

CHALMERS



Design and Analysis of Novel Low-Cost Damper

Application for suspension system of washing machines

PABLO ROJO GUERRA

Department of Applied Mechanics
Division of Dynamics
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2009
Master's Thesis 2009:32

MASTER'S THESIS 2009:32

Design and Analysis of Novel Low-Cost Damper

Application for suspension system of washing machines

PABLO ROJO GUERRA

Department of Applied Mechanics

Division of Dynamics

CHALMERS UNIVERSITY OF TECHNOLOGY

Göteborg, Sweden 2009

Design and Analysis of Novel Low-Cost Damper
Application for suspension system of washing machines

PABLO ROJO GUERRA

© PABLO ROJO GUERRA, THOMAS NYGÅRDS, VIKTOR BERBYUK, 2009

Master's Thesis 2009
ISSN 1652-8557
Department of Applied Mechanics
Division of Dynamics
Chalmers University of Technology
SE-412 96 Göteborg
Sweden
Telephone: + 46 (0)31-772 1000

Cover:
Novel LCD prototype.

Chalmers Reproservice
Göteborg, Sweden 2009

Design and Analysis of Novel Low-Cost Damper
Application for suspension system of washing machines

PABLO ROJO GUERRA
Department of Applied Mechanics
Division of Dynamics
Chalmers University of Technology

ABSTRACT

Suspension systems used nowadays in washing machines to reduce vibrations go from conventional passive dampers to high-tech dampers based on magnetorheological fluids, which change their damping properties by using electronic control. Such mechanical system demands high and low damping mostly depending on current spin speed, thus a low-cost damper which could easy change its properties would be desirable.

The work presented in this thesis concerns the design, prototype development and analysis of a non-linear low-cost damper. Different damping scenarios and principles were studied and a final concept of the damper was proposed to be developed. Several prototypes of the proposed novel damper were manufactured. In order to analyze, validate the estimated “butterfly” behavior and compare with the existing serial passive friction damper, several laboratory tests were performed for different speeds and amplitudes using specified test rig. The measurement data were processed in LabView and MATLAB. From all the models, the best was selected based on damping capacity and dissipated energy. Performance of new suspension system based on proposed novel damper was experimentally studied by using washing machine test rig. Using the obtained experimental data and optimization methodology the mathematical model of proposed LCD was developed.

Key words: Suspension system, Washing machine, Low-Cost Damper (LCD), Development, Physical prototype, Experiment, Butterfly behavior.

CONTENTS

| | |
|---|----|
| ABSTRACT | I |
| PREFACE | V |
| 1 INTRODUCTION | 1 |
| 1.1 Aim of the thesis | 1 |
| 1.2 Methodology | 1 |
| 2 LITERATURE STUDY | 2 |
| 2.1 Control systems | 2 |
| 2.1.1 Passive systems | 3 |
| 2.1.2 Active systems | 3 |
| 2.1.3 Semi-active systems | 3 |
| 2.1.4 Adaptive systems | 3 |
| 2.1.5 Semi-active adaptive systems | 4 |
| 2.1.6 Predictive systems | 4 |
| 2.2 Vibration isolation mountings | 5 |
| 2.3 Overview of typical damper systems | 5 |
| 2.3.1 Friction Dampers | 5 |
| 2.3.2 Hydraulic Dampers | 6 |
| 2.3.3 Hydraulic damper with variable damping characteristic | 6 |
| 2.3.4 Electrorheological Fluid Dampers | 7 |
| 2.3.5 Magnetorheological Fluid Dampers | 7 |
| 2.3.6 Gas Damping System | 11 |
| 2.4 Washing machine description | 11 |
| 3 DESIGN SPECIFICATIONS | 13 |
| 3.1 Introduction | 13 |
| 3.2 Quality function deployment analysis | 16 |
| 3.2.1 QFD Analysis | 16 |
| 3.2.2 Sensitivity analysis | 19 |
| 3.2.3 Percentage of sensitivity | 20 |
| 3.3 Modification analysis | 21 |
| 3.4 Final conclusions | 24 |
| 4 CONCEPT DEVELOPMENT | 25 |
| 4.1 Available patents | 25 |
| 4.1.1 Friction Damper for Washing Machines | 25 |
| 4.1.2 Damper for Spin-Drying Washing Machines | 26 |
| 4.1.3 Damper for Washing Machine | 26 |
| 4.1.4 Frictional Damper, for Washing Machines with Spinning Cycle | 27 |
| 4.1.5 On-Off Damper System, Especially For Washing Machine | 27 |
| 4.1.6 Frictional Vibration Damper | 28 |

| | | |
|--------|---|----|
| 4.1.7 | Combination Type Damping and Washing Machine | 28 |
| 4.1.8 | Magnetically Actuated Motion Control Device | 29 |
| 4.1.9 | Frictional Damper for Spinner-Type Washing Machines | 29 |
| 4.1.10 | Damper of Drum Type Washing Machine | 30 |
| 4.1.11 | Damper in a Washing Machine and Fabricating Method | 30 |
| 4.1.12 | Damper and Washing Machines | 30 |
| 4.2 | Proposed concepts | 31 |
| 4.2.1 | First concept | 31 |
| 4.2.2 | Second concept | 31 |
| 4.2.3 | Third concept | 32 |
| 4.2.4 | Fourth concept | 32 |
| 4.2.5 | Fifth concept | 33 |
| 4.2.6 | Sixth concept | 33 |
| 4.3 | First stage of development | 34 |
| 4.3.1 | Conclusions | 37 |
| 4.4 | Second stage of development | 37 |
| 4.4.1 | Standard damper | 37 |
| 4.4.2 | Input data | 38 |
| 4.4.3 | Prototype model | 39 |
| 5 | LABORATORY STUDIES | 40 |
| 5.1 | Basic friction damper model | 40 |
| 5.2 | Prototype model | 41 |
| 5.2.1 | Schedule of studies | 42 |
| 5.2.2 | High Stroke Damping | 42 |
| 5.2.3 | Low Stroke Damping | 44 |
| 5.2.4 | Final analysis | 44 |
| 5.2.5 | Conclusions | 48 |
| 5.3 | Washing machine tests | 49 |
| 5.3.1 | Schedule of tests | 50 |
| 5.3.2 | Analysis of results | 52 |
| 5.3.3 | Conclusions | 64 |
| 6 | MATHEMATICAL MODEL | 65 |
| 7 | CONCLUSIONS AND RECOMMENDATIONS | 68 |
| 8 | REFERENCES | 69 |

Preface

This master thesis is the final part of the Industrial Engineering Degree that was followed at Universidad de Oviedo, Spain. The work in this thesis has been carried out as ERASMUS student during the period January 2009 to June 2009 at the Department of Applied Mechanics at Chalmers University of Technology.

I would like to express my gratitude to my supervisor and examiner, Professor Viktor Berbyuk for giving me the opportunity to work on this project and also for his support and guidance. I am very grateful to my adviser, Thomas Nygårds for his help during laboratory simulations and his support during this project.

Furthermore, I would like to appreciate the help of Jan Möller for his help manufacturing the prototype.

Finally but not least, I would like to express my heartfelt gratitude to my family, specially my parents for their uncompromising love and support.

Gothenburg, June 2009

Pablo Rojo Guerra

Notations

Roman upper case letters

| | |
|------------------------|---|
| F_R | Friction force |
| N | Normal force |
| L | Housing length |
| E_1 | Housing pad width |
| C_1 | Contact surface of pad 1 |
| C_2 | Contact surface of pad 2 |
| C_C | Contact surface during compression |
| C_E | Contact surface during extension |
| C_{Cfinal} | Contact surface during compression, final value |
| C_{Efinal} | Contact surface during extension, final value |
| C_T | Contact surface, total |
| S_1 | Central position of housing pad |
| W_d | Energy dissipated |
| F_d | Damping force |
| $F^{sim}(\Delta, \xi)$ | Force vector from the simulations |
| $F^{exp}(\Delta, t)$ | Force vector from the experiments |

Roman lower case letters

| | |
|-------|----------------------------|
| c | Damping force parameter |
| k | Damping force parameter |
| l | Shaft length |
| l_1 | Pad 1 length |
| l_2 | Pad 2 length |
| h | Length between pads |
| e_1 | Width of pad 1 and 2 |
| g_1 | Relative position of pad 1 |
| g_2 | Relative position of pad 2 |
| s_1 | Central position of pad 1 |
| s_2 | Central position of pad 2 |

Greek lower case letters

| | |
|----------|-------------------------|
| μ | Friction coefficient |
| α | Damping force parameter |
| β | Damping force parameter |
| γ | Damping force parameter |
| ξ | Damping force parameter |

1 Introduction

Dampers of different kinds are used in many places in a wide variety of machines to reduce vibrations. Cheapest, and therefore most common, are passive dampers with constant properties, (if not degradation etc. is excluded). Unfortunately, in many cases the properties of the damper are determined as a compromise between demands of different desired dynamics of the system. One such case is in washing machines suspension where spinning demands high and low damping mostly depending on current spin speed. Therefore a damper which could change its properties would be desirable. Solutions exist, but they are relatively expensive compared to pure passive dampers. Sometimes the price of the damper is high because the damper has too much control properties, e.g. their damping properties can be set in many steps, and fast, etc. To keep the price down a damper which only would have two settings (on/off, or low/high) would be interesting. Also heavy nonlinear, but continuous solutions are interesting.

1.1 Aim of the thesis

The aim of the project is to develop a new non-linear Low-Cost Damper (LCD) for use in washing machines. Different concepts devices must be analyzed and the most promising developed, manufacturing a prototype and tested at several scenarios.

The project should result in knowledge about at least three different concepts:

- Dampers which change their properties, e.g. on/off with an electric signal
- Dampers which change their properties, e.g. low/high mechanically depending on frequency or speed of excitation
- Dampers which change their properties, e.g. low/high mechanically depending on amplitude of excitation or internal position of e.g. piston.

1.2 Methodology

The project was split into several phases. In the first phase, a wide literature search on the damper systems which are in use today was done. Second phase was dedicated to analyze the design specifications to focus the final solution, in which is included a Quality Function Deployment Analysis (QFDA). In the third phase, a concept development was done based on the results of the before phase. Once selected the best concept, a detailed configuration developing was done. At the last phase, the final prototype was manufactured and several tests were performed at the lab using Lab View and MATLAB software for data processing. Furthermore, a mathematical model of the new damper was developed. Final results and conclusions were exposed to close the work.

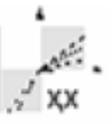
2 Literature study

2.1 Control systems

The first step to understand the damping systems is to know the various forms of control to regulate it. Said control devices are very significant on the behavior of systems, as well as their price, a very important aspect nowadays.

In general, four different control devices can be used. Conventional systems as springs and dampers are passive systems, which have a constant behavior. Semi-active suspension systems work with faster adjustable setting elements. These suspension systems can produce forces in the 1st and 3rd quadrant of the force-displacement and/of force-velocity diagram as can be seen in *Table 2.1.1*. The full active suspensions can produce forces in all four quadrants of the diagrams. In comparison to the semi-active suspension which just needs energy for the drive of the setting elements and the electronics, full active suspensions also need very high energy for the force actuators. Adaptive systems can set elements to realize an adaptation of the chassis performance to different circumstances. [1]

Table 2.1.1 Classification of control suspension systems

| | Forces | Switching Frequency | Energy Consumption | Model |
|-------------|---|---|--------------------|---|
| Passive |  | | |  |
| Adaptive |  | Smaller than the body natural frequency | little |  |
| Semi-active |  | Larger than the body natural frequency | Medium | |
| Active |  | Larger than the body natural frequency | High |  |

Depending on the purpose, it can be chosen five systems related to the way of control used: passive, active, semi-active, adaptive and predictive.

2.1.1 Passive systems

Passive systems are based in elements which offer a constant response, and differ from the others due to the unnecessary contribution of energy into the system. This make a simple system but it has a resonance at the eigenfrequency where the incoming vibration is amplified. By increasing the damping of the system the resonance amplification can be reduced. These systems are usually used for high frequencies. [2]

2.1.2 Active systems

In active systems the damper is replaced by an actuator, to introduce a force in contrast of the excitation force. With this is possible to remove the disturbed force and achieve the system hold as close as possible to its equilibrium position. The actuator is driven by the absolute velocity, so is normal to use accelerometers for this task. The energy consumption of this system is elevated. Another disadvantage is the complex actuators and sensors needed, and on the other hand this device has not a backed system in case of failure. [2]

2.1.3 Semi-active systems

Semi-active systems are located between passive and active. It is possible to change the physics properties of the device components, accuracy adjust of the elasticity k , and damping c to obtain the best isolation in each situation. The energy consumption needed is less than in active systems. This operation is based in change the properties (k and c) in each vibration cycle. Given an harmonic excitation of certain frequency, damping changes according the excitation frequency. [2]

2.1.4 Adaptive systems

Adaptive control systems are able to adapt to the changes that may occur in the system. The primary impetus for using adaptive control techniques is that often the systems to be controlled are noisy (i.e. are subjected to disturbances), or have unknown characteristics. Adaptive control methods for such systems are advantageous over fixed systems, because the controller parameters can be adjusted or tailored to the unknown and varying characteristics of the system.

Most often, the adaptation process occurs in real time at either a high rate (fast-acting systems) or at a slow rate (slow-acting systems). The adaptation rate is selected by the control designer, based on the system environment and performance requirements. [3]

Most adaptive control systems can be divided into two categories: feedforward adaptive controllers and feedback adaptive controllers.

Feedforward adaptive control uses a reference and an error signal to adjust the controller to adapt to the system changes or uncertainties. Thus, the effect of the measurable signals on the control systems behavior should be known a priori. The typical methods are gain scheduling and LMS adaptive filters. The disadvantages are that there is no feedback loop to compensate for unpredictable variations in the systems.

The feedback adaptive control involves automatic experimentation with these adjustments and knowledge of their outcome in order to optimize a measured system performance. There are two approaches for implementing the feedback adaptive control. One is the direct adaptive control that incorporates the plant parameters into the control law. The other is the indirect adaptive control in which the adaptation is performed based on the system identification of the unknown plant parameters. [4]

The choice of feedback versus feedforward adaptive control depends on such conditions as the availability of input and output signals, and computational capabilities of the microprocessors or DSP used in the system. Feedback adaptive control can be more effective in resisting the variations and disturbances, as compared to feedforward adaptive control techniques. [5]

2.1.5 Semi-active adaptive systems

In order to take advantage of the features of semi-active and adaptive systems, several studies in the past have discussed semi-active adaptive systems. The main effort of most of these studies has been on linear systems that use semi-active and skyhook control policy or its variations. A number of these studies performed have emphasized the application of such techniques as self-tuning control, model reference adaptive control (MRAC) or LMS adaptive filters to the semi-active systems based on the system linearization.

The vast majority of semi-active devices, however, are nonlinear. For instance, shock absorbers have a bilinear characteristic, and electro-rheological (ER) dampers and magneto-rheological (MR) dampers exhibit hysteretic nonlinearities. As mentioned earlier, although other past studies have addressed semi-active adaptive systems, none has addressed how to design semi-active adaptive control directly to deal with the nonlinearities of semi-active devices as subjected to immeasurable non-stationary vibration sources. [5]

2.1.6 Predictive systems

Predictive systems are the most complicated ways of control, due to estimation difficulties. Two main types of predictive systems exist: variable damping and variable elasticity.

Variable damping varies between minimum and maximum through control logic. The vibration response peak is higher than using passive systems, both of them with same maximum damping.

Using variable elasticity can be obtained a better response before impact. The system becomes softer during impact. This device is used to remove residual vibrations. Is needed reduce the stiffness at least an 80%, there are two ways to do that: using piezoelectric materials with low reaction times and low elasticity change capacity, or mechanical devices with high values in both of them.

The most important parameter to control the maximum response is elasticity. Whereas damping is used for residual vibrations and it is useful for amplification zone of impact spectrum, where its effects are more considerable.

2.2 Vibration isolation mountings

In a parametric study the impact of varying the attributes of an isolated system (damping, stiffness, and mass) on its vibration isolation effectiveness is analyzed. The results of this parametric study are summarized in *Table 2.2.1*. The table shows the positive (enhancement), negative (deterioration), and zero (no effect) impacts of change in damping, stiffness, and adding mass (to the isolated object) on the vibration isolation effectiveness.

Table 2.2.1 Impact of mount parameter variation on vibration isolation effectiveness

| | | <i>Effect On</i> | | |
|-------------------|------------------|------------------------|--|---|
| | | <i>Shock Isolation</i> | <i>Low-Frequency Vibration Isolation</i> | <i>High-Frequency Vibration Isolation</i> |
| <i>Parameter</i> | <i>Variation</i> | | | |
| <i>Damping</i> | increase | + | - | - |
| | decrease | - | + | + |
| <i>Stiffness</i> | increase | + | - | 0 |
| | decrease | - | + | 0 |
| <i>Added Mass</i> | increase | - | + | + |

Evident from the Table, no single solution enhances all the attributes of a mounting mechanism isolating an object subject to simultaneous vibration and shock loadings, e.g., a diesel-generator on-board a watercraft. For example, enhanced low-frequency vibration isolation performance of a system with low damping and low stiffness is normally achieved at the expense of excessive displacement of the mounted mass around the resonant frequency of the system diminishing its shock isolation effectiveness. [6]

2.3 Overview of typical damper systems

Dampers are used in several applications in order to dissipate energy, reducing vibrations. Some of the types of dampers are mounted in washing machines, specifically friction dampers as passive systems. But nowadays companies are adding new devices, developing new semi-active systems in which are used fluids which change its rheological behavior or smart materials adapted in real time, what means a price and complexity enhancement.

2.3.1 Friction Dampers

They are based in the friction between two bodies. They consist of an external housing, in which a coaxial piston rod is mounted, with a spring. At the end of the rod a friction material is placed. These are the cheapest dampers for the washing machine application, but with some limitations related to the system adaption against frequency changes, as the passage by resonance frequency. These are a clear example of passive damping, but some companies have already developed new friction dampers with semi-active behavior.

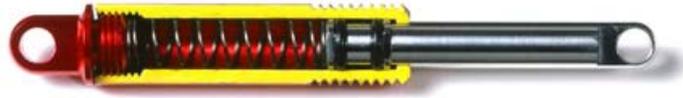


Figure 2.3.1.1 Friction damper with integrated spring

2.3.2 Hydraulic Dampers

These kinds of dampers use a fluid as friction device. They dissipate the energy by pumping oil through small orifices. Resistance of the hydraulic damper increases as the speed of spring deflection increases. They are formed by two chambers inside a housing, wherein a piston rod is mounted. The fluid passes from one chamber to the other. The gap between the two chambers makes the damper behavior. It is possible to add a semi-active control with a valve which varies the size of the gap. This valve is command by an electrical signal from a control unit. Although this system is used in several applications, is not common to see in washing machines.

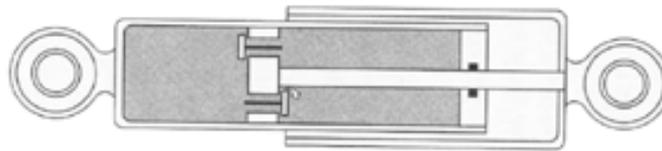


Figure 2.3.2.1 Basic hydraulic damper

2.3.3 Hydraulic damper with variable damping characteristic

Hydraulic damper with variable damping characteristic comprises a cylinder with a piston rod and a piston which divides the volume of the cylinder into two working chambers. The piston parts which move relative to the cylinder are provided with a passage arranged between the working chambers which can be opened or closed by a shut-off valve. The cross-section of the piston is less than the free cross-section of the cylinder, so that a gap is formed as a passage between the cylinder wall and the piston. The piston has a groove in the region of the gap which extends across the whole width of the gap. Arranged in said groove is an elastic expanded body which is curved like a tire in the direction of the cylinder wall and which closes the gap and seals the volume of the groove from the passage. Two channels, in each of which is arranged a shut-off valve with a passage, lead from each working chamber and open into the groove. [7] This method can be carried out in different ways, for example, using piezo-ceramic valve over the gap to control the flow of hydraulic fluid. [8]

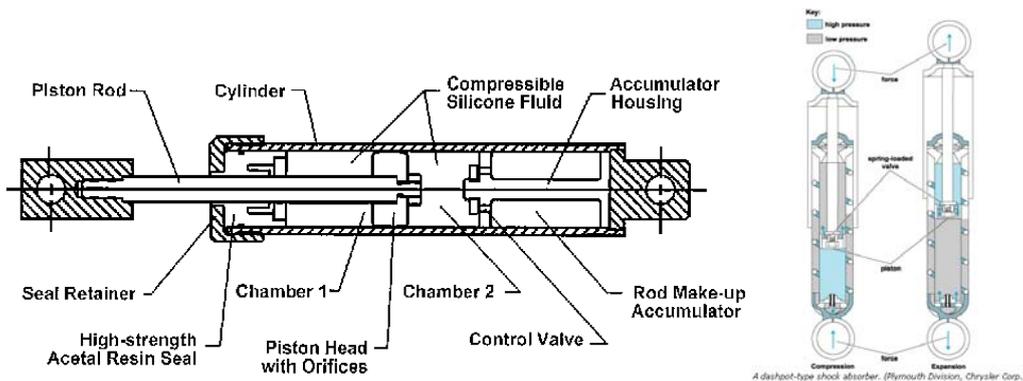


Figure 2.3.3.1 Two Hydraulic damper configurations with variable valve

2.3.4 Electrorheological Fluid Dampers

Electrorheological (ER) fluids are fluids which exhibit fast and reversible changes in their rheological properties under the influence of external electrical fields. ER fluids commonly are composed of polarisable solid particles dispersed in non conducting oil. Upon the imposition of external electric field, the particles are polarized and form a chainlike structure along the direction of the field. This structure is responsible for the rheological properties alteration of the ER fluid from a fluid like state to a solid like state which exhibits a yield stress. The reversible and dramatic change in the rheological properties of ER fluids coupled with the instant response offers wide potential in industrial application.

Basically, the flow mode ER-damper consists of a piston rod and a piston head in concentric cylinder tubes which also act as electrodes. If a certain level of forces given to the piston rod, there will be a pressure drop between upper section and lower section inside the annular gap and the fluid will move with a certain flux or velocity. [9]

ER fluid filled dampers can be designed in a different number of ways. The key fact is that the fluid flow properties in the damper can be directly controlled through the applied electric field, thereby avoiding the need for mechanical servo valves. [10]

The damper is made of hydraulic and pneumatic reservoirs separated by a floating piston. Inside the hydraulic reservoir, the piston rod is attached to a piston head holding two concentric tubular electrode lengths. During piston rod motion, ER fluids pass through the electrode gap between two concentric tubes in the piston head and can be energized by applied electric fields. The plastic spacer electrically insulates the two concentric tube electrodes to create an electrical voltage potential in the gap. [11]

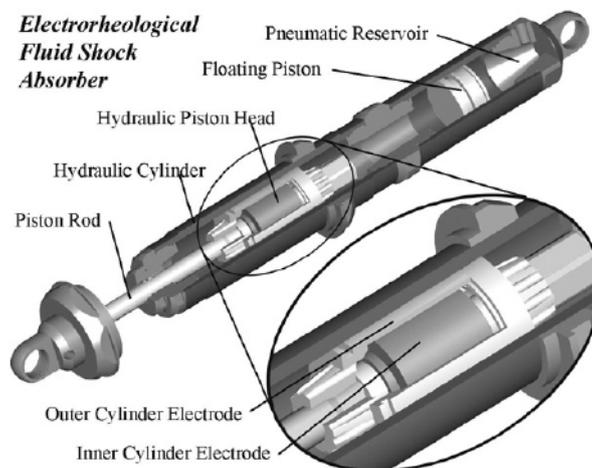


Figure 2.3.4.1 Schematic diagram of the ER damper

2.3.5 Magnetorheological Fluid Dampers

Magnetorheological (MR) fluids are materials that respond to an applied field with a dramatic change in the rheological behavior. These fluids' essential characteristic is their ability to reversibly change from a free-flowing, linear, viscous liquid to a semisolid with controllable yield strength in milliseconds when exposed to a magnetic field. A typical MR fluid consists of 20% to 40%, by volume, of relatively pure iron

particles, suspended in a carrier liquid such as mineral oil, synthetic oil, water, or glycol.

