions a computational cluster was developed enabling parallel calculation of the response of different dynamic models implemented in MBS software Adams/View. The principal functionality of the cluster is presented together with a description on how the interface to dynamics calculation software and to the optimizer is built up.

7 Conclusions and outlook

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. With the help of optimal structural design and control it will be possible to construct more silent washing machines for higher wash loads within the same housing dimensions.

In this thesis a MBS model of a commercial frontloaded washing machine built for analysis of spinning related performance has been presented. The model has been built using a theoretical-experimental methodology with which experimentally validated models of functional components have been incorporated into a computational multibody system model.

The computational model was implemented in MSC.Software/Adams to enable use with a developed Adams-Matlab interface for clustered simulation and optimization. In the models CAD drawings available from Asko Appliances AB were used giving accurate mass and geometry data of the machine for the simulations.

A full-scale test rig for horizontal-axis washing machines comprising sensors measuring transmitted force of the machine, accelerations and movement of the tub and rotational speed of the drum, were built for validation of developed models. Comparisons of obtained results from simulations with measurements have shown acceptable agreement of drum movement as well as level of force transmitted to the hosting structure under the tested conditions.

Component test rigs were designed, constructed and used for experimental study of force-displacement characteristics of dampers and machine feet in the vertical direction. By using the different test rigs and optimization routines, estimation of the parameters of several damper and foot models has been done successfully. The different models of the strut friction damper component show relative force prediction error down to 8.5% and relative energy dissipation error down to 5%. The best of the found models for foot dynamics show relative prediction errors of 14.6% and 5.6 % respectively.

The developed models and the test rigs have been successfully used for dynamic analysis (eigenfrequencies, eigenmodes, force transmission) and kinematic analysis (tub motion) of a washing machine during spinning. Numerical simulations have also shown the important role in quality of performance of suspension systems in washing machines dynamics that the suspension structural parameters play.

The vibrations originating from badly distributed load inside the drum have in this thesis been channeled into kinematic, dynamic and stability cost functions. The defined kinematic cost function deals with performance of tub motion and can ensure margins to collision of parts inside the washing machine or constitute a step in the process to increase the machine capacity. The dynamic cost function measures transmitted vertical force to the hosting structure. Forces which cause noise and vibration impact on the surroundings. Two different cost functions for stability of a washing machine in the sense of walking avoidance are also presented. They are based on sufficient, and the necessary and sufficient criteria of system stability. These cost functions have been used as objectives and as parts of constraints in formulation and solutions of various multiobjective optimization problems.

To facilitate the solution process of the washing machine optimization problems a multistep approach has been developed and presented. The approach divides the general large-scale multiobjective optimization problem into smaller problems based on specific requirements of the system such as esthetics and on engineering knowledge. This substructuring makes it possible to find certain parameters such as counterweight masses, strut spring preloads and stiffness if desired combined center of mass, desired principal axes of inertia as well as desired positions of static equilibrium are specified. One advantage with formulating optimization problems for the finding of preload and stiffness parameters of strut springs, instead of explicit calculation, is that more often results may be found. This usually increases the robustness of the whole optimization process.

To calculate values of the cost functions and efficiently solve the considered optimization problems a computer cluster has been developed. Its functionality has been described and the developed interface to dynamics solver program is presented. The cluster has been successfully used for parallel evaluation of the performance of washing machines with different suspensions.

Three applied optimization problems have been stated and solved by using the developed computational models of front loaded horizontal-axis washing machines. The obtained results of the solution of the optimization problems have been used for development of novel prototype machines as well as for studies on washing machines close to production status.

The developed MBS models of washing machines have also been used for studying different technologies applied for vibration control like passive suspension dampers, automatic balancing rings and magnetorheological fluid based variable damping. The feasibility of a two plane automatic balancing device for vibration reduction in washing machines was shown. A parametric study was done and the sensitivity of the system performance with respect to the viscous ball-driving force model's parameter of the balancing device has been shown. A lower limit for the viscous coefficient of the driving torque to ensure robust solutions was determined. Automatic balancing rings have shown great potential for vibration reduction at overcritical speeds. It is believed that with such a device implemented together with a passive free-stroke damper with high damping in its damping zone could be a good solution improving both kinematic and dynamic criteria. Some of the studies have also been done on semi-active control in washing machines. By using a spinning speed based on-off control algorithm for the tub suspension damping the amplitudes of vibration output has been found possible to reduce by 40% whilst preserving maximum motion on a given measurement point. This semi-active control strategy has been validated with experiments.