There are some ways to keep the liquid in damper applications. One of them is using a chamber inside the housing. Another is using a sponge in which the MR fluid is contained. The sponge technique eliminates any kind of seal and it is quite cheaper than the first. The MR fluid sponge damper requires no seals or bearings and uses the same inexpensive components found in existing passive dampers, but with a few important modifications. The damper consists of a layer of open-celled polyurethane foam, or other suitable absorbent matrix material, saturated with about 3 milliliters of MR fluid surrounding a steel bobbin and coil. Together, these elements form a piston on the end of a shaft. The piston is free to move axially inside a steel housing that provides the magnetic flux return path. Damping force is proportional to the sponge's active area. The amount of force produced is proportional to the area of active MR sponge that's exposed to the magnetic field.

While passing through resonance, these controllable dampers may be energized to provide a high level of damping. At high speed, the MR sponge dampers are turned off, enabling a high level of vibration isolation.

With this enhanced vibration control, the drum may be made larger (or the housing made smaller) because less overall tub motion must be accommodated. Ideally, a pair of controllable dampers provides 50-150 N of damping force when energized and a low residual force of <5N when turned off. [12]

Removing damping during the machine's high-speed spin phase decreases the motor's power consumption. This is an improvement over conventional passive dampers, which require the motor to work against their force to achieve the same spin speeds.

These kinds of dampers are the most advance technology used in washing machines. They offer one of the best behaviors in this area, but with high prices.

Damper force variation with current, displacement and speed:

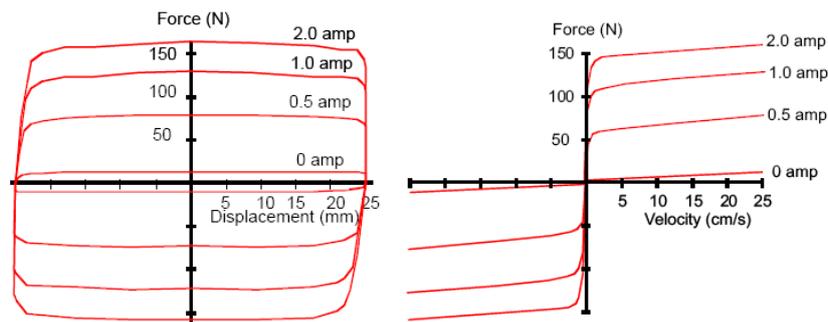


Figure 2.3.5.1 MR damper behavior depend on supplied current

The advantage of this system is that MR fluid damper provides a simple, cost-effective solution to the dilemma of providing high-level damping at resonance because they can simply be turned off at high spin speeds for a high degree of vibration isolation.

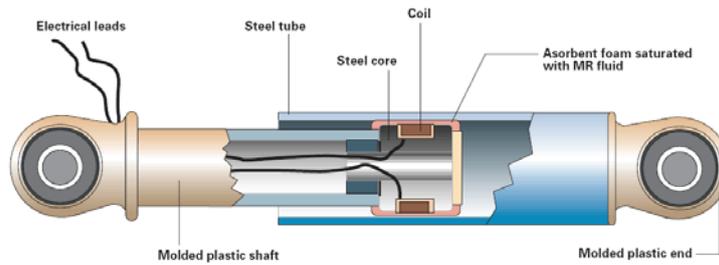


Figure 2.3.5.2 Schematic diagram of a low-cost MR damper

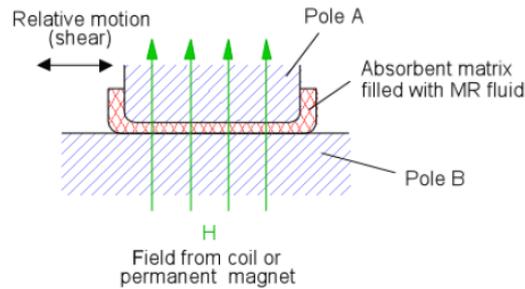


Figure 2.3.5.3 Elements of basic MR sponge damper

According to [3], MR sponge dampers can exhibit long life. Little wear of the sponge occurs as the stresses are primarily carried by the iron particle structure established within the MR fluid. Further, performance is rather unaffected by wear of the sponge. The role of the sponge is simply to hold MR fluid in the working gap. As long as the sponge is able to hold sufficient MR fluid to fill the working space region of the magnetic field, damping force is largely unaffected by the condition of the sponge.

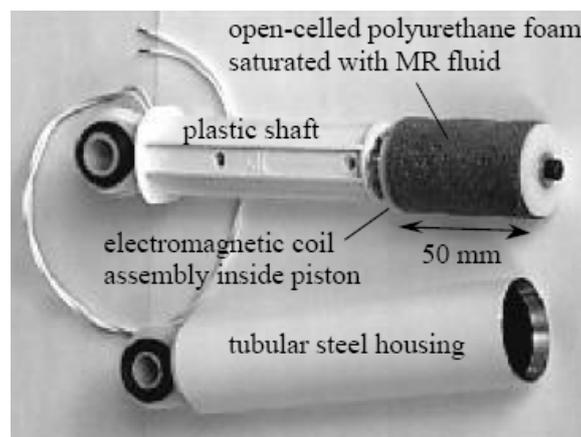


Figure 2.3.5.4 Real low-cost MR sponge damper

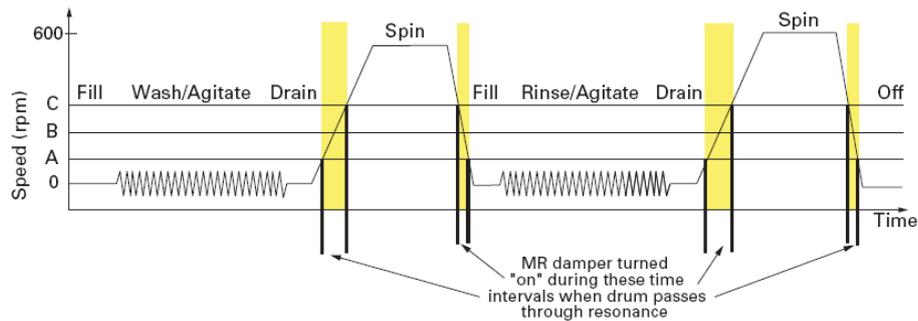


Figure 2.3.5.5 On-off MR damper work cycle

Many of the benefits of passive damping schemes built around MR technology are intuitive:

Efficiency

- Washing machines achieve greater performance in terms of higher spin speeds without the increased energy consumption of more powerful motors.
- Energy efficiency is enhanced because clothes come out drier.
- With heightened vibration control, tubs can be designed larger and the housing smaller.
- Machines can accurately weigh loads and thus control the use of water and detergent.

Functionality

- The damping system uses onboard electronics.
- No additional operator control is required.
- MR provides real-time controllability.

Cost

- Because existing materials are used, the slight increase in materials cost is balanced by improved energy efficiency.

System Integration

- Additional electronic controls are easily adaptable to the existing machine's electronics footprint. [3]

As shown before, nowadays the most powerful damper system that can be used in washing machines are based on friction-sponge-MR-fluid, which combines reliability, adaptive control and low-cost.

On the other hand, it is possible to find on the market new damper systems using other type of technology in other application, e.g. in the automotive industry.

2.3.6 Gas Damping System

Gas damping systems are similar to hydraulic. The damper is divided into two or more chambers wherein the gas passes through. The most common gas used is the air. This device is used frequently to support weights, e.g. vehicle doors, height of a chair. Adjustability can be achieved by reducing the gas volume and hence increasing its internal pressure by means of a movable end stop or allowing one tube to slide over another. Gas dampers of high pressure contain a very large amount of energy and could be used as a power pack.

It requires just a few components: a piston that moves up and down inside an aluminum cylinder that is filled with air instead of oil. As the piston moves in and out, some of the air is displaced, thereby creating an additional progressive spring action. Some of the remaining air flows through valves to create the damping effect.

Gas dampers are not common used in washing machines suspension systems.

2.4 Washing machine description

This thesis has been focused in front load washing machines in which exist several ways of drum support. The most common is based on the standard four-strut configuration, where the drum, motor and all driving devices are hanging inside the housing using springs on the up zone and four dampers on the bottom, as can be seen in *Figures 2.4.1-2*. Said configuration is usually built with friction dampers, and in case of this thesis, with internal spring included.

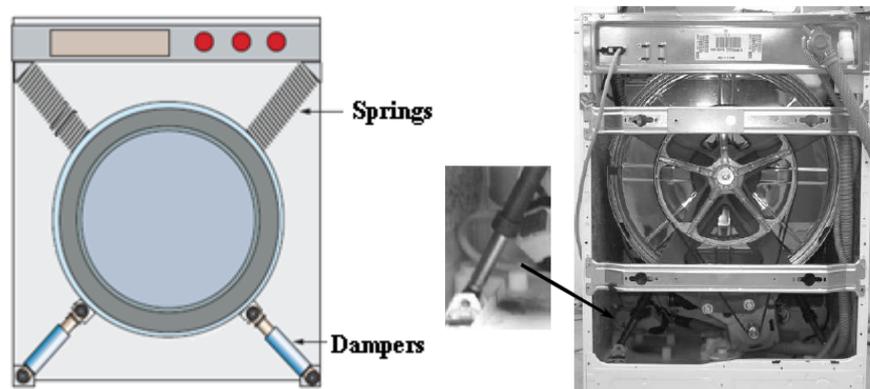


Figure 2.4.1 Dampers location in front-load washing machines

The washing machine work cycle can be divided into several stages, and according to vibrations it is depending majority on the rotational speed of the drum and the unbalanced loads. The speed range involved during the cycle of normal washing machines is from 0 to 1200-1800 rpm which represents the spinning stage. When the system speed rises about to 200 rpm the resonant frequency is reached, increasing the amplitude of vibrations. During these low frequencies the systems require high damping to lessen said vibrations. After that, once the system increases its spin speed the vibrations are reduced so low damping at these high frequencies is enough.

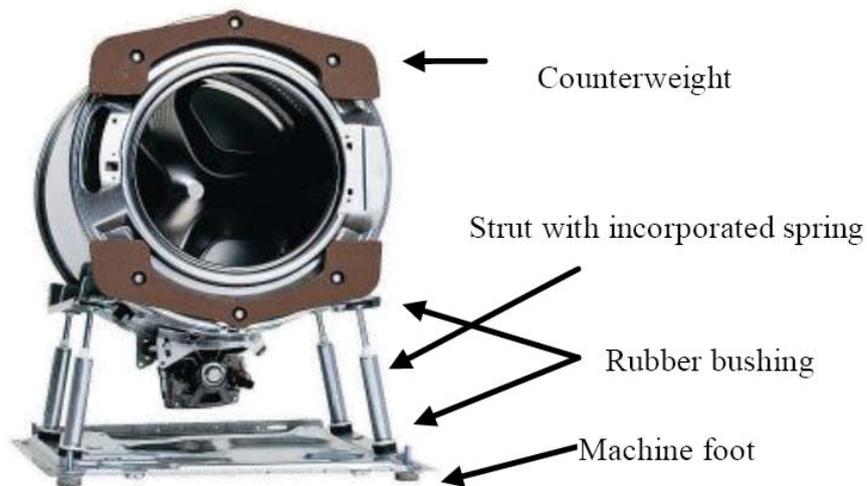


Figure 2.4.2 Standard four-strut configuration. Extract from [27]

The standard dampers used to isolate said vibrations are based on classical coulomb friction between two bodies. In this case, the dampers mounted in the washing machine offers a constant damping value during the whole work cycle. As can be seen in Figure 2.4.3, the friction force is represented against the displacement, showing the standard damper behavior.

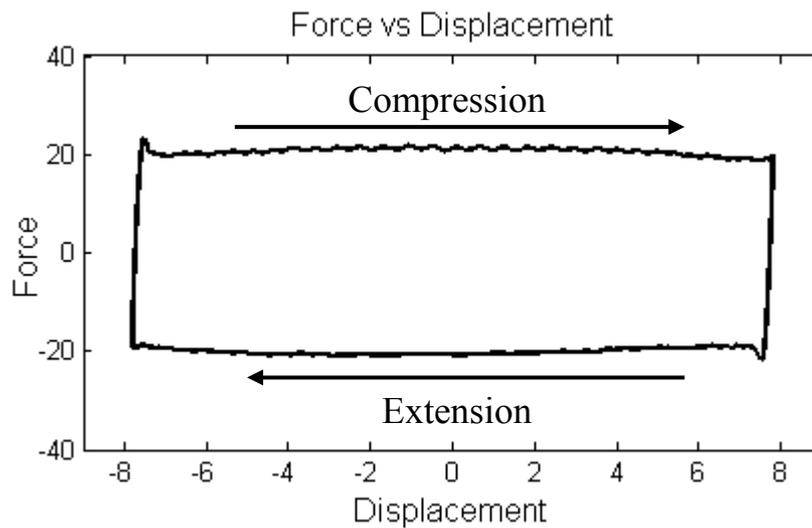


Figure 2.4.3 Standard damper behavior

3 Design Specifications

3.1 Introduction

The first design stage is to define and describe the technical systems, thus it is possible to separate the general problem in partial problems and look for partial solutions, that join together, represent the general solution.

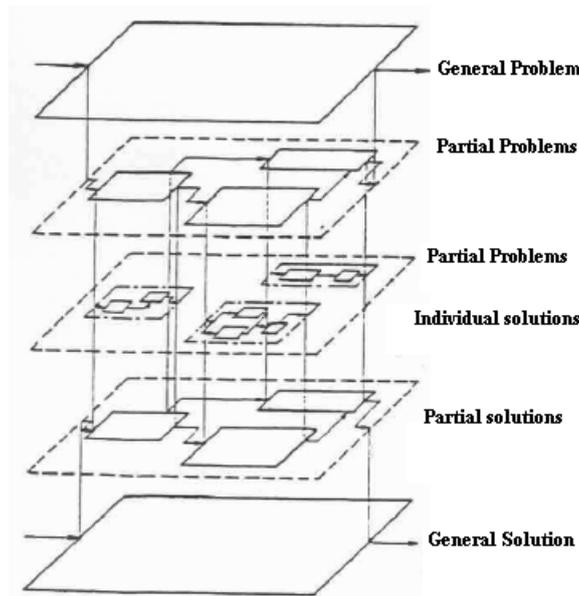


Figure 3.1.1 Technical system division

The technical systems description is made by the observation of the prevailing conditions. To complete the task, it should be exists a transmission of:

Energy: mechanical, thermal, electrical, chemical

Matter: material (liquid, solid...), test piece.

Information: magnitude to measure, data, notice signal.

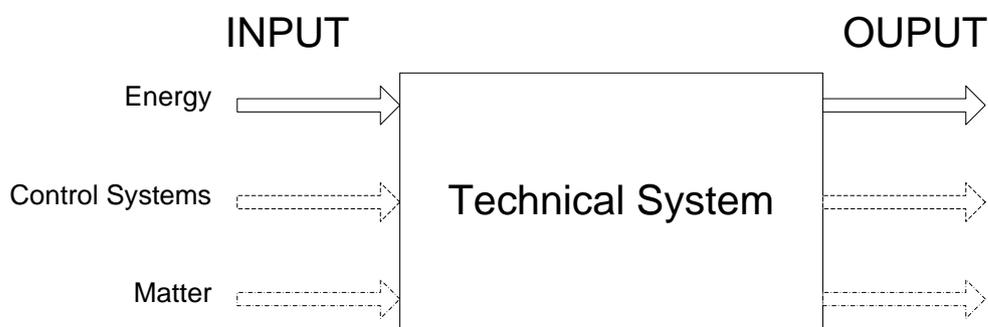


Figure 3.1.2 Technical system inputs and outputs

The technical systems description is formed by the relations between inputs and outputs magnitudes.

Each work starts with the confrontation of the problem. Thus, in many cases the task is confused, lack of information or disoriented due to the problem is already fixed with existing solutions or exist a disinformation that make false circumstances. The technical responsible needs a detailed specification of the task according to its own criteria:

What characteristics should contain the solution?

What characteristics should not contain?

For what application should be used?

Given that in mostly of the cases the customer does not supply these requirements, the technical have to ask him for it. Among others (inter alia), which existing products have to be replaced, which market cover or if the production will be self or for concession.

Black Box

The first step to define the system is to insulate it (as a black box) and only take into consideration the inputs and outputs characteristics. The following graph shows said black box wherein as outputs it has considered the washing machine vibrations, and a control signal (if applicable). On the other side, reduced WM vibrations, vibrations, heat, noise (itself) and sensor are outputs. As a union device it has represented an ECU (Electric Control Unit) to analyze the signals coming from the sensor/s and send new signals to the system to improve the behavior, modifying internal parameters as magnetic field, electric field, material position, etc. This last element just appears if the control is applicable

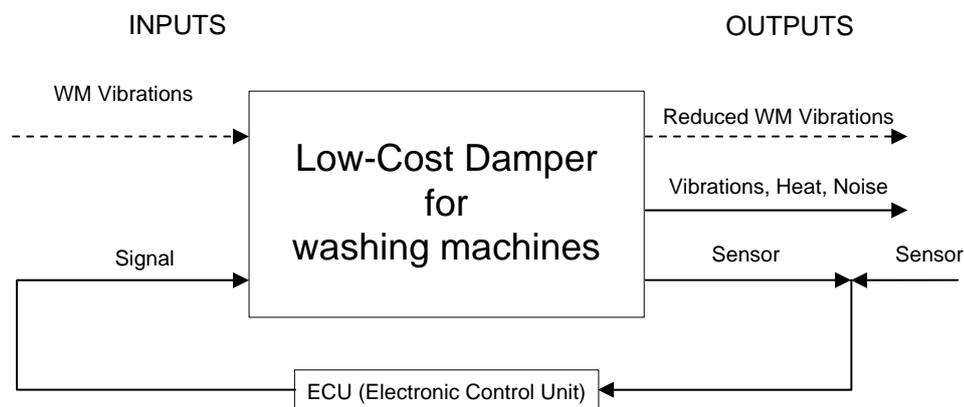


Figure 3.1.3 Technical system black box

Requirements/Wishes list

The next table shows the different demands which has been taken into account for the damper. One can see there are two kinds of demands: Requirements and wishes.

Requirement means that it “must” be included in the final design, in other words, they are mandatory requirements. Wishes are less important requirements for the final design, for example the type of technology used, or the kind of material (in this case). Thus, wishes can be carried out or not.

Table 3.1.1 Table of requirements and wishes

| Requirement Wish | Description |
|-----------------------------|---|
| R | Low-cost of manufacturing |
| R | Reduce washing machine vibrations |
| R | Variable response depending on level vibrations |
| R | Compatibility with mechanical structure |
| R | Support washing machine load/drum |
| R | Minimum friction force 15 N |
| R | Stroke range (70-80mm) |
| W | Change behavior mechanically with frequency |
| W | Change behavior mechanically with amplitude |
| W | Change behavior with external signal |
| W | Low damping at non-resonance frequency |
| W | High damping during resonance |
| W | On/Off behavior |
| W | Low/high behavior |
| W | Friction system |
| W | Hydraulic system |
| W | Pneumatic system |
| W | Magnetorheological system |
| W | Electrorheological system |
| W | Quick fixations |
| W | External plastic material housing |
| W | Memory alloy shapes (e.g. piezoelectric) |
| W | Low wear rate material |

3.2 Quality function deployment analysis

3.2.1 QFD Analysis

Quality function deployment (QFD) is a “method to transform user demands into design quality, to deploy the functions forming quality, and to deploy methods for achieving the design quality into subsystems and component parts, and ultimately to specific elements of the manufacturing process”, as described by Dr. Yoji Akao, who originally developed QFD in Japan in 1966, when the author combined his work in quality assurance and quality control points with function deployment used in Value Engineering. QFD is designed to help planners focus on characteristics of a new or existing product or service from the viewpoints of market segments, company, or technology-development needs. The technique yields graphs and matrices.

QFD helps transform customer needs (the voice of the customer [VOC]) into engineering characteristics (and appropriate test methods) for a product or service, prioritizing each product or service characteristic while simultaneously setting development targets for product or service. One of the fundamental matrices in QFD is the House of Quality (HoQ). This matrix relates customer requirements to business objectives (and engineering actions).

Even though the matrix used in the selection process looked different than the conventional House of Quality matrix, the mentality behind the use of a matrix was the same.

The first matrix indicated the customer requirements versus engineering requirements. Each engineering technical requirements has been graded from 1 to 5 depending on its importance, wherein 1 is less important requirement and 5 is the most important requirement. It has graded the relationship from one to three such that: 1 is for negative effect, 2 is for normal effect and 3 is for positive effect. Besides, some of the engineering requirements were marked with “0” representing no relation. The necessity of each engineering requirement had, thus, been determined over the customer requirements they are related to. *Table 3.2.1.1* shows the contents of the QFD analysis.

Table 3.2.1.1 Table of contents QFD Analysis

| | | j → | | | | |
|--------|------------------------------|---|-----------------|-----------------|-----------------|-----------------|
| | | Customer Requirements (CR) | | | | |
| | | A | B | C | D | |
| i ↓ | Importance (g _i) | Engineering Technical Requirements (ETR) | | | | |
| | g ₁ | a | W ₁₁ | W ₁₂ | - | - |
| | g ₂ | b | W ₂₁ | .. | - | - |
| | . | c | - | - | W _{ij} | - |
| | g _n | d | - | - | - | W _{nn} |
| | | G_{wj} = G | | | | |
| | | (N° ETR - W_{ij}Zeros) = F | | | | |
| | | (W_{ij} Max * F) = H | | | | |
| | | RATIO = G/H | | | | |

Wherein G_{wj} is:

$$G_{wj} = \sum_{i=1}^n g_i \cdot w_{ij} \quad (3.2.1.1)$$

F is the number of ETR to take into account, so it can be defined as the number of ETR minus the number of zeros in the grades related to each Customer Requirement. H shows the maximum sum of each CR, thus, shows the best relation between both requirements. Finally, RATIO shows what percentage of the Customer Requirements fulfills the Engineering Technical Requirements.

As can be seen in *Table 3.2.1.2*, several requirements were taken into account in the ETR, from “Reliability” to “Number of parts”. Regarding the CR, some of them show the different damping systems, and the others show for example two sorts of behavior, “On/Off” and “Low/High”.

Table 3.2.1.2 List of ETR and CR

| Engineering Technical Requirements | Customer Requirements |
|---|------------------------------|
| Durability | Friction Damping |
| Reliability | Hydraulic Damping |
| Temperature resistance | Gas Damping |
| Comfort | MR Fluid |
| Low Maintenance | ER Fluid |
| Power Consumption | Piezo-electric |
| Comfort noise | Without external control |
| Ease of manufacturing | Control by external signal |
| Complex structure | On/Off Behavior |
| Environmental Requirements | Low/High Behavior |
| Response Time | Continue Variable control |
| Level of Technology | Sensors |
| Volume | Requires Power |
| Safety | Rubber pads |
| Cost | Metal Housing |
| Materials | Plastic Housing |
| Number of parts | |
| Ease of assembly | |
| Need for a control | |
| Weight | |

Once done the analysis, as an example, “On-Off behavior” got an overall mark of 9 over 12 and is, thus, more important to include in the design than “Requires power” which obtained 7 over 18. The ratio of these two numbers led us to the necessities. The bar chart in *Figure 3.2.1.1* shows the importance of each engineering requirements for the development of the damper.

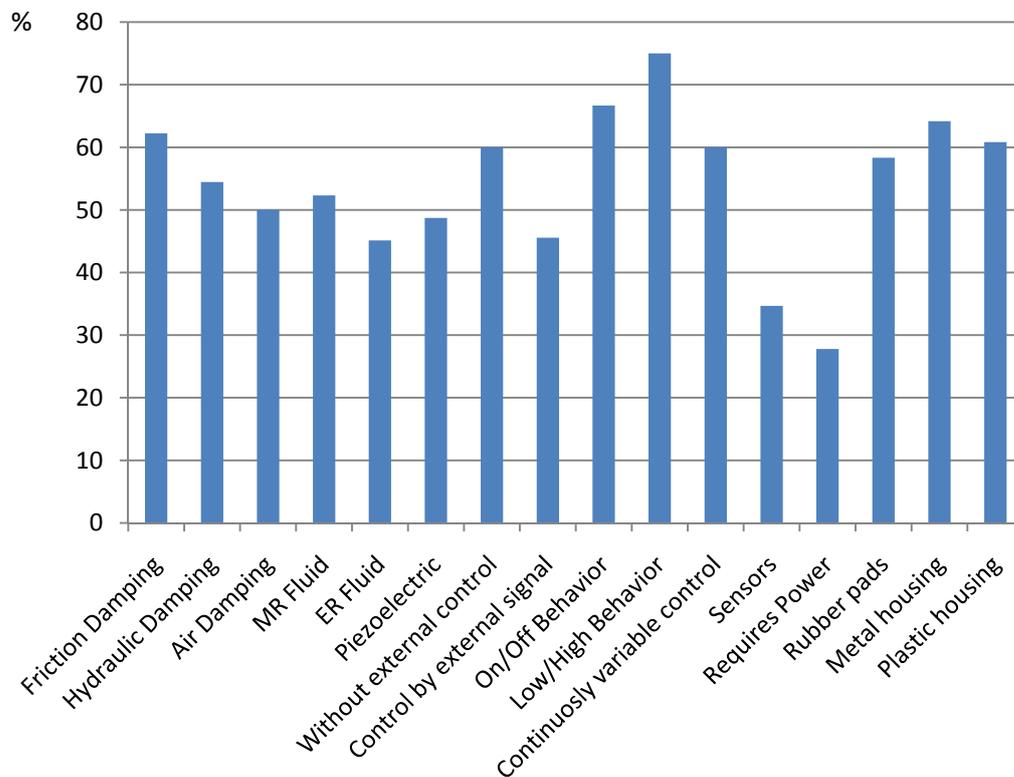


Figure 3.2.1.1 Bar chart of QDF results

On the second step, the engineering requirements and their importance levels obtained are used to grade initial concepts and define their characteristics. Grading the concepts 1 to 5 where 1 is for minimal use of the engineering requirement and 5 is for maximum use of it. Since some of the engineering requirements were undesirable whilst some were, their normalized grades were calculated as their differences between the mean value, either positives (likes) or negatives (dislikes). Finally, every concept obtained a final mark which is the sum of the weighed points they got. The bar chart in *Figure 3.2.1.2* shows the end results of the selection process. One can see that the concept named “Variable Friction” and “Depending on Amplitude”, are chosen directly. However, “Variable Valve” and “Variable Magnetic Field” have a quite similar level, so both might be the next possible solutions. Regarding the behavior, the two types considered shows the same level so it will depend on the complexity of the design to select one or the other.

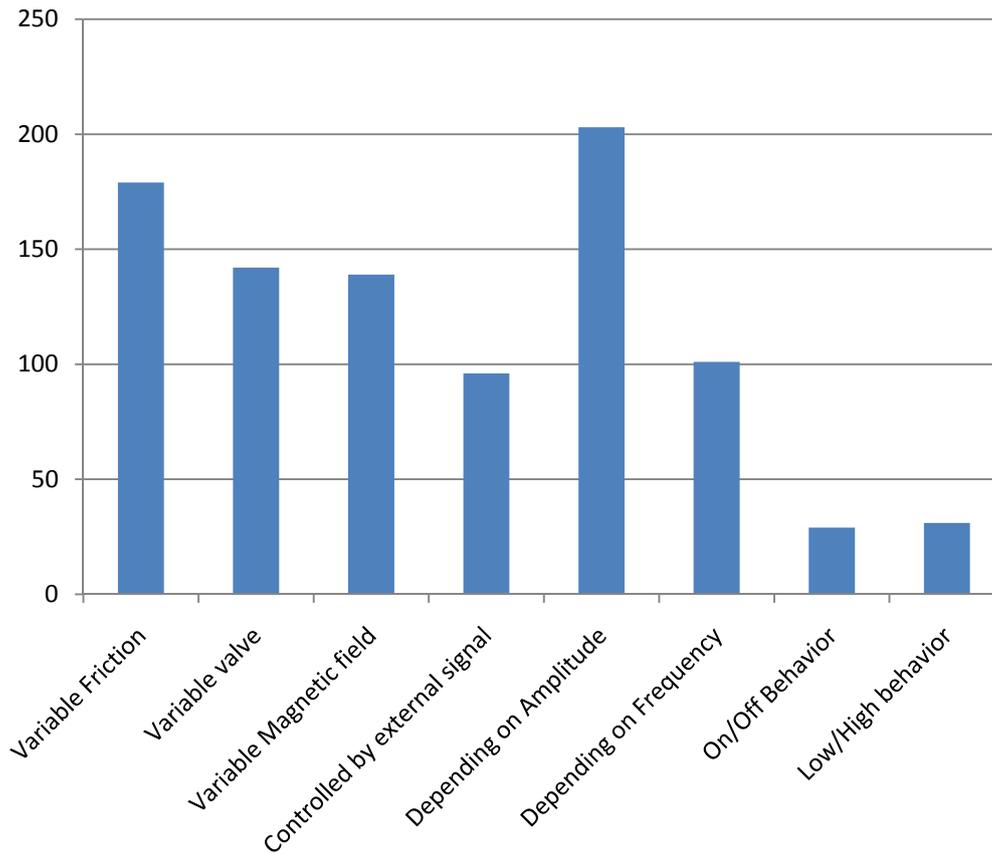


Figure 3.2.1.2 Bar chart of selection process

Conclusions

In this final selection of concepts session, the QFD analysis was done, considering various parameters (quality functions) on which the damper depends or may depend. The new damper concept, therefore, must be designed according to the QFD analysis results. The three best solutions which fit better the requirements are: Variable friction, Depending on Amplitude and Low/High behavior. Thus, the new design will be based on said specifications, even though the other solutions may be considered as Hydraulic and MR fluid.

3.2.2 Sensitivity analysis

As one can see, the QFD method is not 100% reliable, owing to the subjectivity of grading marks. For that reason a simple sensitivity analysis to figure out how can vary the results according to the marks selected for each requirement is worth to do.

The first method used is to compare the bar charts changing the graded marks. For example, change all the “1” by “2” and see what happen. The same is done with the other marks, thus, there are four sorts of variation: change 1 by 2; 2 by 1; 2 by 3 and 3 by 2.

So, after viewing the results of this comparison one can draw conclusions about the sensitivity of the graded marks.

Table 3.2.2.1 Conclusions table of sensitivity analysis

| Change | Conclusions |
|---------------|---|
| 1 by 2 | Customer Requirements fits better Engineering Technical Requirements. More distance for friction damping, the same first three types. More distance between without and with external control. On/Off drops. Continuous and Low/High wins (more difference). Sensor (the same), Require Power and Rubber drop. Metal-Plastic (less difference %). |
| 2 by 1 | Customer Requirements fits worst Engineering Technical Requirements. More distance for friction damping, the same first three types. More distance between without and with external control. On/Off drops. Continuous and Low/High wins (more difference). Sensor (the same), Require Power and Rubber drop. Metal-Plastic (less difference %) |
| 2 by 3 | Enhance global percentage. Modify selection (Air and Hydro enhance). Low/High-On/Off same percentage. |
| 3 by 2 | Decrease global percentage. Modify selection (Air and Hydro enhance). Low/High-On/Off same percentage. |

Table 3.2.2.2 Increasing values of each grade

| Grade | Change by | Value increase (%) | Results |
|--------------|------------------|---------------------------|----------------|
| 1 | 2 | +8,47 | The same |
| 2 | 1 | -20,37 | The same |
| | 3 | +16,92 | Change |
| 3 | 2 | -16,6 | Change |

The grade 1 is not very significant in order to change the results. Besides, increase the percentage of compatibility between the two lists of requirements.

The grade 2 has more influence in the results, due to it can raise or decrease depending on subjectivity.

The grade 3 changes completely the results with a percentage decrease.

3.2.3 Percentage of sensitivity

Another way to measure the sensitivity of the graded marks is to know if the grades are right, and what percentage of rightness has each one. To do this in a simple way, one can compare the total amount of each mark to the marks that are not sure. For example, there are marked as mark three, sixty times, and for these 60 times, 11 are

not sure, so the ratio is $11/60=18.33\%$ of sensibility, one cannot secure the rightness of this percentage.

The next table shows the sensibility ratios of the marks.

Table 3.2.3.1 Percentage of sensitivity

| Mark | Total | Not sure | Ratio (%) |
|--------------|-------|----------|-----------|
| 1 | 36 | 17 | 47.22 |
| 2 | 56 | 6 | 10.71 |
| 3 | 60 | 11 | 18.33 |
| Total | 152 | 34 | 22,36 |

Thus, the total percentage of sensibility is 22.36%

3.3 Modification analysis

Once finalized the selection of the main technology used (Friction, Hydraulic or MR Fluid) one may compare the methods for achieving the behavior change in each one.

Friction (classical Coulomb fiction model)

In case of friction, there are two parameters which can be modified as shows the equation (3.3.1); the frictional coefficient (μ) and the normal reaction (N).

$$Fr = \mu \cdot N \quad (3.3.1)$$

The next table shows the different sorts of modifications that can be done to modify these two parameters.

Table 3.3.1 Friction parameters modification

| Parameter | How to modify | |
|-----------|--------------------------|---|
| μ | Modify material surface | |
| | Lubricate | |
| | Change Temperature | |
| | Modify polishing surface | |
| N | Modify bodies shape | |
| | Expand on body | Temperature |
| | | Electric signal (e.g. piezoelectric) |

Hydraulic/Fluid

In a hydraulic damper, the two parameters that can be modified are the fluid viscosity and the gap which control the flux between chambers, most of the time valves. As variable valve, it is said that it is possible to change its size (change the size of the hole/gap). There is also other parameter related to valves that can be changed, intake valves. Varying the geometry or the position of the intakes in respect of the piston position.

Table 3.3.2 Hydraulic parameters modification

| Parameter | How to modify | |
|----------------|------------------------------------|----------------------|
| Variable valve | External signal | Piezoelectric valve |
| | No signal | Modify piston rod |
| | | Modify inner housing |
| Viscosity | Change temperature | |
| | Variable magnetic field (MR fluid) | |
| | Variable electric field (ER Fluid) | |
| Intake valve | Modify geometry | |

The next step is to compare all these sorts of variation and select which fits better some list of requirements. These requirements are in order of importance: Cost, Durability, Reliability, Materials Ease manufacture and Power consumption. The method is similar to the QFD, grading from 1 to 3 each type of modification. It is used the same table:

Table 3.3.3 Table of contents QFD Analysis

| | | j | | | | |
|---|-------------------------|---|----------|----------|----------|----------|
| | | Modifications | A | B | C | D |
| i | Importance (g_i) | Requirements | | | | |
| | g_1 | a | w_{11} | w_{12} | - | - |
| | g_2 | b | w_{21} | -- | - | - |
| | . | c | - | - | w_{ij} | - |
| | g_n | d | - | - | - | w_{nn} |
| | | $G_{wj} = G$ | | | | |
| | | $(N^{\circ} \text{ ETR} - W_{ij} \text{Zeros}) = F$ | | | | |
| | | $(W_{ij} \text{ Max} * F) = H$ | | | | |
| | | RATIO = G/H | | | | |

Wherein G_{wj} is:

$$G_{wj} = \sum_{i=1}^n g_i \cdot w_{ij} \quad (3.3.2)$$

FRICTION

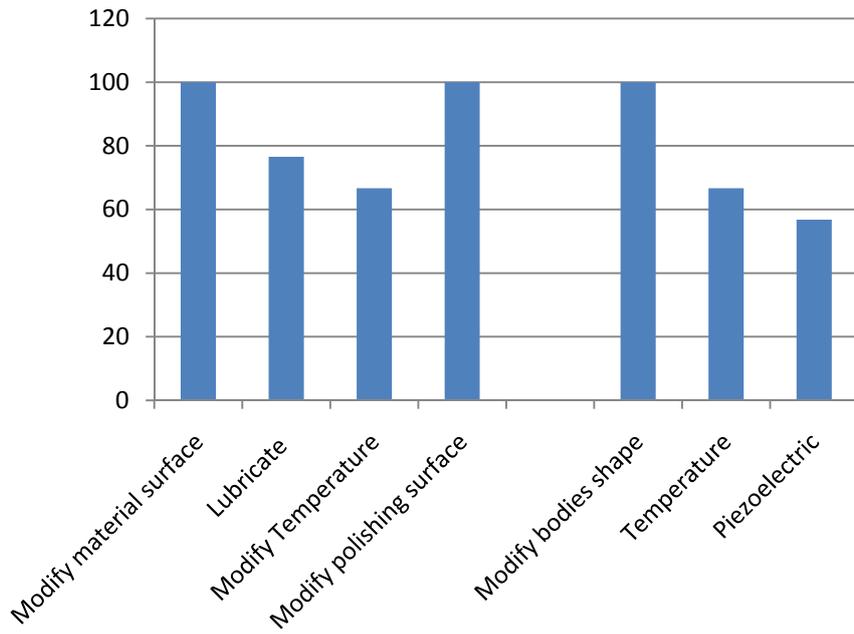


Figure 3.3.1 Friction bar chart solution

HYDRAULIC

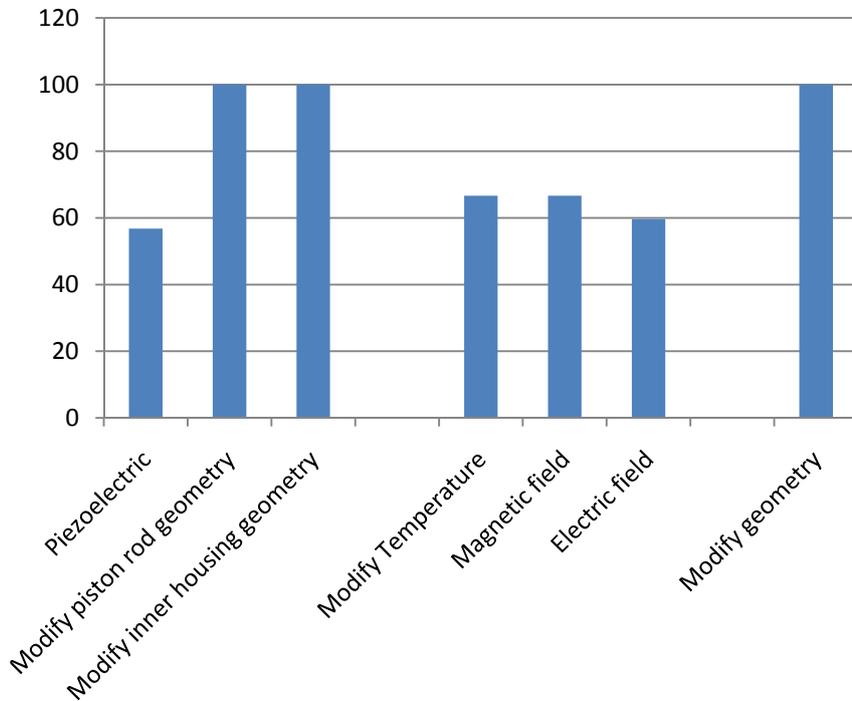


Figure 3.3.2 Hydraulic and MR Fluid bar chart solution

Conclusions

As show in *Figure 3.3.1* the best choice to modify the frictional constant (μ) was “Modify material surface” or “Modify polishing surface”. To modify the normal reaction (N) “Modify bodies shape” obtained the best mark. These results were logical owing to both modifying temperature and change lubricate conditions were not as simple as the chosen.

On the hydraulic section, one can see in *Figure 3.3.2* that “Modify piston rod geometry” and “Modify inner housing geometry” were better than “piezoelectric valve” related to variable valve parameters. In terms of viscosity, the three possibilities were over the same level. The last parameter “Intake valve” and its sort of modification “Modify geometry” may be considered.

3.4 Final conclusions

In this stage the design specifications were analyzed. For that, a deep QFD analysis was done to focus the possible solutions. Different sorts of technology and configurations were considered and valuated. Due to said analysis does not offer the best and unique solution for each problem, one cannot leave aside any of the discarded solutions, since it would be possible to use them in a new way and turned into the best solution.

The main final proposed solutions to implement the new damper were:

- Friction damping
 - Modify material surface
 - Modify bodies shape
- Without external control / pure-mechanical system
- Depending on amplitude
- Low-High behavior

4 Concept development

4.1 Available patents

A deep study was done in order to overview a wide number of available patents, in which dampers with a variable damping behavior capacity are used in washing machines suspension systems. Each patent is described hereinafter.

4.1.1 Friction Damper for Washing Machines

This patent is based on a low/high friction damper with mechanical action, depending on amplitude. The friction damper comprises a hollow shell and a rod, coaxially arranged with respect to one another and forming a telescopic motion construction. Said rod has an outer diameter less than an inner diameter of said shell. A friction damper element is arranged between said rod and hollow shell in a space delimited by a pair of top and bottom resilient means. Said damper element comprises a substantially cylindrical holding bushing having a thoroughgoing axial hole and fitted about the rod. The bushing has an inner surface and an outer surface in which seats are former for housing a first friction clamp and a second friction clamp, wherein said clamps are arranged concentrically.

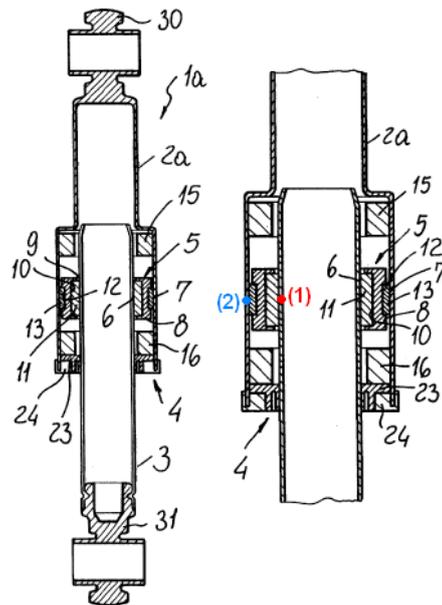


Figure 4.1.1 Patent 4.1.1

The first friction clamps have a first working surface frictionally engaging an outer surface of the rod (1), and the second friction clamp have a second working surface frictionally engaging an inner surface of the hollow shell (2), wherein the first working surface is larger than the second working surface thereby said first working surface provides against the rod a friction reaction greater than a friction reaction provided by the second working surface against the hollow shell. [13]

So at low amplitudes, the damper element and the rod move as one piece, because of the first friction reaction is greater than the second, taking place friction damping (low damping) on the second surface, between the second friction clamp and the hollow shell. Whereas when the amplitude increases, the damper element fits against the top and bottom resilient means, so at this point, the first friction clamp starts to bear against the rod, taking place the second frictional behavior (high damping), greater than before. It is a very simple device, with no complex pieces. One problem might be the free position of the holder once loaded.

4.1.2 Damper for Spin-Drying Washing Machines

In this patent is discussed a damper for spin-drying washing machines, provided with an amplitude-dependent damping behavior, being obtained accompanied with manufacture at a low-cost, comprising a tubular casing which is guided for displacement in the casing and projects from an end thereof. Fastening elements which are mounted on a free end of the casing and of the tappet, respectively, and a frictional damping unit, which is disposed inside the casing, comprising at least one elastic frictional damping lining which is displaceable in relation to the casing and the tappet along the central longitudinal axis and which lies bare at least sectionally in a lengthwise axial direction, producing a given frictional damping effect.

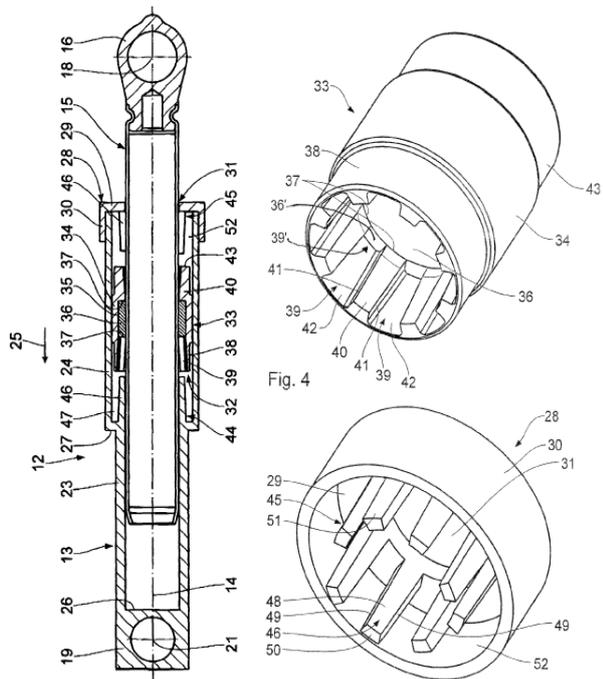


Figure 4.1.2 Patent 4.1.2

At least one stop element which is stationary in relation to the casing and turned towards the at least one frictional damping lining, defining the motion of the at least one frictional damping lining, with the at least one stop elements being configured such that, for motion damping, it directly cooperates with the at least one frictional damping lining. The gist of the invention resides in that the elastic frictional damping lining lies open at least sectionally in the axial direction so that the stop element, in the case of great vibration amplitudes, cooperates directly with the frictional damping lining. Thus, the frictional damping lining simultaneously fulfills the task of a stop buffer, this leading to constructional simplicity of design and to manufacture of the damper at a low-cost. [14]

4.1.3 Damper for Washing Machine

Disclosed is a damper for a washing machine. The damper includes a movable part for adjusting damping force using an external electric source, a damping pad for varying an exerting state of the damping force in response to an operation of the movable part, a pad holder for supporting the damping pad, a cylinder contacting the damping pad to generate the damping force, a rod inserted in the cylinder.

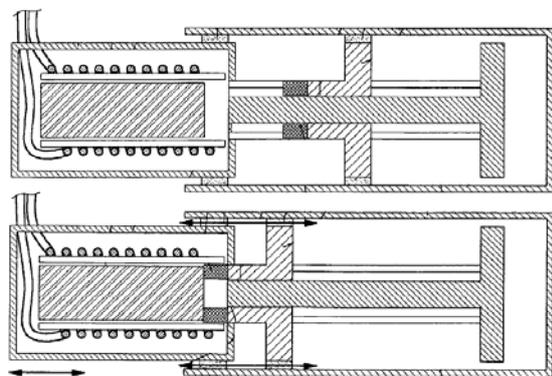


Figure 4.1.3 Patent 4.1.3

The cylinder is connected to one of a movable part and case of the washing machines, and the rod is connected to the other of the movable part and case of the washing

machine. The pad holder is disposed on an outer circumference of the rod. [15] The pad holder contains magnetically elements in one end. There is a coil and a magnetic body disposed in one end of the movable part, being activated with the external current source to attach the pad holder and allowing the friction action between the pad and the inner surface of the housing. Without supplied current, the friction takes place between the rod and the inner surface of the pad holder, which has no movement owing to that the friction force between the pad and the housing is greater than the one before.