7.1 Contributions

The main contributions of the work within this thesis are:

- Development of computational model of a horizontal axis washing machine by using a theoretical-experimental methodology consisting of integration of multibody system formalism, detailed modeling of machine functional components, experimental data based model validation, and its implementation in the commercial MBS environment ADAMS/View from MSC.Software.
- Development of performance criteria for evaluation of washing machine kinematics, vibration isolation and stability and formulation of various multiobjective optimization problems regarding washing machine design.
- A proposed multistep approach for solution to the formulated optimization problems for washing machines, design and implementation of the numerical environment for clustered multiobjective optimization with parallel computation by using the developed MBS models of washing machines and Matlab/Adams graphical user friendly interface for pre and post processing.
- Kinematic and dynamic analysis of vibrations in bottom mount washing machines by using computer simulations as well as by experiments using the developed test rigs and the software for data acquisition and data processing.
- A validated control algorithm for semi-active control of washing machine tub suspension.

7.2 Future research and recommendations

The washing machines will be even more mechatronic in the future than they are today. More machines having automatic dosing of detergent and water will see the market, maybe even some with sensing the cleanliness of the clothes. The dosing of water and detergent is now done based on displacement sensors weighing the load. This hardware available inside the washing machine might be used to prevent excessive vibration in case of a failure in the imbalance detection algorithm. The usage of these sensors to detect tub kinematics for the estimation of imbalance amount and imbalance position is interesting. Measurements of the kinematics of the tub have in preliminary studies shown potential for use in identification of load amount and imbalance. With more detailed information on imbalance available it might be possible to improve the load distribution algorithm. This information about the load condition might also be used to enable feedback in the spinning process. It is an interesting subject which even here can help to save resources. Apart from saving time by spinning faster or detecting when the clothes are dry, energy can be saved if the clothes are dried further with help of spinning instead of with hot air coming from an electric heater.

With increased capacity of washing machines the possibility of incorporating effective tumble-drying capability with loads that are common today will be possible. Research with the aim to increase the tub volume can help this type of machines grow. To push the capacity of the machines the modeling and optimization approach described in this paper can be used as a tool, a tool that also keeps other aspects of the vibration dynamics of the washing machines within acceptable limit.

The stated aim with the optimization formulation given in this thesis has been to give solutions for suspensions that consider the defined cost functions. The formulation is given to ensure good (optimized) or acceptable (constrained maximum) values with respect to the different performances. Optimization problems defined in this way have been solved with respect to conventional suspension component and passive technology. Although semi-active control strategies, gap dampers and balance rings have shown promising results regarding vibration propagation, and some of them also with respect to collision avoidance, none of these technologies have still been studied with respect to stability in the sense of walking. This is something that should be done to ensure optimal or acceptable operational performance in new generations of washing machines.

The main motivation of semi-active control of the suspension damper for washing machines have been the potential the technology has to lower transmitted forces to the hosting structure or to reduce acceleration of the housing of the machine. In this thesis also the added kinematic behavior of the drum was given importance. To accomplish an improvement compared to the current suspension configuration, an on-off algorithm based on rotational speed has been found to work well. It would also be interesting to see the effect of a control algorithm on the third presented criterion, namely walking. If the damping forces could be varied during the rotational cycle of the drum the forces affecting vertical and lateral forces variations could maybe be controlled to some extent, and walking performance improved.

Simulations have shown that the washing machine model output is dependent on to the functional component model used to describe the strut damper. Therefore, studying the sensitivity of different damper functional components, in particular their stick-slip behavior with respect to walking and kinematic performance would be of interest. For example, the finding of a damper which has optimal characteristics in terms of reducing risk of walking could be an outcome of such a study.

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