In case of current supplier failure, the damper continues offering damping due to the friction between the rod and the pad holder. The system contains pieces hard to mount. It is needed a good lubrication between the rod and the pad holder, and try to seal as much as possible the magnetic chamber.

4.1.4 Frictional Damper, for Washing Machines with Spinning Cycle

A frictional damper, in particular for spinner-type washing machines, comprises a tubular housing and a tappet guided displaceably in the housing, articulation elements being mounted on the free end of the housing and the tappet. A damping element is formed on the end of the housing in the tappet exit side, in which a damping element carrying a friction lining is disposed displaceably in the direction of the central longitudinal axis. The damping element bears against prestressed helical compression springs. [16]

At low amplitudes, the damping force is provided by the compression springs, wherein the rod and the pad holders move together, only a certain displacement. Then, at high amplitudes, one spring compress all and the pad holder fits against the stoppers, starting to bear the pads against the rod, namely high damping.

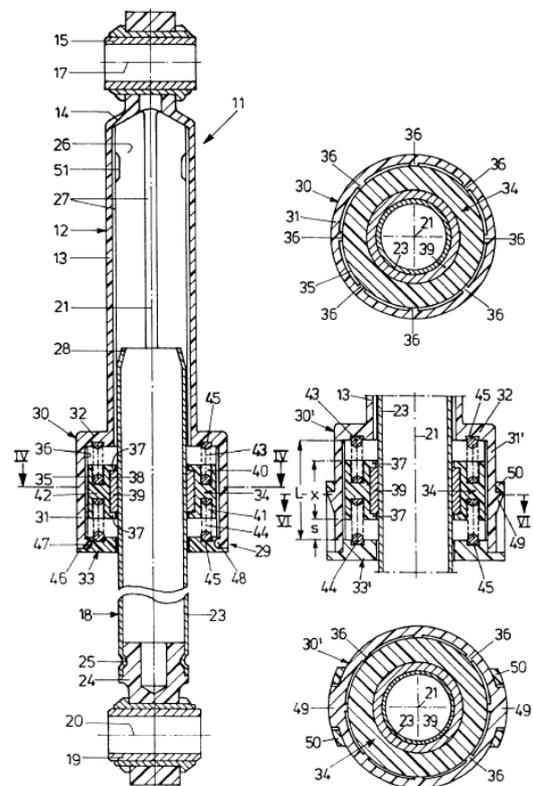


Figure 4.1.4 Patent 4.1.4

4.1.5 On-Off Damper System, Especially For Washing Machine

The on-off binary damper system has a friction pad and a housing friction damper surface, with the friction pad in contact with said housing friction damper surface. The on-off binary damper system has an electromagnetic coil core and a magnetic locking slide with a slide gap between said electromagnetic coil core and said magnetic locking slide to provide for relative sliding motion between the electromagnetic coil core and the magnetic locking slide, wherein a current supplied to said electromagnetic coil core removes the slide gap and electromagnetically locks the

electromagnetic coil core and the magnetic locking slide together with the relative sliding motion transferred to the friction pad, with the friction pad rubbing against said housing friction damper surface in order to dampen the problematic movement between the frame and the rotating tub. [17]

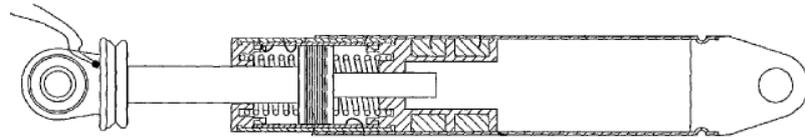


Figure 4.1.5 Patent 4.1.5

Wear can appear over the contact zone between the coil and the slide, causing noise generation or rough movements.

4.1.6 Frictional Vibration Damper

A frictional vibration damper, in particular for washing machines with an oscillatory support of a washer unit has an approximately tube-shaped housing and a tappet displaceable in it. An articulation element is provided at the outer end of each the housing and the tappet. Further, a friction piston bearing with a friction coating on the internal wall of the housing is formed on the tappet, which friction piston is extendable by means of a linearly operation servomotor. In order to achieve high safety in operation at little expenditure, the servomotor is arranged in the tappet. It is formed by an electrically heatable thermoactuator element having a housing with a piston which changes its position relative to the housing upon heating. The piston of the servomotor directly bears against an expansion member serving to expand the friction piston. [18]

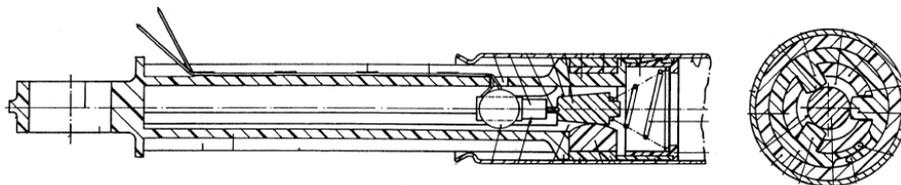


Figure 4.1.6 Patent 4.1.6

The complex structure gives this device an added difficulty of manufacturing and mounting. Also, the existence of the servomotor and the heatable thermoactuator, increase the complexity of the device, making it less reliable.

4.1.7 Combination Type Damping and Washing Machine

Disclosed are a combination type damper and a washing machine having the same. The combination type damper comprises: a cylinder provided with a receiving space having a certain depth at one side thereof; a rod relative movably inserted into the receiving space of the cylinder; and a composite damping means for generating a damping force by an electromagnetic force when a displacement of a relative movement of the cylinder and the rod is less than a preset length, and for generating a frictional damping force when the displacement of the relative movement is more than the preset length. Transient state vibration and steady state vibration generated from a tub assembly during the entire processes for washing laundry are prevented from

being transmitted to a cabinet, thereby minimizing vibration noise towards outside while using the washing machine. [19]

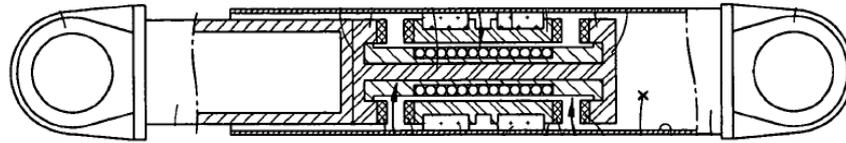


Figure 4.1.7 Patent 4.1.7

Necessity of good alignment to offer constant magnetic force once loaded. The first damping stage takes place without wear and noise.

4.1.8 Magnetically Actuated Motion Control Device

A magnetically actuated motion control device includes a housing defining a cavity and including a slot therethrough. A movable member is located within the cavity and is movable relative to the housing. A magnetic field generator located on either the housing or the movable member causes the housing to press against the movable member to develop a friction force. [20]

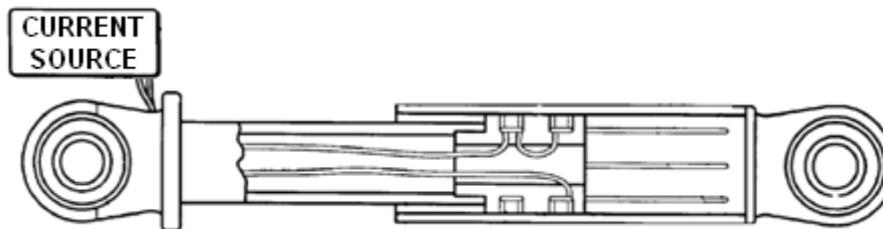


Figure 4.1.8 Patent 4.1.8

High wear over the inner surface of the housing is produced due to continuously contraction and expansion thereof. It would be possible to add a second friction point to get low/high behavior.

4.1.9 Frictional Damper for Spinner-Type Washing Machines

A frictional damper for spinner-type washing machines comprises a housing and a tappet, which is coaxially displaceable within the latter and the inner end of which is provided with a damping piston. The damping piston is disposed on a piston-bearing section of the tappet to be displaceable between biased compression springs bearing against stops, and provided with a friction lining elastically pressed against the inside of the housing. [21]

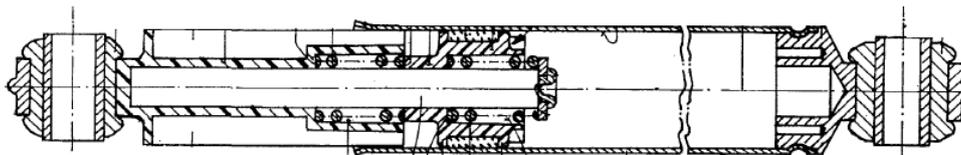


Figure 4.1.9 Patent 4.1.9

The first friction stage is achieved by spring damping over the piston-bearing section. Then, when the amplitude increases the pad starts bearing against the inner surface of the housing. The rod has a complex geometry and the number of pieces is elevated.

4.1.10 Damper of Drum Type Washing Machine

A damper includes: a cylinder; a piston rod inserted to be movable linearly in the cylinder, a guide member provided with a coupling hole to which an insertion portion of the rod member is coupled at a center thereof, and provided with first and second grooves at an outer circumferential surface thereof; a fixed damping member fitted into the first groove and adhered to an inner surface of the cylinder; and a movable damping member up and down movably fitted into the second groove and selectively adhered to the inner surface of the cylinder. [22]

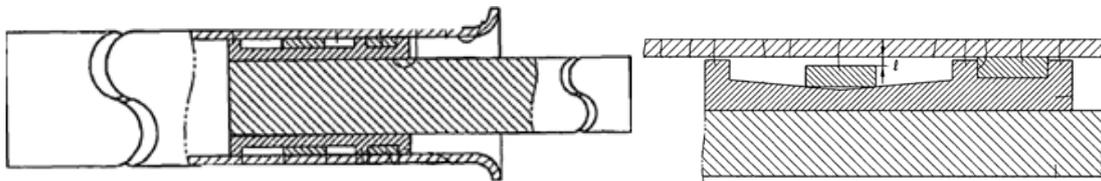


Figure 4.1.10 Patent 4.1.10

It is a simple device and easy to modify its parameters to achieve the required behavior. The main problem is the difficulty of centered.

4.1.11 Damper in a Washing Machine and Fabricating Method

A damper for a washing machine includes: a cylinder; a piston body having one end inserted into the cylinder, a first friction member interposed in a contact surface between the piston body and the cylinder, an extension bar having a circumferential stopper at the extended portion; a friction ring formed between the circumferential stopper and the first friction member; and a second friction member formed on an outer circumferential surface of the friction ring to contact with an inner surface of the cylinder. [23]

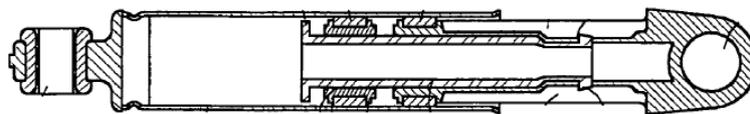


Figure 4.1.11 Patent 4.1.11

4.1.12 Damper and Washing Machines

The damper is formed by a cylinder having a receiving space with a predetermined depth in a longitudinal direction on one side thereof. A rod inserted to be movable relatively in a longitudinal direction in the receiving space of the cylinder. The friction surface is formed at one side of the cylinder or the rod and rubbing on the friction surface. There is a thermally deformable member (memory shape alloy) that is deformed according to a change in a temperature to make the friction member tightly attached to the friction surface or separated therefrom. To achieve this thermally

deformation is used a heating unit. The damper absorbs the small and large amount of vibrations to thereby prevent transfer of vibration to other parts. [24]

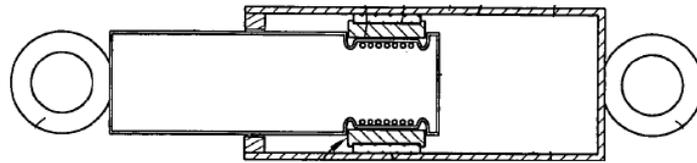


Figure 4.1.12 Patent 4.1.12

4.2 Proposed concepts

The following concepts have been developed based on all the patents of aforementioned and the conclusions of the QFD analysis. Some of them show only a first idea on which to base a possible final concept. The best concept for using in washing machines, taking into account all the main restrictions and appropriated considerations, is the first patent showed.

4.2.1 First concept

This first concept grows from patent number one, in order to reduce de inner parts and enhance the ease of manufacturing. Instead of using a holder supporting two pads of different section, it would be worth to remove the holder and replace the two pads by only one with variable section. This is based on the supposition of the pads which more contact surface offers more friction, as it is said in the patent report, so that, the outer pad surface is smaller than the inner, allowing the low-high behavior. In *Figure 4.2.1.1* one can see the proposed concept on the right with two different pad shapes.

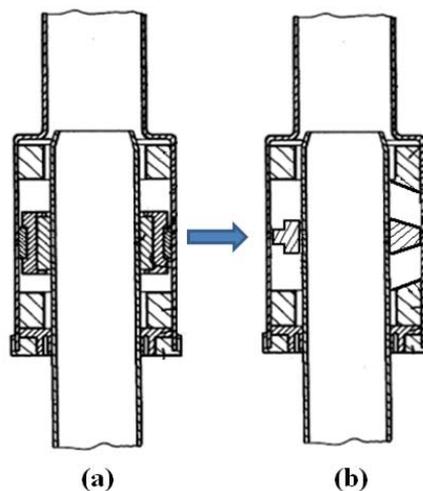


Figure 4.2.1.1 First concept (b) compared to patent (a)

4.2.2 Second concept

This concept deals the same supposition of aforementioned. Over the piston are fixed the dark gray pads which bear against the pad disposed inner the housing. As can be

seen in *Figure 4.2.2.1* in a central position the contact surface between the three pads is represented by $2x$. When the piston moves, reaches a point where the contact surface starts to increase until only two pads are bearing, noticed that $y > 2x$.

One of the main disadvantages could be the possibility of interference between pads during damping, although can be worth making recesses at the corners of the pads.

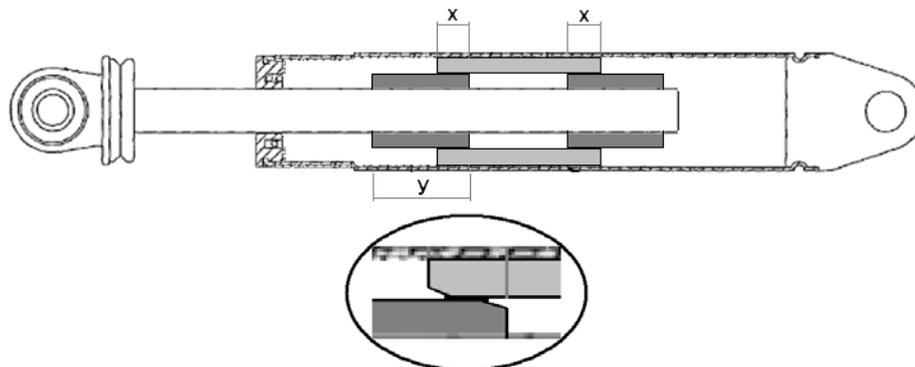


Figure 4.2.2.1 Second concept

4.2.3 Third concept

Even though this concept is not based on coulomb friction damping, it has clear references to the two-stroke engine, which uses intakes to allow passage of fluid. Hydraulic dampers can change its behavior varying the valve section whereas the fluid pass from one chamber to the other, and in this case is basically the same but instead of valve there are intake ducts which are opened depending on the amplitude of vibrations. As can be seen in *Figure 4.2.3.1* once passing the x amplitude value, the red duct is closed, so that the only passage way of the fluid is the blue one. This means that the section lessens, increasing the friction force and reducing the vibrations.

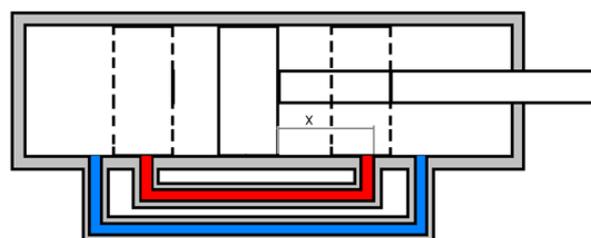


Figure 4.2.3.1 Third concept

Some possible disadvantages could be the risk of reflux from the ducts and how does it affect to the damper behavior. Another problem would be the high pressures inside de ducts.

4.2.4 Fourth concept

The next concept is based on patent 4.1.2, trying to avoid the complex inner piece full of ribs and slots. The proposed piece can be conical, so that, depending on the design

it could be possible to adjust the friction at the ends of the movement due to vibrations.

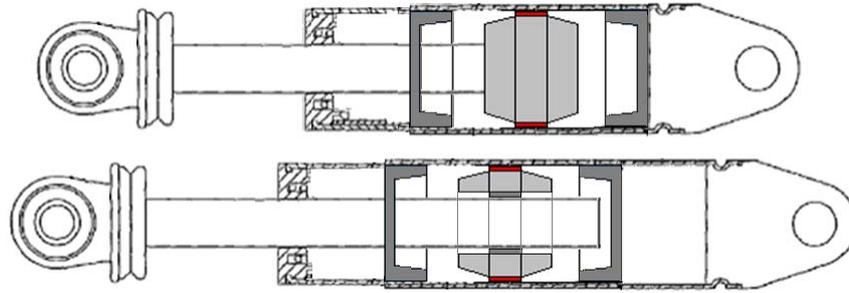


Figure 4.2.4.1 Fourth concept

4.2.5 Fifth concept

The following concept shows the piston rod with a constant slot on one side. The friction element is over a holder that moves along the piston rod. The holder works as a compression resort and when passes throughout the reduced gap on the piston lessen its force. Thus, the friction between the friction element and the inner surface of the housing can be controlled. This is the first considered idea, thus exist some restrictions that make it difficult to achieve. In this case, the holder is not attached over the rod, thus, when the holder passes through the gap, the contact between the rod and said holder disappears, so not a feasible solution.

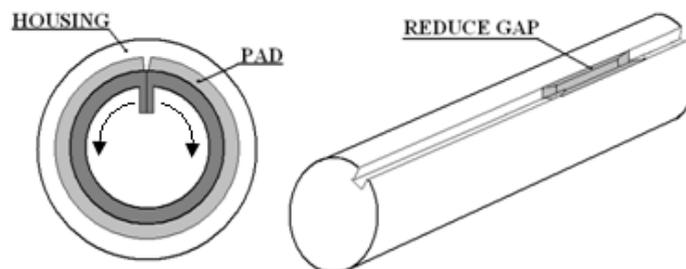


Figure 4.2.5.1 Fifth concept

4.2.6 Sixth concept

The last concept is based on multiple springs trying to obtain different stiffness depending on displacement. The idea is to add an extra stiffness during high damping, but this method is not very suitable for the suspension requirements.

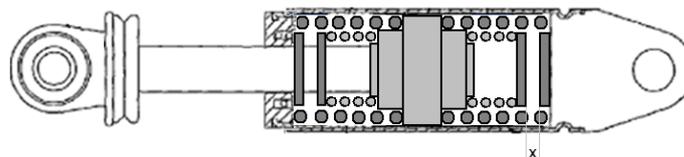


Figure 4.2.6.1 Sixth concept

Conclusions

Once showed all concepts the one which fits better all the requirements is the second 4.2.2. It offers a variable behavior in two damping stages depending on the amplitude

of vibrations. One advantage of this concept is its feasibility to be implemented using a normal friction damper as starting point, in order to manufacture a prototype.

So, finally the selected concept to be developed was 4.2.2

4.3 First stage of development

The first stage of development leads to know and estimate how will be the concept behavior and which parameters have to be taken into account. As aforementioned, the main behavior principle of the concept is due to the surface contact change between friction pads. The concept is based in two pads over the piston rod and one in the housing, although is it possible to change that, and do it reverse, two pad inside the housing and one over the piston rod. In *Figure 4.3.1* can be seen the parameterized concept with the main parameters involved.

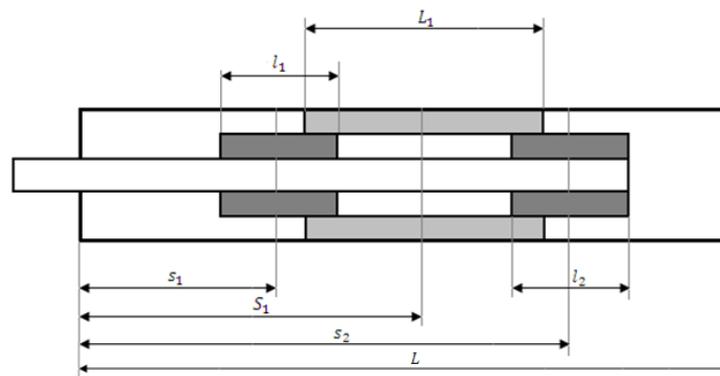


Figure 4.3.1 Main parameters involved

Adding to the before, *Figure 4.3.2* shows the extended concept parameters to develop a first geometrical analysis.

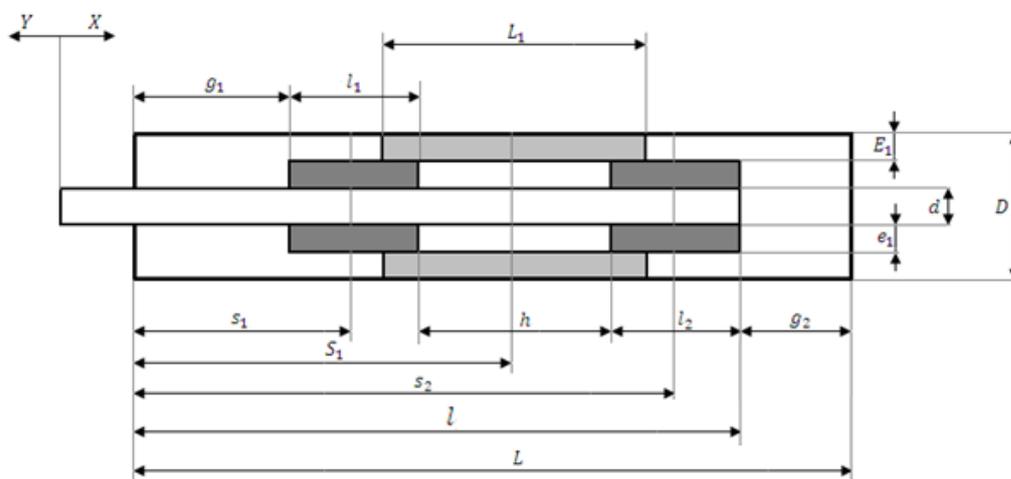


Figure 4.3.2 Extended parameters

Contact at centered position (only in one side): $C_T = C_1 + C_2$

$$C_1 = \left(s_1 + \frac{l_1}{2}\right) - \left(S_1 + \frac{L_1}{2}\right) \quad (4.3.1)$$

$$C_2 = \left(S_1 + \frac{L_1}{2}\right) - \left(s_2 + \frac{l_2}{2}\right) \quad (4.3.2)$$

$$C_T = s_1 - s_2 + L_1 + \frac{l_1}{2} + \frac{l_2}{2} \quad (4.3.3)$$

Contact during compression:

$$C_C = (C_1 + X) + (C_2 - X) = C_T \quad (4.3.4)$$

$$C_{Cfinal} = l_1 \quad (4.3.5)$$

$$\text{Transition: } X = l_2 - C_1 \quad (4.3.6)$$

Contact during Extension:

$$C_E = (C_1 - Y) + (C_2 + Y) = C_T \quad (4.3.7)$$

$$C_{Efinal} = 0 \quad (4.3.8)$$

$$\text{Transition: } Y = l_2 - C_2 \quad (4.3.9)$$

After this first geometric analysis, the next is to observe how will be the damping behavior depending on the geometry. Thus, different types of geometry have been studied, showing the estimated behavior diagrams. The following diagrams show the contact surface between friction pads against displacement using examples values of the parameters. With this one can figure out how vary the behavior.

Version 2.1

This first version is based in the following geometry *Figure 4.3.3*, wherein the pads 1 and 2 are in contact with the housing pad, at the centered position. The *Table 4.3.1* shows the used values and the cases studied.

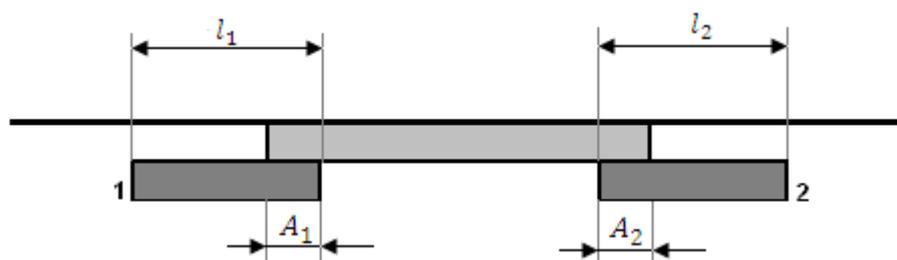


Figure 4.3.3 Version 2.1 at centered position

Table 4.3.1 Parameters values of version 2.1

| Case | 2.1.1 | 2.1.2 | 2.1.3 | 2.1.4 | 2.1.5 | 2.1.6 |
|------|-------|-------|-------|-------|-------|-------|
| L1 | 40 | 40 | 40 | 30 | 40 | 50 |
| L2 | 40 | 40 | 40 | 20 | 30 | 40 |
| A1 | 10 | 15 | 20 | 10 | 15 | 20 |
| A2 | 10 | 15 | 0 | 10 | 10 | 15 |

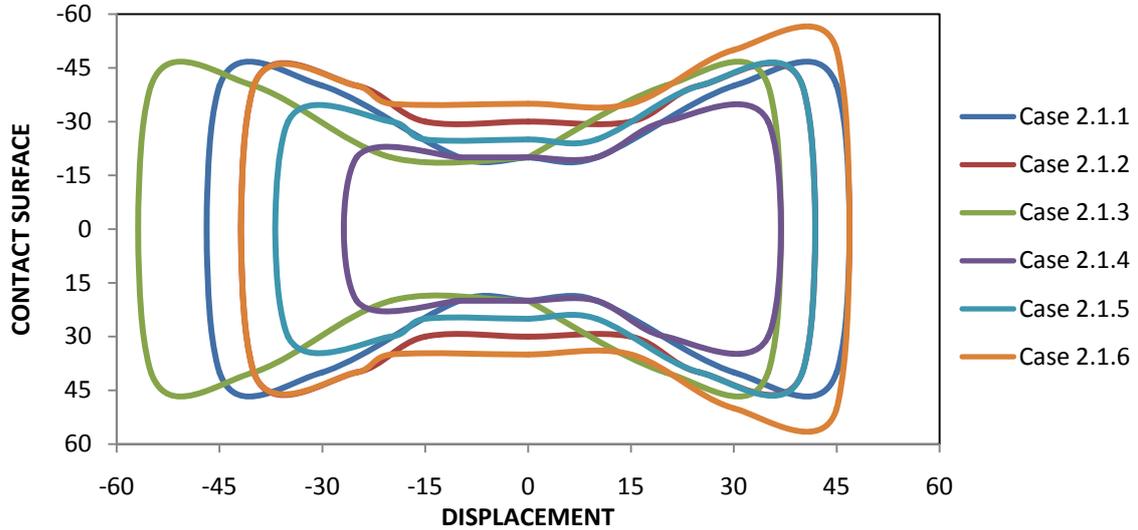


Figure 4.3.4 Contact surface of version 2.1

As can see in *Figure 4.3.4* this version 2.1 allows to adapt the behavior at each side and center of the displacement. It is possible to vary the contact surface at low displacements and obtain different values at both sides.

Version 2.2

In this second version, the two side pads have been reduced and separated from the central pad not to obtain contact between them at the central position as show *Figure 4.3.5*.



Figure 4.3.5 Version 2.2 at centered position

Table 4.3.2 Parameters values of version 2.2

| Case | 2.2.1 | 2.2.2 | 2.2.3 | 2.2.4 |
|------|-------|-------|-------|-------|
| L1 | 60 | 50 | 40 | 40 |
| L2 | 25 | 20 | 30 | 30 |
| A1 | 20 | 5 | 15 | 5 |
| A2 | 5 | 5 | 15 | 10 |

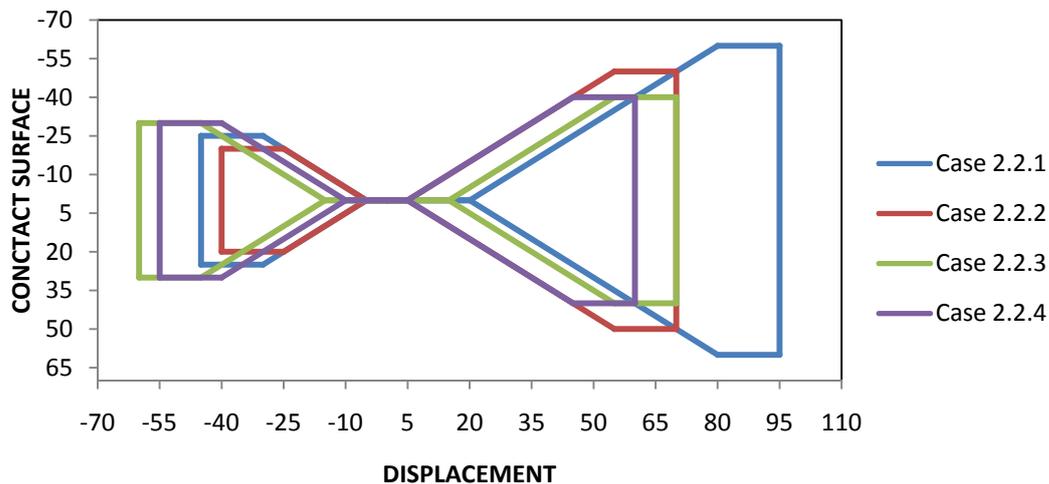


Figure 4.3.6 Contact surface of version 2.2

Version 2.2 seems to be the same as version 2.1 but in this case one can obtain an off contact surface at low displacements as can be seen in Figure 4.3.6. The slope at each curve is always the same during transition (45°), which represents the contact increment of one surface over the other. This is an important aspect to take into account in order to be related to the friction force.

The first version can be named as *low-high* behavior, whilst the second one as *on-off*. Thus, the best version is the second which offers

4.3.1 Conclusions

As conclusion of this first stage of development, the final version selected for total implementation was number 2.2. Said contact surface behavior offered better promising results than the first version, in order to obtain as much variable damping force as possible.

4.4 Second stage of development

4.4.1 Standard damper

Once analyzed the two proposed concepts it was decided that the best which fits better all the requirements was concept 2 (see apart 4.2.2) in its version 2.2, capable to achieve on-off contact surface behavior. It has already said that is only necessary

damping during resonance range speed, so the rest of the cycle damper works in its off stage.

Once selected the final concept, one must be sure that the concept is able to adapt to the available damper model, thus, in this case is necessary to change the shape of the concept, specifically the position of the pads and the surfaces of friction.

The available damper model belongs to the basic sort of damper from Suspa, mounted inside Asko's washing machines. This damper is formed by a plain housing in which is mounted a spring and the holder with the friction pad, as can be seen in *Figures 4.4.1.1-2*. In this case, the piston need not be attached to the holder owing to both the spring and the force over the piston rod transmitted by the washing machine, allowing ease of mounting.

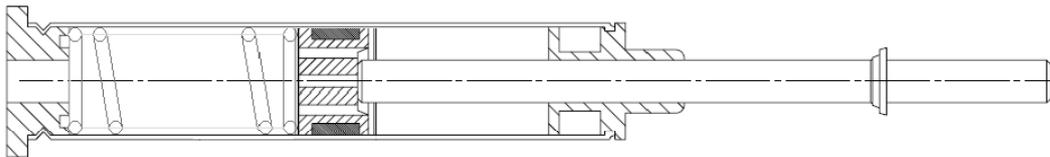


Figure 4.4.1.1 Available normal friction damper

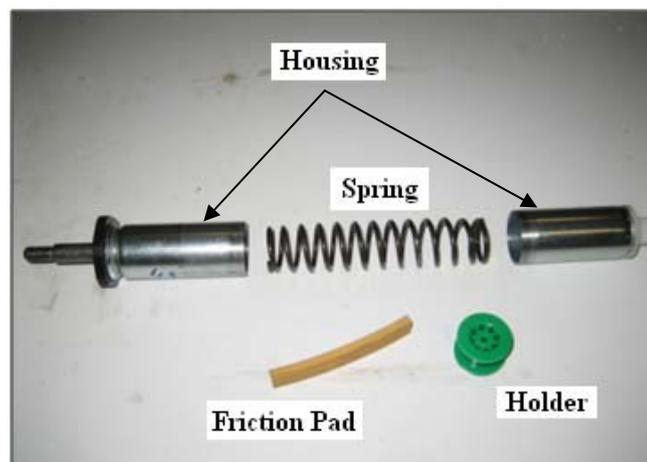


Figure 4.4.1.2 Internal parts of the standard friction damper

As final solution, version 2.2 can be modified and adapt maintaining the same concept.

4.4.2 Input data

Based on studies carried out by ASKO Appliances AB showing the amount of water used for each range of dry load, it can be said that the most common value of the dry clothes used in washing machines at home is about 2.5 Kg, what represents 6.9 liters of water. In order reduce problems in future tests, regarding high forces during spinning, it has been selected as final load the value of 4.8 Kg. The results of the amount of water used for each dry load with cotton sheets are shown in *Table 4.4.2.1*.

Table 4.4.2.1 Nominal amount of water used for each dry load with cotton sheets

| Dry Clothes (kg) | Water (l) |
|------------------|-----------|
| <1,5 | 4,7 |
| 1,5-2,5 | 6,9 |
| 2,5-3,5 | 9,1 |
| 3,5-4,5 | 11,1 |
| 4,5-5,5 | 13,5 |
| 5,5-6,5 | 15,3 |
| 6,5-7,5 | 18,3 |
| >7,5 | >19,1 |

4.4.3 Prototype model

The final prototype damper was manufactured and mounted from the basic friction damper, according to all the design specifications which enclose all development stages. In *Figure 4.4.3.1* is showed the new low-cost damper prototype once mounted.

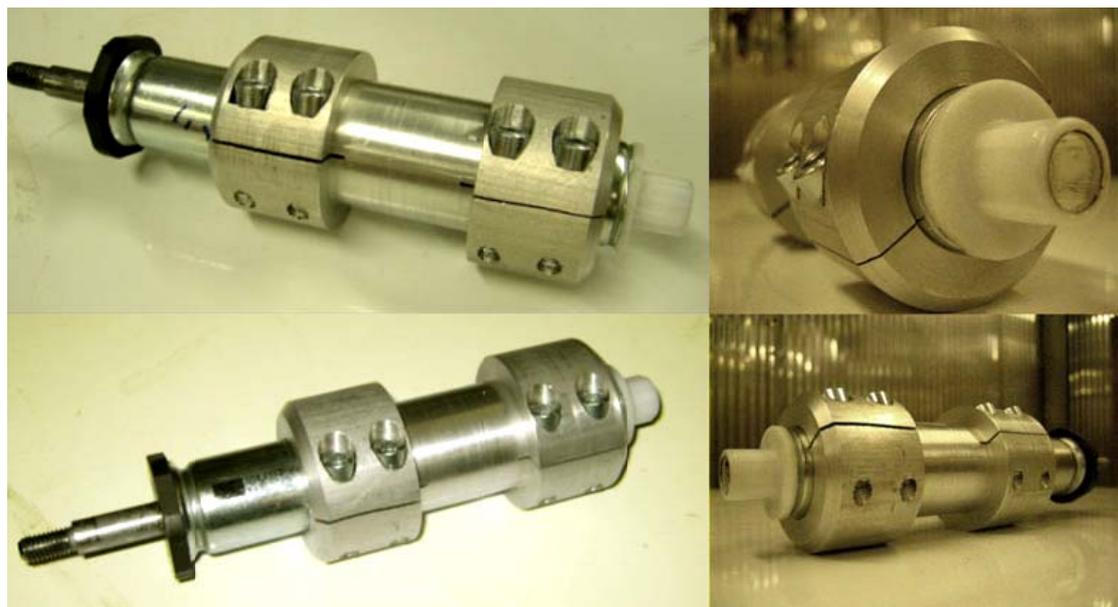


Figure 4.4.3.1 New LCD prototype

5 Laboratory Studies

At the department of Applied Mechanics at Chalmers a whole machine test rig for dampers was developed in cooperation between ASKO Appliances AB and Chalmers University of Technology. As can be seen in *Figure 5.1*, the damper is mounted in said position and due to the disc rotation, a translational movement is obtained. This movement is measured using a displacement sensor. On the top, there is placed a load cell, which measures the damper force. It is based on a standard test rig used for damper classification, but made a bit more rigid and adapted with a frequency converter to cover a frequency range of 0-30 Hz. The possible stroke setting is 0 to 50mm.

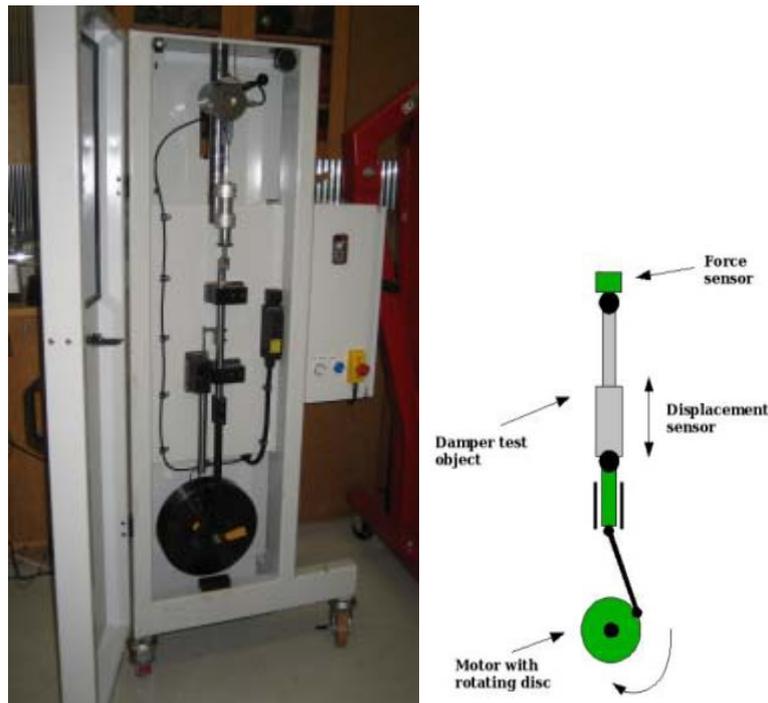


Figure 5.1 Damper test rig and sketch. Extract from [29]

The washing machine test rig was designed to measure the vertical force output from the damper shaft and relative position of the housing relative to the holder-pad-shaft group. Data acquisition was made with a PC using LabView, and then processed in MATLAB.

5.1 Basic friction damper model

Preliminary tests were performed in the timeframe while final prototype was manufactured. These tests consisted of figuring out the damping behavior of the standard damper used in the washing machine. For that, it was used the test rig disposed at the lab.

The tests were done using two types of dampers at different spin speeds: 15, 100, 300, 600 and 1200 rpm. Said dampers offers different friction forces and are used by ASKO in their washing machines. Marked with black and red reference colors, which mean two range of friction force, 32.5 ± 12.5 N and 32.5 ± 12.5 N respectively. In *Figure 5.1.1* one can see both behaviors at each spin speed.

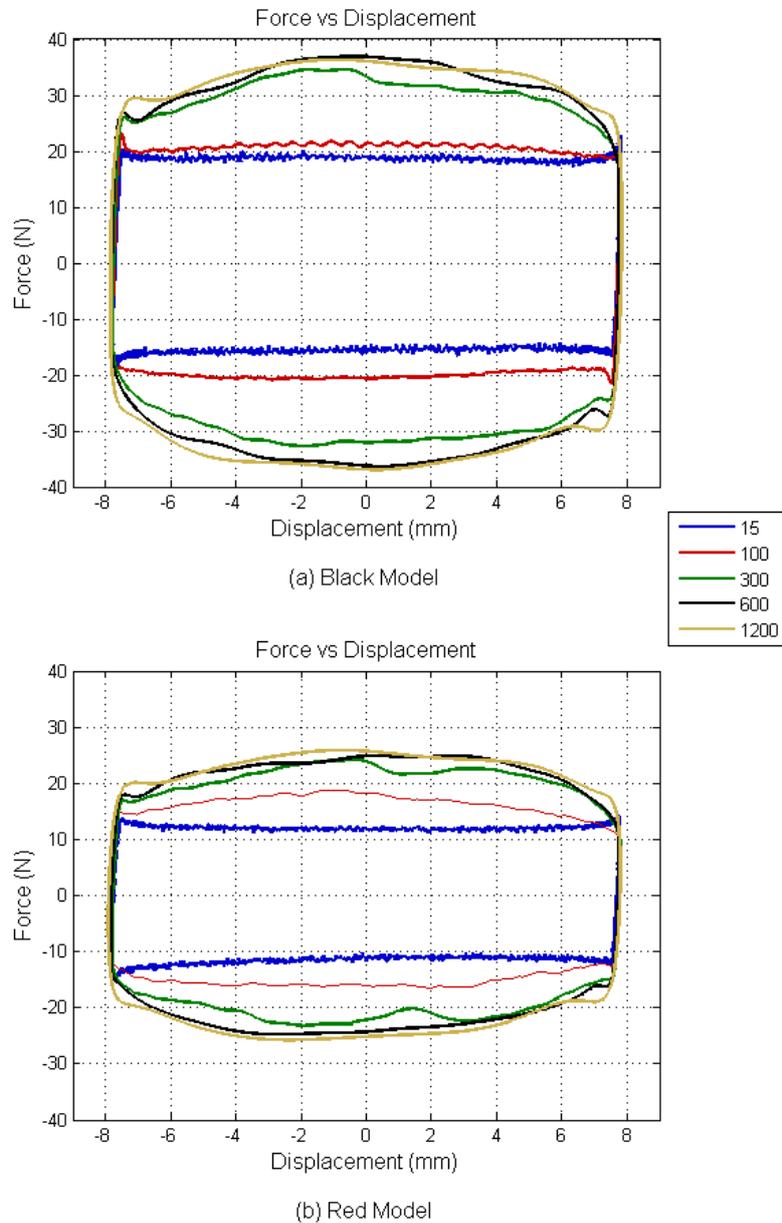


Figure 5.1.1 (a) Black and (b) Red basic friction dampers

5.2 Prototype model

The following tests were performed using the new LCD prototype. In order to select the best prototype, four models were manufactured with different internal parts configurations.

5.2.1 Schedule of studies

The studies were carried out for each version to show the damping behavior at different speeds: 15, 100, 300, 600 and 1200 rpm. The first speed (15 rpm) shows the detailed behavior, in which is possible to notice small anomalies during the whole work operation. The second one (100) is the speed used by ASKO to classify their basic friction dampers, so is useful to compare between them. The second and third speeds (100 – 300) shows the behavior in a speed range wherein the resonant frequency of the washing machine (about 200 rpm) is reached. Finally, the last speeds of 600-1200 rpm show the transition to the high spin speed.

This first studies showed the damping behavior, done with a high stroke value. As has been mentioned before, in these studies were used the four configuration models.

In the second studies, the stroke was reduced from high to low value in order to show the behavior at said low strokes, using the same models. To conclude, about 70 different tests were performed to figure out the behavior of the new prototype.

5.2.2 High Stroke Damping

As can be seen in *Figure 5.2.2.1* the damping behavior seems close to the “butterfly” behavior predicted at the first stage of the development. On the other hand, the dependency between force and velocity is showed. It can be also seen the differences using the four different internal configurations.

The main abnormality seen were the peaks formed between the lower and the higher friction zone as show the arrows in figures (c) and (d). Said increase of friction force can be understood due to collisions between internal parts. As shown *Figure 5.2.2.1* at low displacements the force is not zero, as the kinematic model predicted.

From all these models, the best damping behavior for using in washing machines belongs to *Model 3*, due to:

- Being the most precise related to the change between low-high friction, using 45° of slope.
- Being the one which produce more damping, using the white holder.

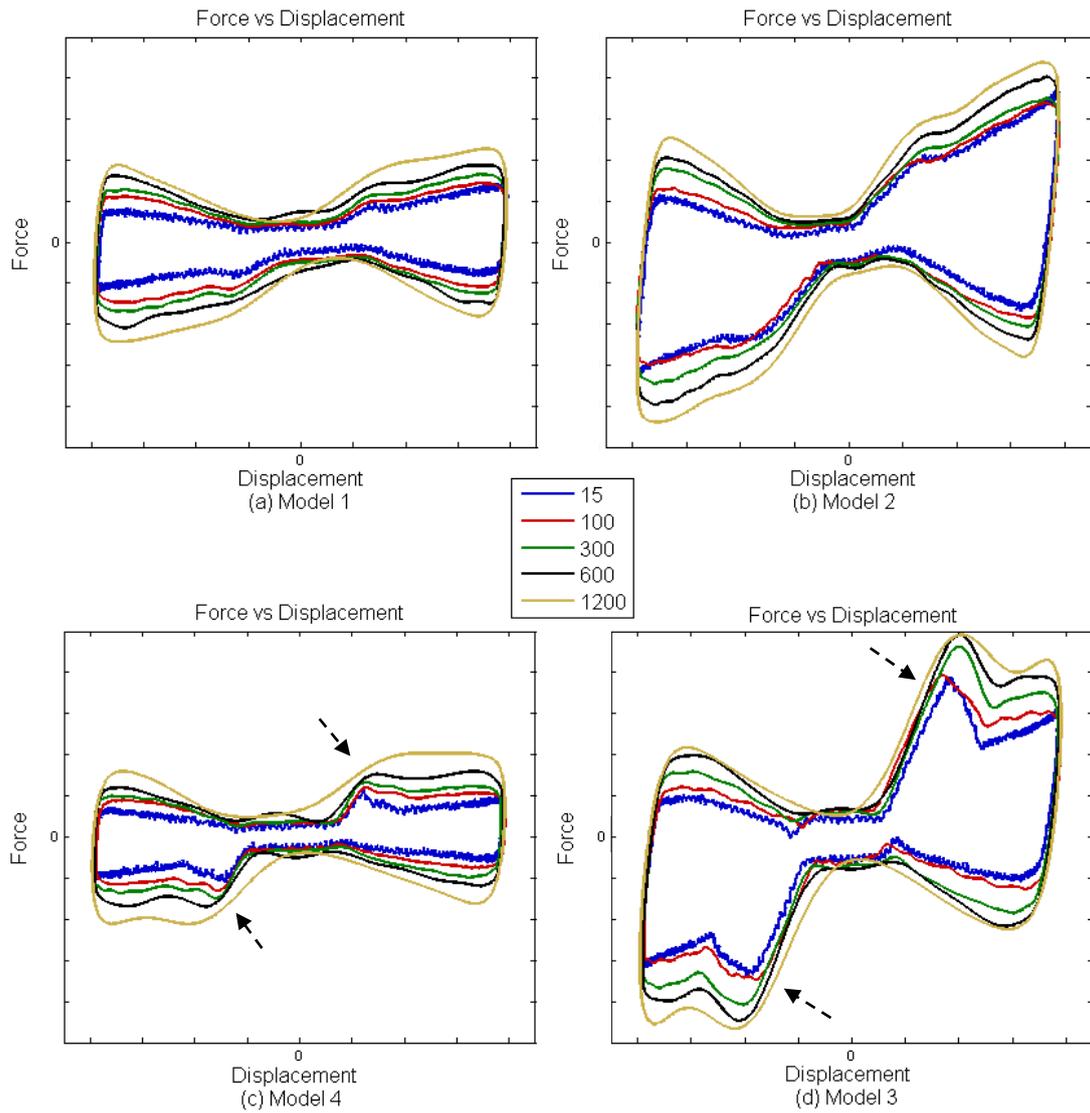


Figure 5.2.2.1 High stroke damping

5.2.3 Low Stroke Damping

The following low stroke damping studies show the behavior at low stroke. The methodology was the same as before. Several tests were performed using the four available prototype models with different configurations at several spin speeds.

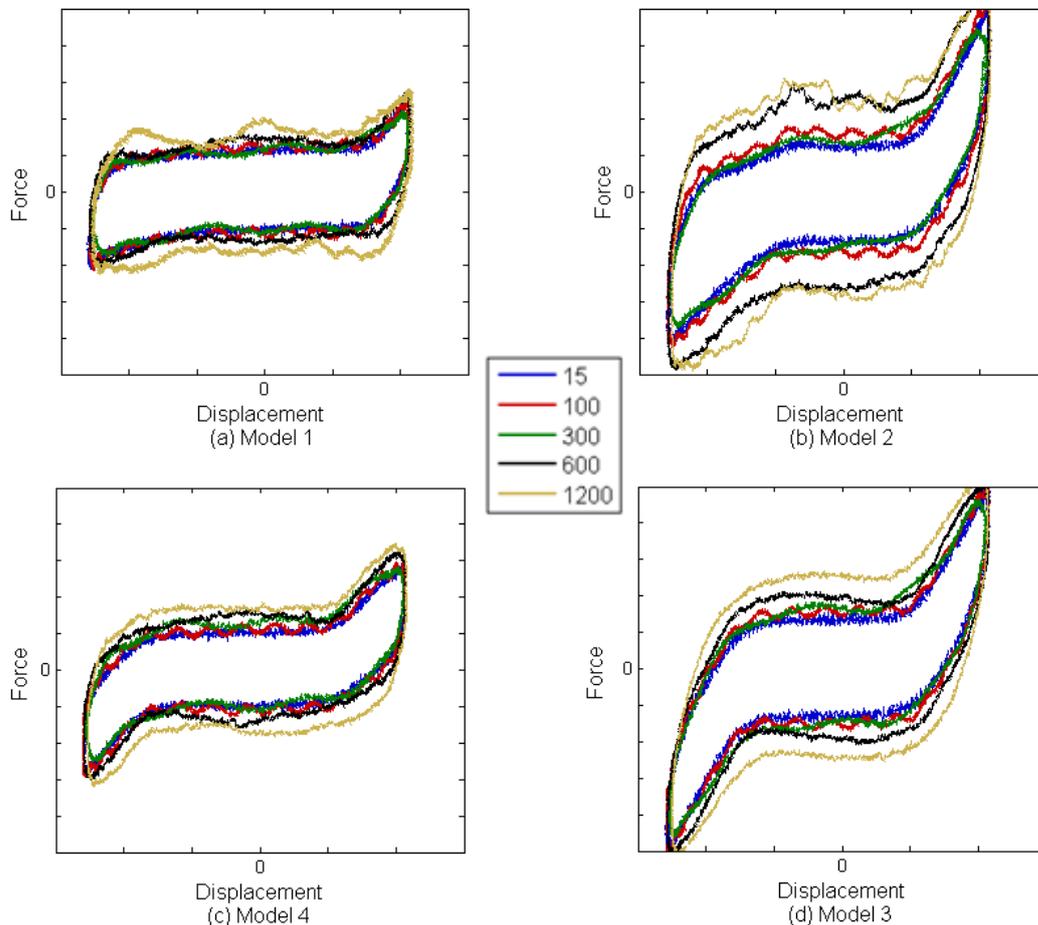


Figure 5.2.5.1 Low stroke damping

As can be seen in *Figure 5.2.5.1*, models 2 and 3 offer more damping than models 1 and 4 at the end of the stroke due to internal configurations. Also the friction force during said low stroke can be considered constant. So, in this zone, the new prototype works as a standard friction damper, with a low value of damping.

5.2.4 Final analysis

In order to compare all tested models, the damping behavior at 100 rpm is the most representative. In the *Figure 5.2.4.1* can be seen the four LCD models against the two standard models: black and red. It can be observed the differences between all behaviors. Focusing on model 3 and black standard damper, both damping behaviors are represented as shows *Figure 5.2.4.2*, where can be seen the friction force reduction at low displacements in the new LCD prototype.

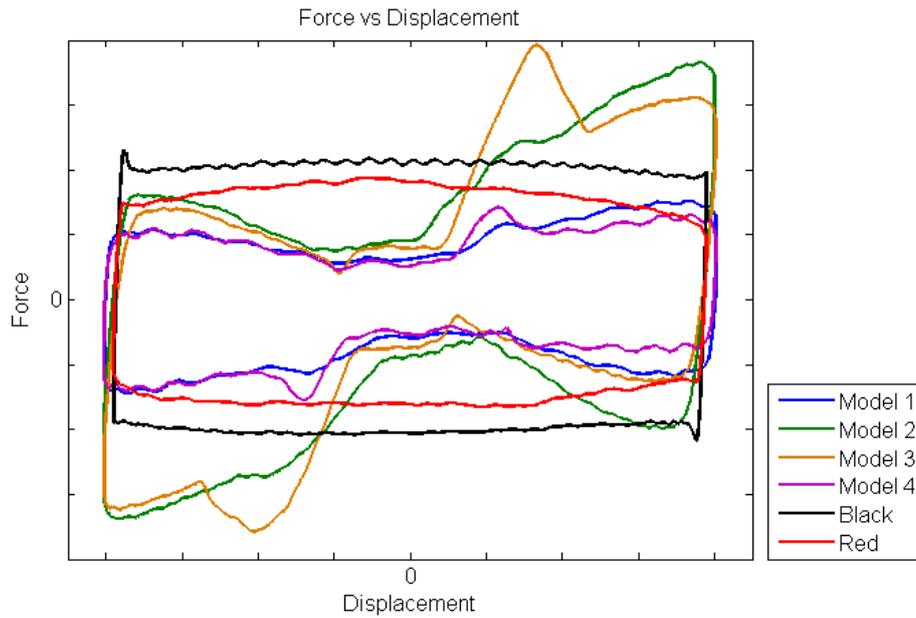


Figure 5.2.4.1 Comparison of all models at 100 rpm

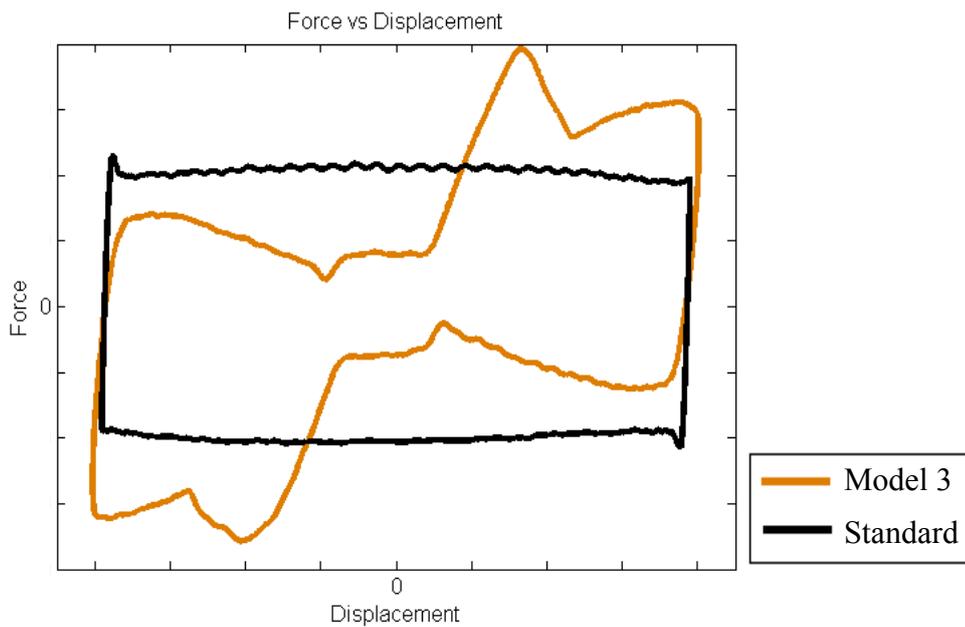


Figure 5.2.4.2 Comparison between standard and selected Damper

On the other hand, the comparison was also done analyzing two concepts based on the amount of damping and the energy dissipation of each damper model. These two quantitative methods are described hereinafter.

Amount of damping

A method to measure the adaptivity of damping of each model can be done taking into account the index that shows the difference of force between high and low damping

seen in the “butterfly” behavior of the new damper model. On the other hand, this method allows us to show the relation between the parameters of which depends the friction force.

The parameters needed are the “peak to peak” ptp F_{max} and ptp F_{min} , as show *Figure 5.2.4.3*. The lower value of \mathcal{F}_1 the better, in order to obtain the minimum value, which means, the difference between both forces (low and high) is the highest.

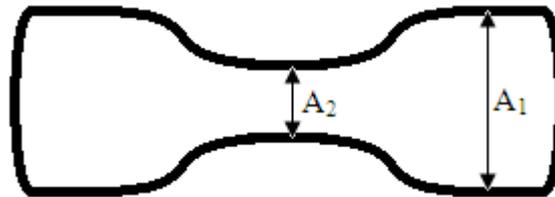


Figure 5.2.4.3 Cost function parameters

$$\mathcal{F}_1 = \frac{ptp F_{min}}{ptp F_{max}} = \frac{A_2}{A_1} \rightarrow \text{Minimum} \quad (5.2.4.1)$$

Table 5.2.4.1 Cost function values

| Model | 1 | 2 | 3 | 4 |
|----------------|-----------------|-----------------|-----------------|-----------------|
| r.p.m. | \mathcal{F}_1 | \mathcal{F}_1 | \mathcal{F}_1 | \mathcal{F}_1 |
| 15 | 0,416 | 0,452 | 0,325 | 0,470 |
| 100 | 0,423 | 0,384 | 0,325 | 0,434 |
| 300 | 0,607 | 0,358 | 0,407 | 0,653 |
| 600 | 0,514 | 0,308 | 0,387 | 0,4 |
| 1200 | 0,444 | 0,394 | 0,333 | 0,441 |
| Average | 0,481 | 0,379 | 0,355 | 0,480 |

In *Figure 5.2.4.4* one can see the cost function value at each speed for the four models represented in the *Table 5.2.4.1*. The two best models regarding \mathcal{F}_1 were 2 and 3, with the lowest values from 0.3 to 0.45. The best model was the third, in which the average value of \mathcal{F}_1 was 0.355, so this was the most versatile model regarding said cost function. Also, the model 3 was the one in which the value of \mathcal{F}_1 maintains more constant, that means the model behaves apparently in the same way at different speeds.

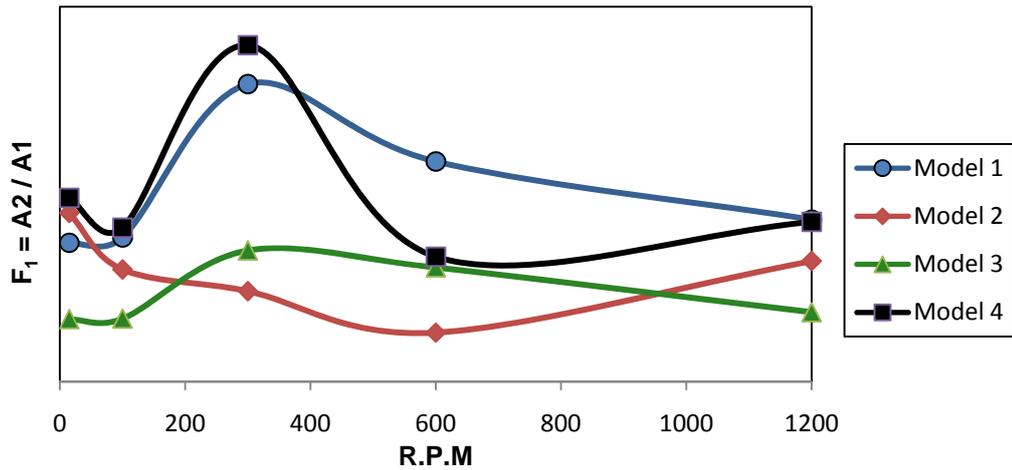


Figure 5.2.4.4 Cost function \mathcal{F}_1

Energy dissipation

The main effect of damping is to remove energy from the system. Energy in a vibrating system is either dissipated into heat or radiated away. As can be read in [25] regarding vibration analysis, the loss of energy from an oscillatory system results in the decay of amplitude of free vibration. In steady-state forced vibration, the loss of energy is balanced by the energy that is supplied by the excitation. Energy dissipation is usually determined under conditions of cyclic oscillations. Depending on the type of damping present, the force-displacement relationship when plotted can differ greatly. In all cases, however, the force-displacement curve encloses an area referred to as the hysteresis loop, which is proportional to the energy lost per cycle as can be seen in Figure 5.2.4.5. The energy lost per cycle due to a damping force F_d is computed from the general equation:

$$W_d = \oint F_d \cdot dx \quad (5.2.4.2)$$

The energy dissipated per cycle is then given by the area enclosed by the curve.

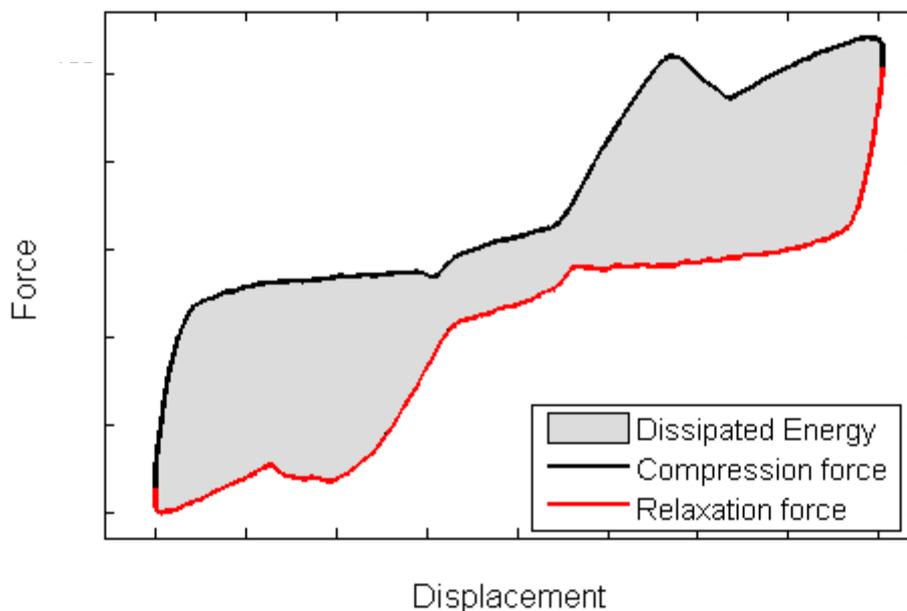


Figure 5.2.4.5 Enclosed area by the curve

In *Figure 5.2.4.6* the dissipated energy per cycle is represented at each rpm for all the dampers models studied. The values shown in *Table 5.2.4.2* are the energy loss calculated from equation (5.2.4.2)

Table 5.2.4.2 Dissipated energy in Joules (J) of each damper model at several rpm

| Model | 1 | 2 | 3 | 4 | Black | Red |
|---------|--------------|--------------|-------------|--------------|--------------|--------------|
| r.p.m. | | | | | | |
| 15 | 0,25 | 0,47 | 0,47 | 0,24 | 0,53 | 0,36 |
| 100 | 0,31 | 0,58 | 0,54 | 0,29 | 0,63 | 0,49 |
| 300 | 0,38 | 0,61 | 0,69 | 0,35 | 0,92 | 0,63 |
| 600 | 0,47 | 0,75 | 0,79 | 0,44 | 1 | 0,68 |
| 1200 | 0,55 | 0,91 | 0,91 | 0,55 | 1,03 | 0,71 |
| Average | 0,392 | 0,664 | 0,68 | 0,374 | 0,822 | 0,574 |

The model which dissipates more energy along the whole cycle is the normal black. Related to the new models, number 3 obtains the highest value of lost energy, closely followed by model 2. One should be into account that these values are per cycle, and in case of the new models, at low amplitudes the friction force is as less as possible, thus most of the lost energy comes from the high damping stage.

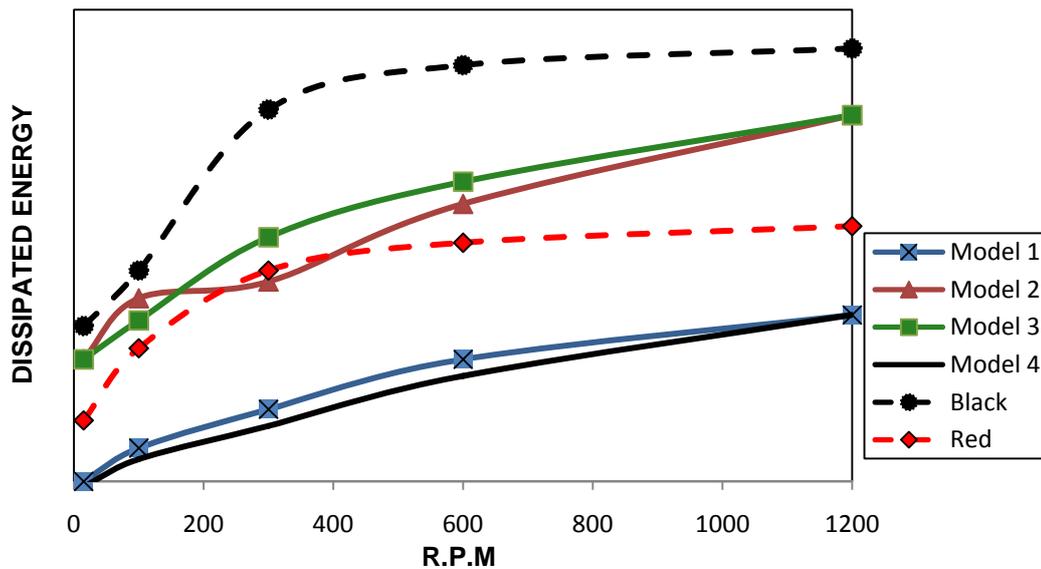


Figure 5.2.4.6 Dissipated energy

5.2.5 Conclusions

Several tests of the prototype models were performed in this part using the damper classification test rig. The new behavior was studied and analyzed at different scenarios and configurations in order to compare with the standard friction damper. Model 3 was the selected damper to be mounted in the washing machine and tested after the analyses done. The good behavior at low and high frequencies was the cause of its selection above the rest models. Also the last stage, in which the amount of damping and energy dissipation were analyzed, has contributed to clarify the best qualities of model 3.

5.3 Washing machine tests

At the department of Applied Mechanics at Chalmers a whole washing machine test rig was developed. As can be seen in *Figure 5.3.1*, it is based on a standard ASKO washing machine supported by four load cells on the bottom, thus forces at each damper location were measured. On the top, a Hall Sensor-3axes is placed in order to measure the displacement of the drum. The reference system used is placed in the same figure. All forces as well as displacements were processed using LabView and MATLAB. More details about the test rigs and other performed tests can be read at [26].

Once performed tests using the damper test rig and obtained the dynamic behavior of the prototype, several tests were done mounting four new prototypes, based on model 3, inside the washing machine test rig, as can be seen in *Figure 5.3.2*.

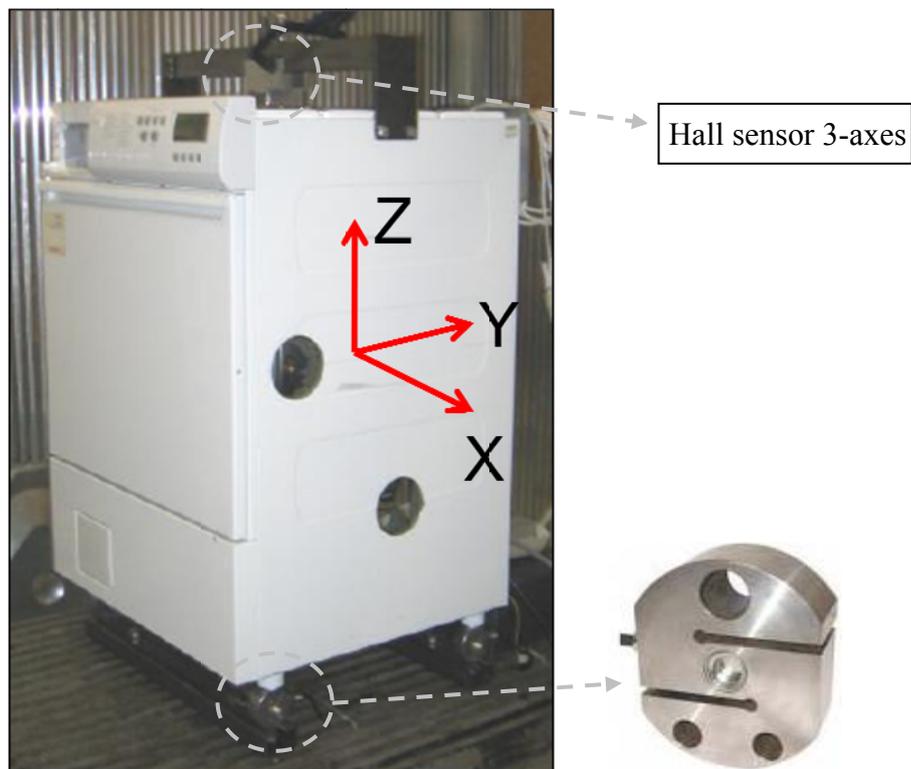


Figure 5.3.1 Washing machine test rig and load cell



Figure 5.3.2 Prototypes mounted in the washing machine test rig

5.3.1 Schedule of tests

The aim of these experiments is to compare the standard 4-struct washing machine configuration, in which the standard friction dampers are used, with the new prototype gap damper 4-struct disposal. In the *Table 5.3.1* one can see in detail the four different tests performed using 1000g-500g-300g of imbalance loads on front and spread out position inside the drum, see *Figures 5.3.1.1-2*. In order to mount the balanced load of 3.8-4.8 kg, a new anchoring system was used as shows the *Figure 5.3.1.3*. Several sheets of metal were fixed to the drum along all its length taking advantage of the drainage holes in the drum. This new system achieves a better balanced load disposal.

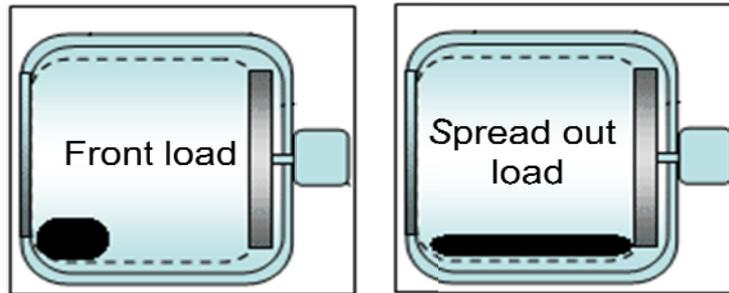


Figure 5.3.1.1 Front and spread out load disposal

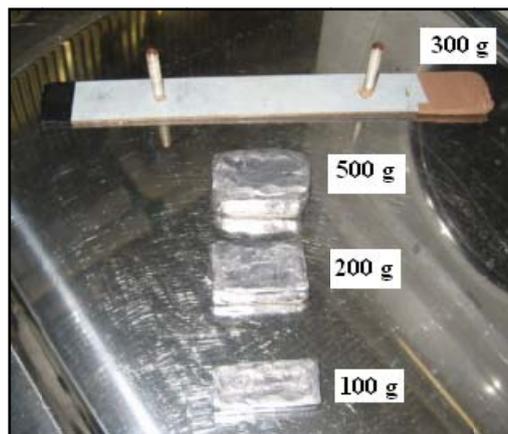


Figure 5.3.1.2 Imbalance loads



Figure 5.3.1.3 Disposal of metal sheets inside the drum

The tests started with the standard and prototype damper without any balanced load (0 Kg), adding the three imbalance load cases. After that, the balanced load was increased to 3.8 Kg and the inner parts of the prototype were modified obtaining three different configurations: 1, 2 and 3; in order to compare how said modifications affect. The next test done was using the concept prototype 3 against 4.8 Kg of balance load.

For the two firsts cases of imbalance load (1000g and 500g) the tests were done increasing the drum spin speed from 0 to 800 rpm, and in the other case (300g), the speed was increased to 1800 rpm. All these information is summarized in *Table 5.3.1.1*.

Table 5.3.1.1 Tests performed in washing machine

| Test | Damper Model | Balanced Load (Kg) | Spin Speed (rpm) | Imbalance Load | |
|------|--------------------------|--------------------|------------------|----------------|------------|
| | | | | Weight (Kg) | Position |
| 1 | Standard Friction Damper | 0-3.8 - 4.8 | 800 | 1000 | Front |
| | | | | 500 | Front |
| | | | 1800 | 300 | Spread out |
| 2 | Concept Prototype 1 | 0-3.8 | 800 | 1000 | Front |
| | | | | 500 | Front |
| | | | 1800 | 300 | Spread out |
| 3 | Concept Prototype 2 | 3.8 | 800 | 1000 | Front |
| | | | | 500 | Front |
| | | | 1800 | 300 | Spread out |
| 4 | Concept Prototype 3 | 3.8-4.8 | 800 | 1000 | Front |
| | | | | 500 | Front |
| | | | 1800 | 300 | Spread out |

5.3.2 Analysis of results

0 Kg balance load

Below can be seen the comparison between tests 1 and 2: standard friction damper vs concept prototype 1 damper with 0 Kg of balanced load. *Figure 5.3.2.1* shows the amplitude of displacement of each damper against the time and the spin speed. One can see that it was obtained a reduction of displacement with the new prototype in the three axes at low frequencies. This peak of displacement at low frequencies is due to the resonant frequency is reached at about 200 rpm. Regarding *Figure 5.3.2.2* which shows the amplitude of force, notice that with the new prototype damper the forces were increased.

So at these conditions of test 1 (1000g-front of imbalance load) the new prototype offers less amount of displacement at low frequencies but not as much as desirable of force reduction. This is reasonable because with 0 Kg of balance load using the prototype configuration 1, the damper works as a standard damper but with more friction force.

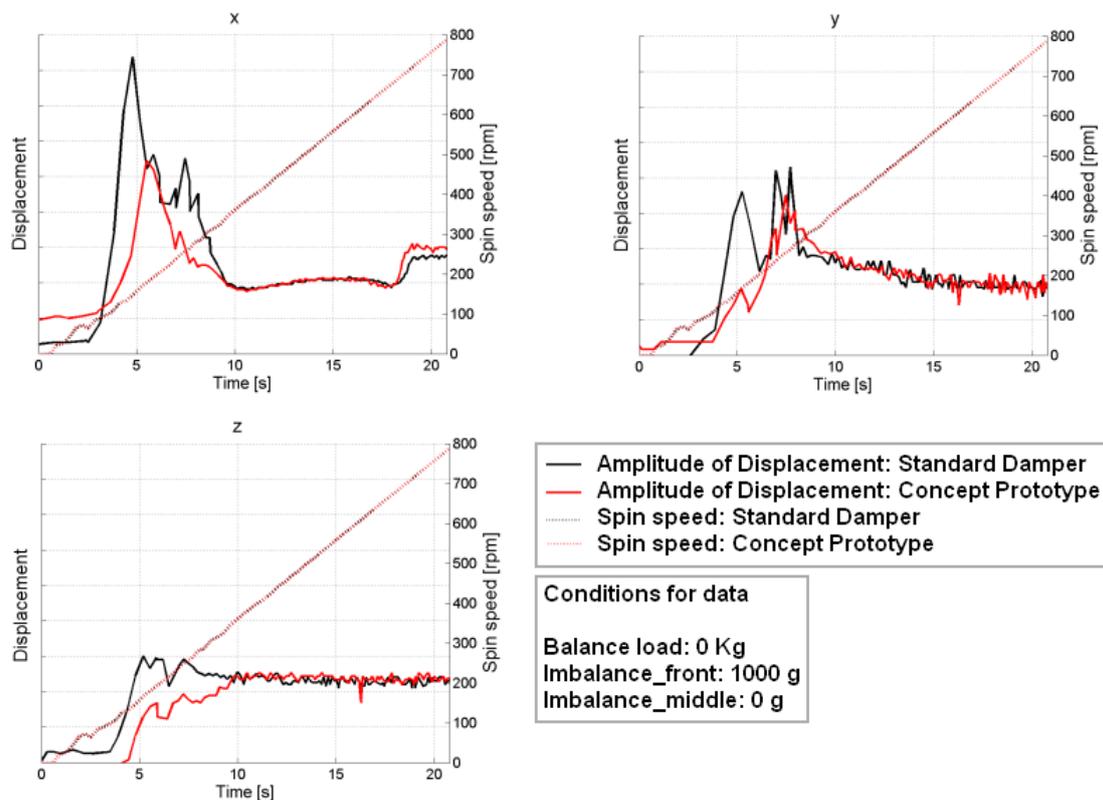


Figure 5.3.2.1 Amplitude of displacement with standard and prototype damper 1

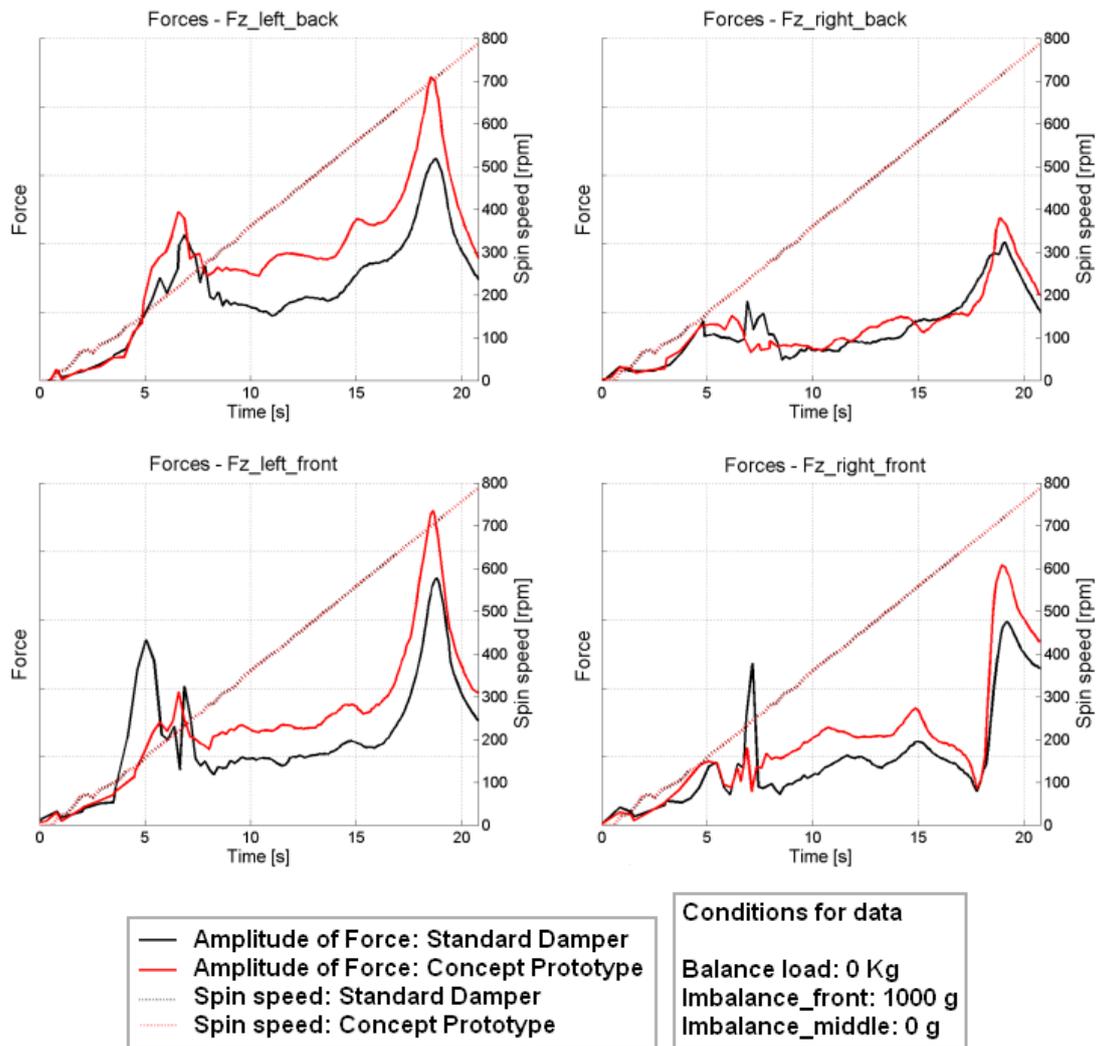


Figure 5.3.2.2 Amplitude of force with standard and prototype damper1

The next performed test was using 500 g of imbalance load in front position. As can be seen in *Figure 5.3.2.3*, the displacements of the prototype in the three axes were almost the same than the standard damper. Regarding forces, the prototype offered more damping as shown *Figure 5.3.2.4*.

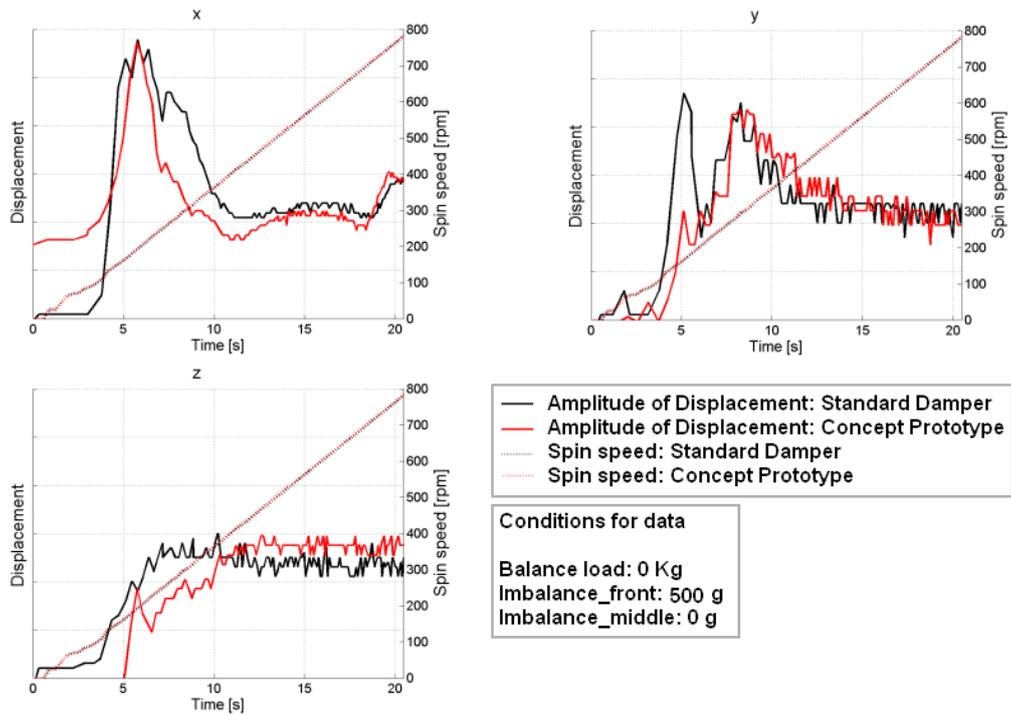


Figure 5.3.2.3 Amplitude of displacement with standard and prototype damper 1

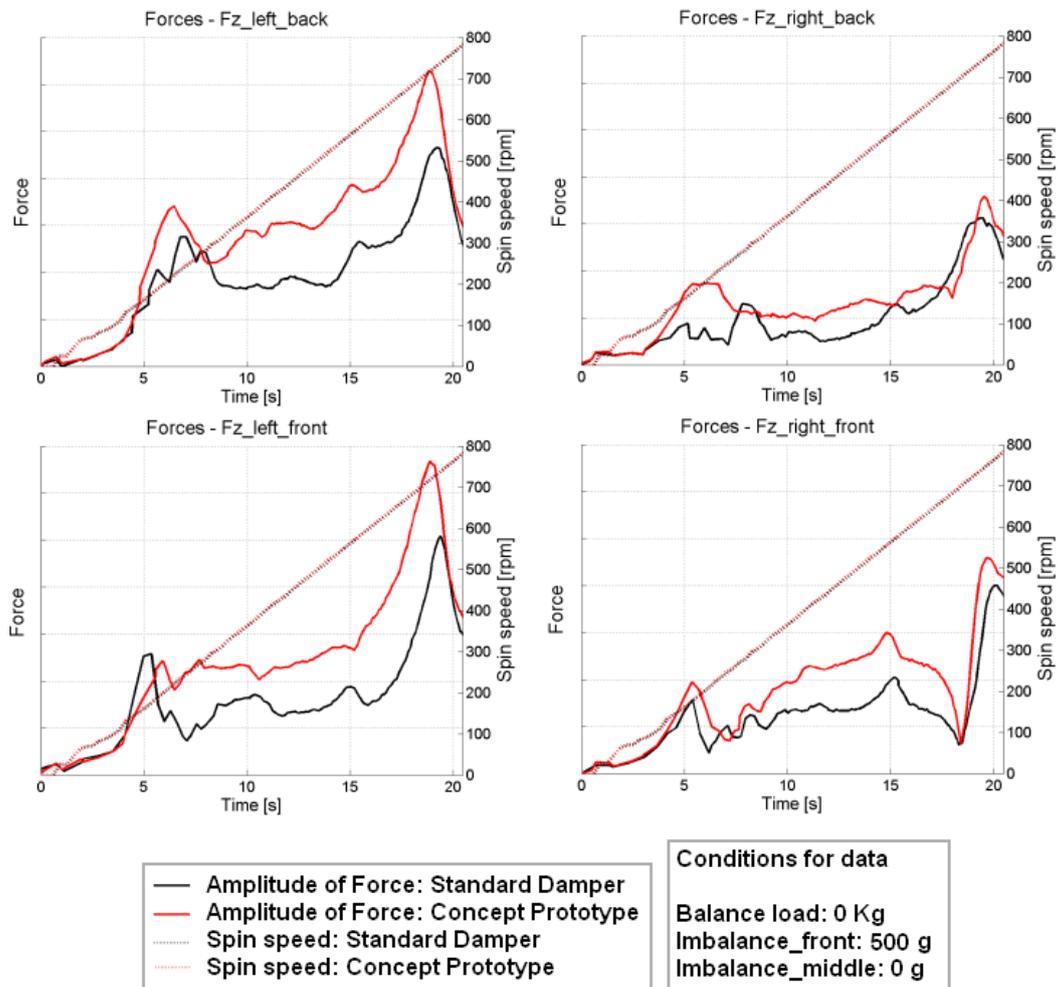


Figure 5.3.2.4 Amplitude of force with standard and prototype damper 1

The last test performed with 0 Kg of balance load was using 300 g of imbalance load spread out inside the drum. *Figure 5.3.2.5* shows the amplitude of displacement, where can be seen that the displacement in Z and Y axes were the same, and were increased in the X axis. Regarding forces, as seen before, the new prototype produced more damping than the standard damper at all frequencies.

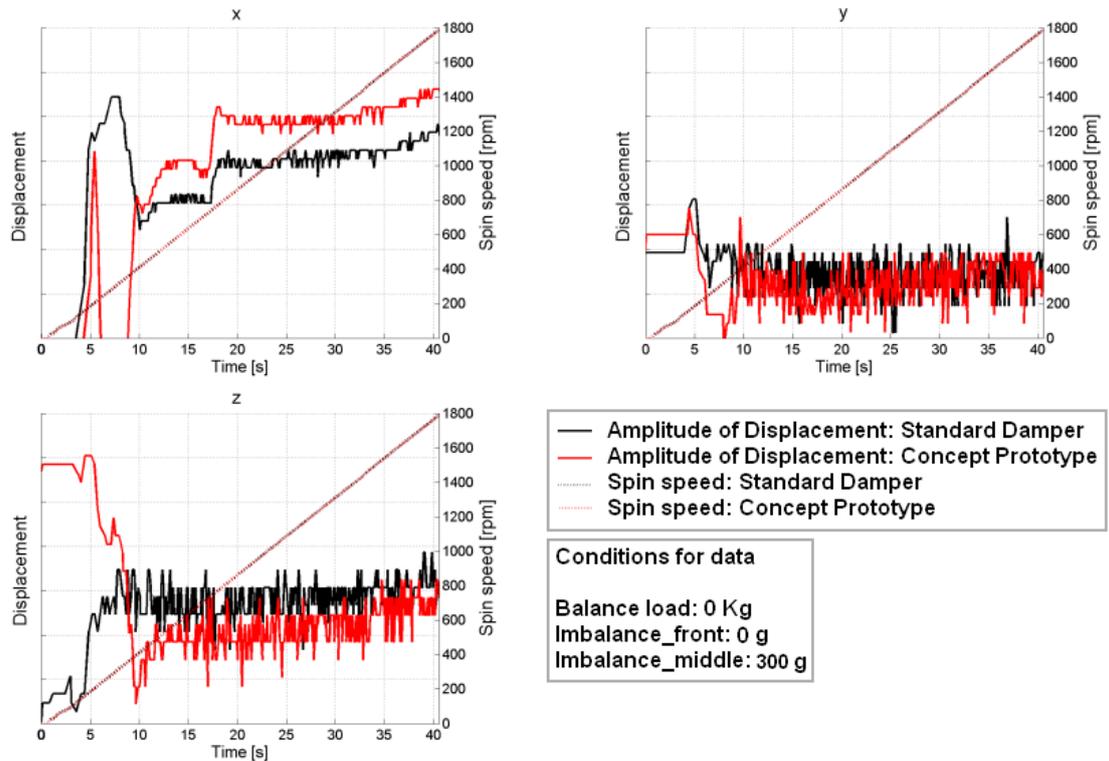


Figure 5.3.2.5 Amplitude of displacement with standard and prototype damper 1

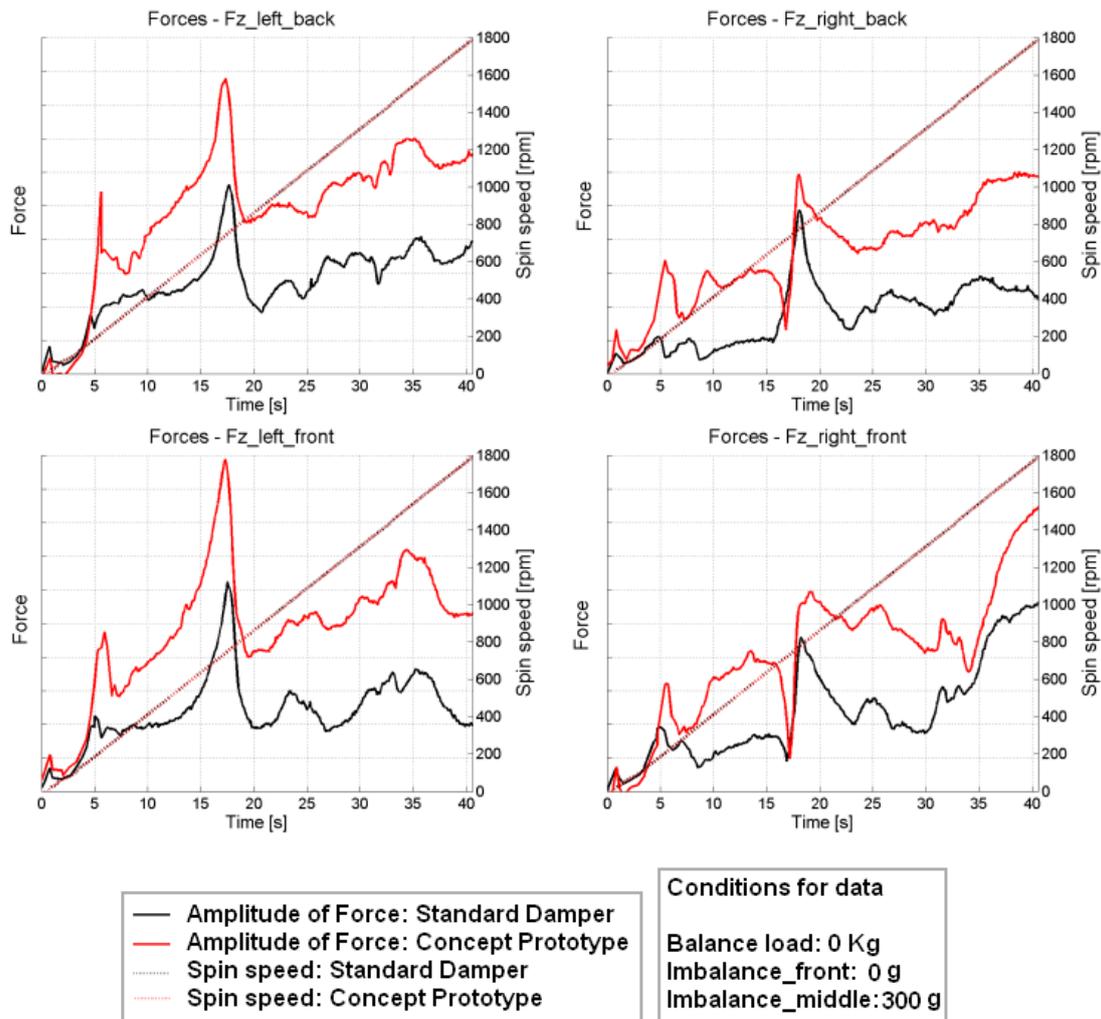


Figure 5.3.2.6 Amplitude of force with standard and prototype damper 1

3.8 Kg balance load

The last tests done were using 3.8 Kg of balance load comparing the standard damper and the concept prototype using three different internal configurations: 1, 2 and 3.

The results obtained were mostly the same as the one before with 4.8 kg. In the first test using 1000g of front imbalance load, see *Figures 5.3.2.7-8*, the prototype 3 obtained less displacement than with the others configurations, being similar than the standard damper. The forces did not suffer any major variations to the standard model.

The next test based on 500g of front imbalance load showed a high displacement of all prototypes at low frequencies, as shows *Figure 5.3.2.9*. On the other hand forces maintained a slightly value above the standard model. See *Figure 5.3.2.10*.

The last test done, with 300 g spread out imbalance load, shown the same displacement increasing as before, see *Figure 5.3.2.11*. At low frequencies the forces were a bit higher than the standard model but at high frequencies the forces were reduced considerably, as can be seen in *Figure 5.3.2.12*.

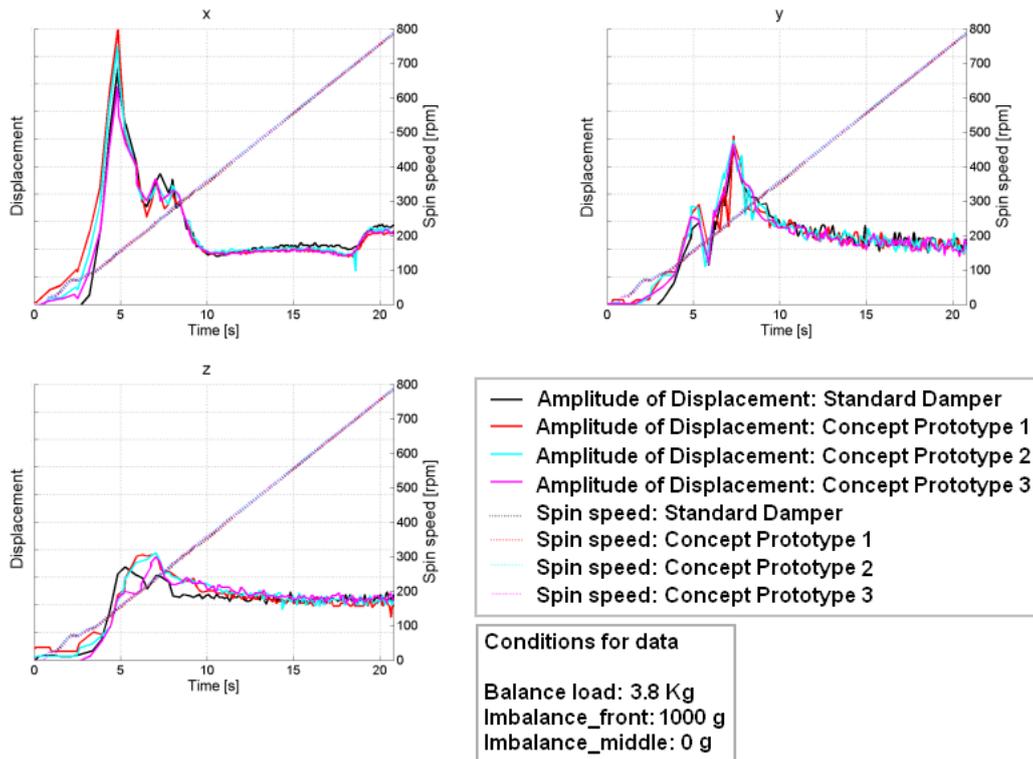


Figure 5.3.2.7 Amplitude of displacement with standard and prototype dampers 1-2-3

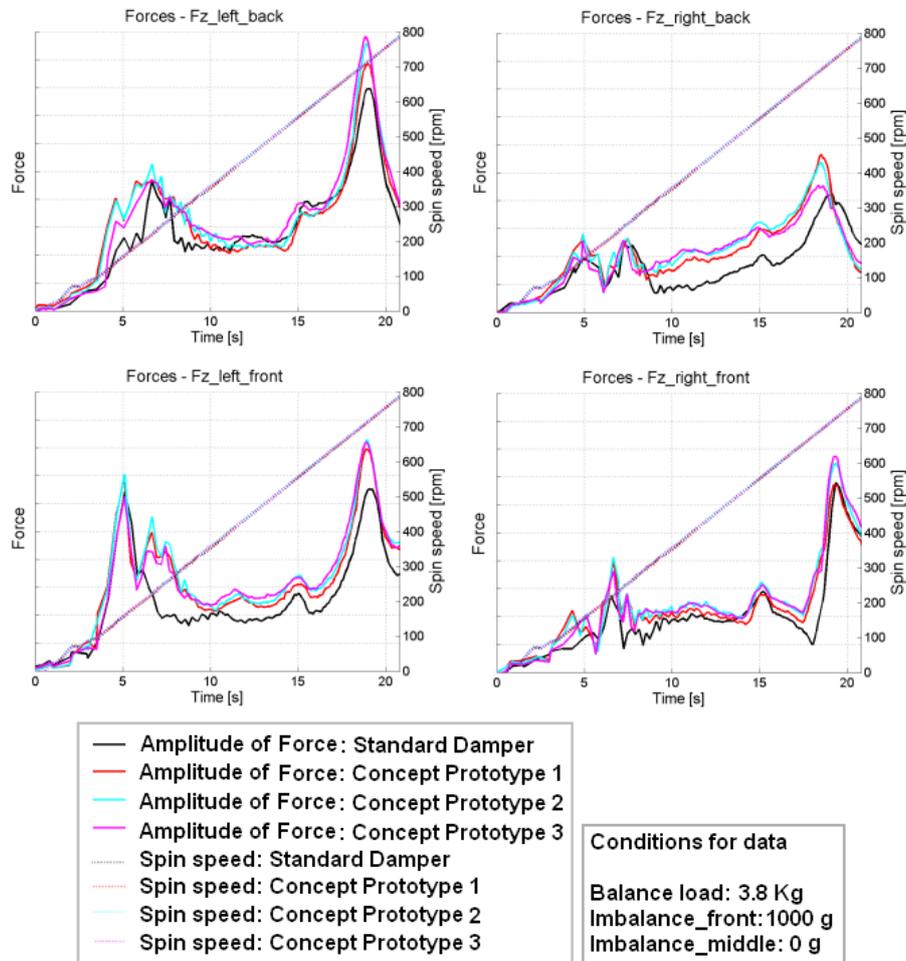


Figure 5.3.2.8 Amplitude of force with standard and prototype dampers 1-2-3

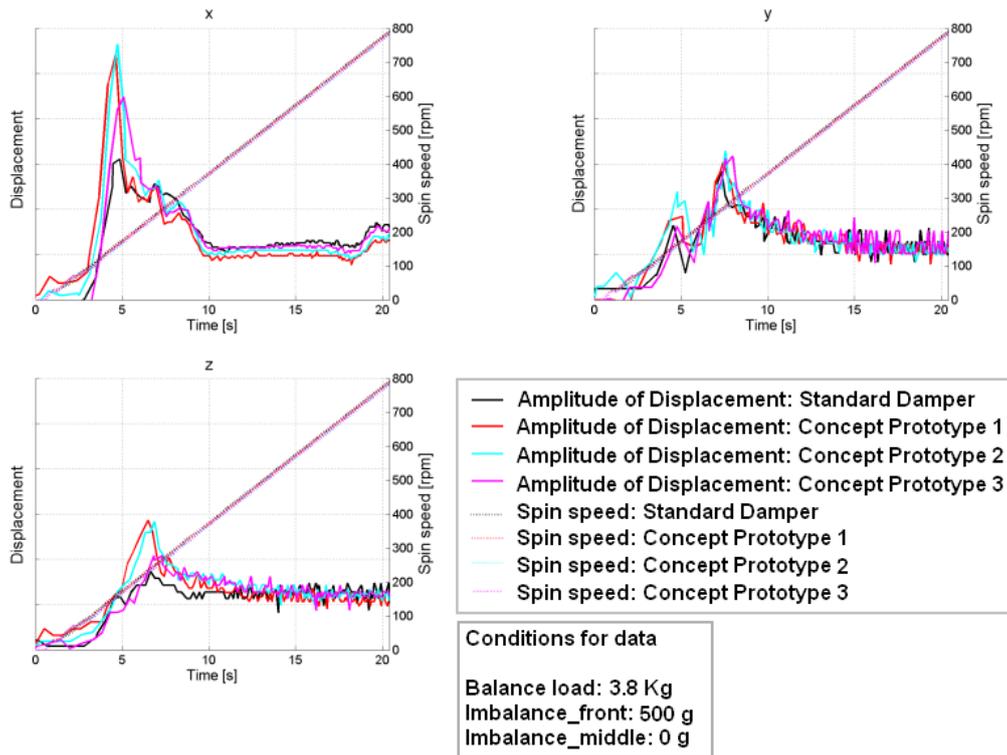


Figure 5.3.2.9 Amplitude of displacement with standard and prototype dampers 1-2-3

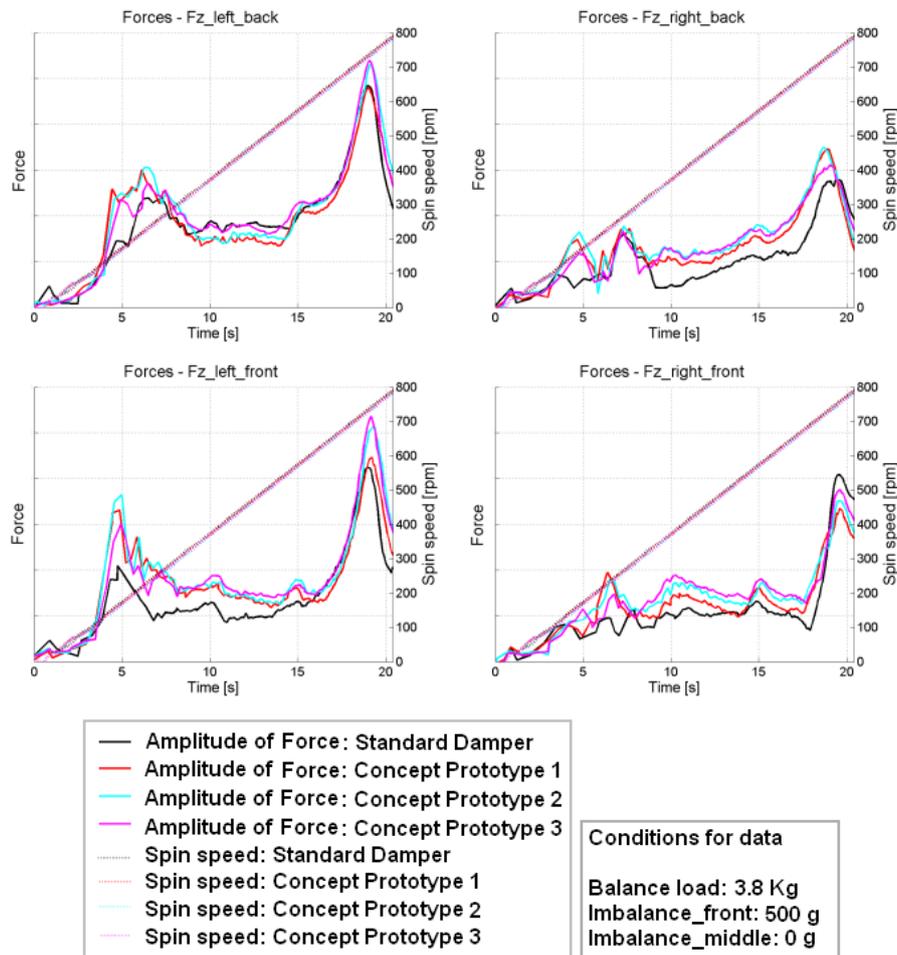


Figure 5.3.2.10 Amplitude of force with standard and prototype dampers 1-2-3

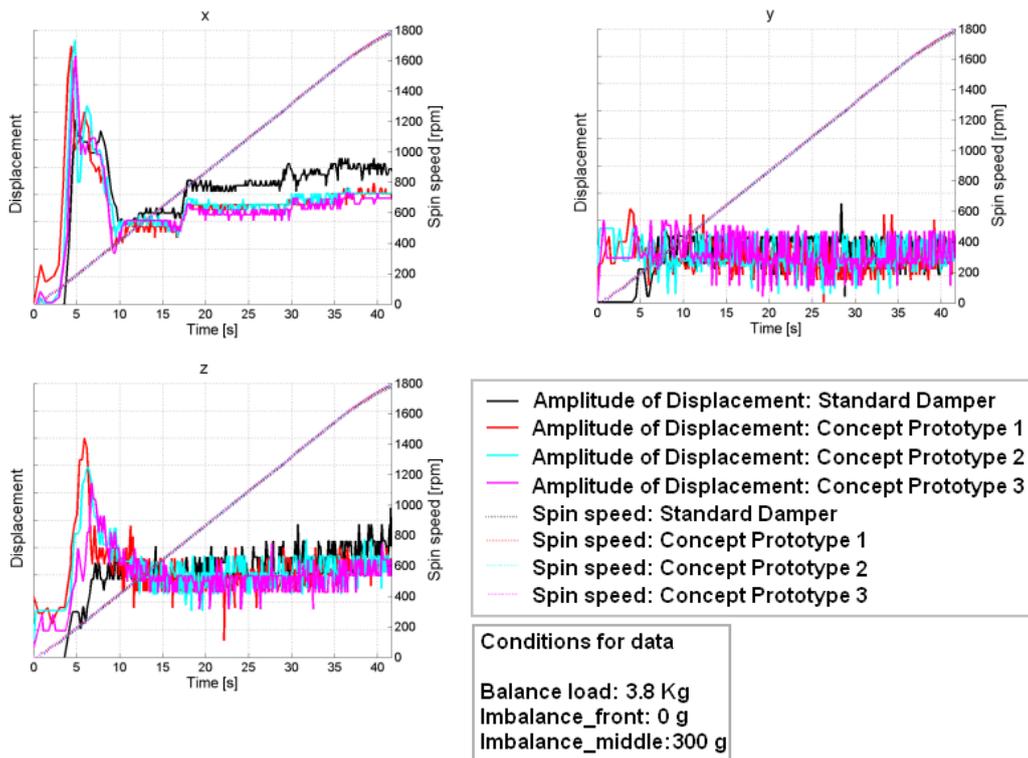


Figure 5.3.2.11 Amplitude of displacement with standard and prototype dampers 1-2-3

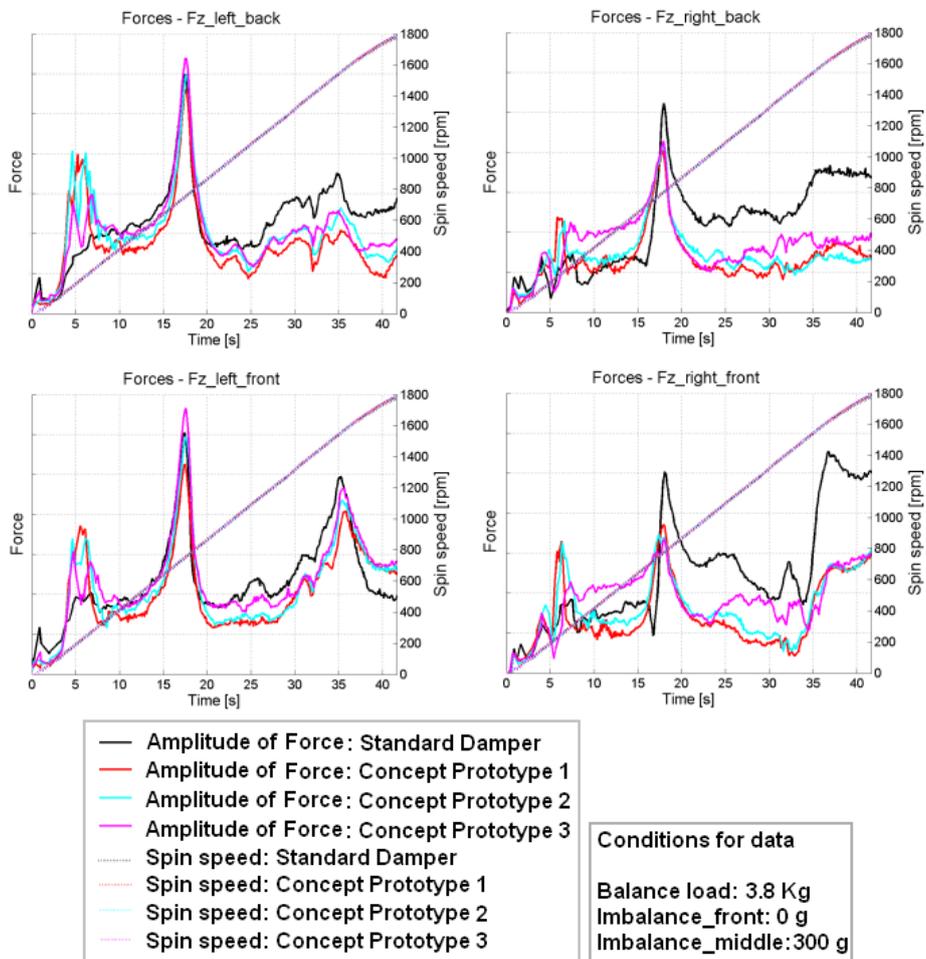


Figure 5.3.2.12 Amplitude of force with standard and prototype dampers 1-2-3

4.8 Kg balance load

In the following tests the standard damper and the concept prototype damper 3 were tested with 4.8 Kg of balance load. In *Figures 5.3.2.13-14* can be seen the test done with 1000 g of imbalance load, in which the displacements and forces were mostly the same. The prototype damper forces were higher at the upper-mid part of the frequency range.

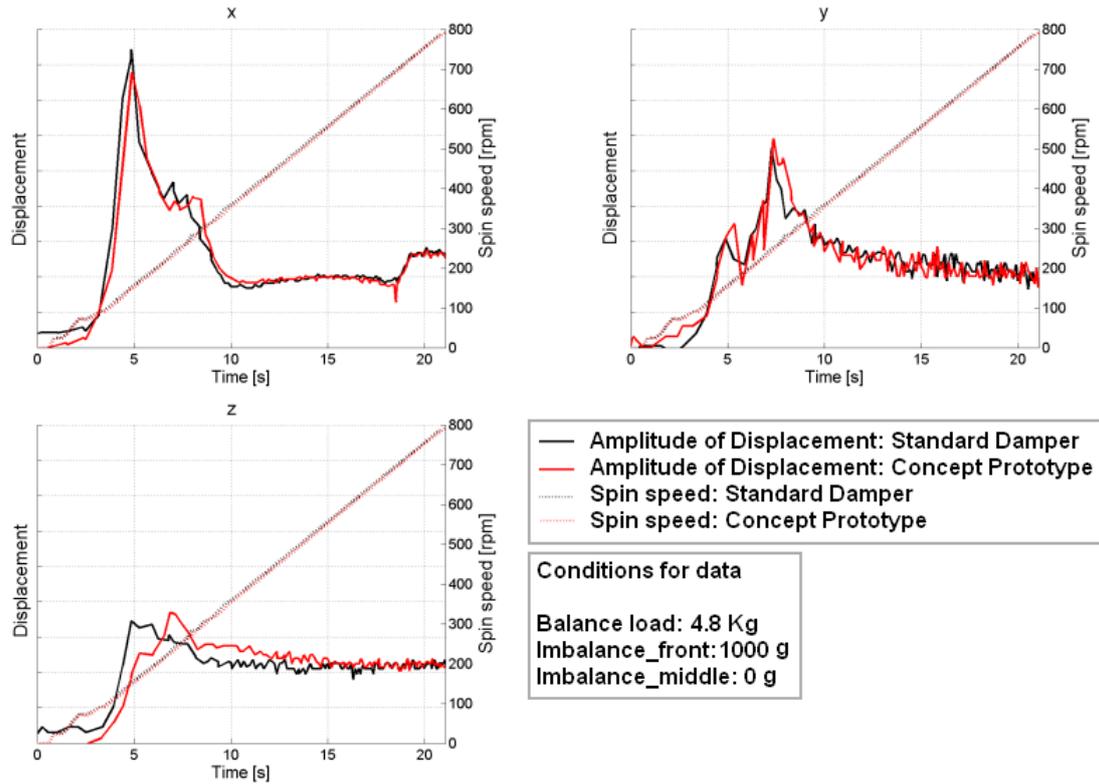


Figure 5.3.2.13 Amplitude of displacement with standard and prototype damper 3

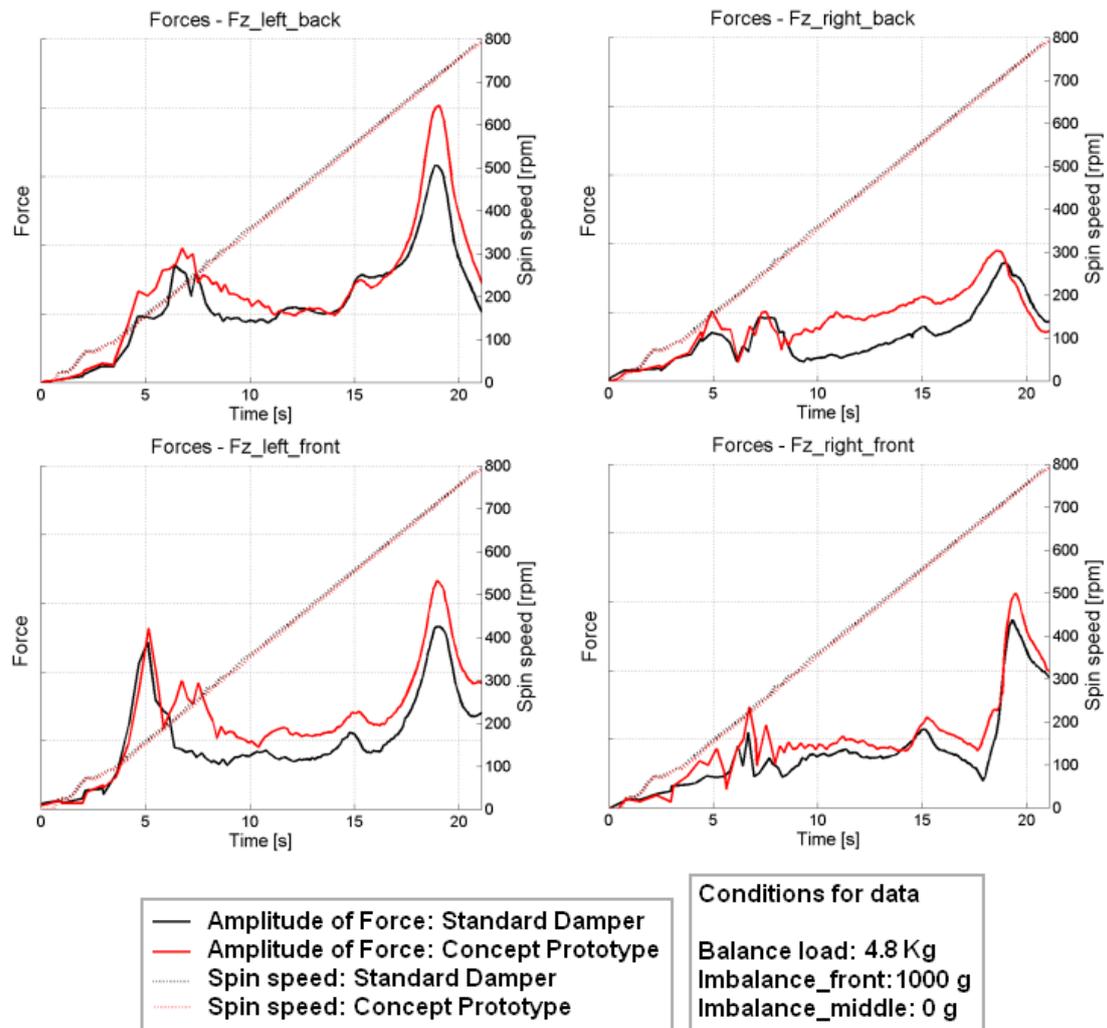


Figure 5.3.2.14 Amplitude of force with standard and prototype damper 3

As the imbalance load was modified in position and value, displacements were increased at low frequencies, close to the resonant frequency, as shows *Figure 5.3.2.15-17*. Said peak of displacement is lower than the admissible displacement regarding the washing machine design, thus, the drum did not hit any internal part during the performed tests.

The forces were increased with the 500g front imbalance load and decreased with the spread out 300g load as can be seen in *Figures 5.3.2.16-18* respectively. The best behavior of the prototype damper was obtained at this last configuration: 4.8 Kg and 300g spread out, wherein the forces were decreased at high frequencies about a 40% as shows *Figure 5.3.2.18*.

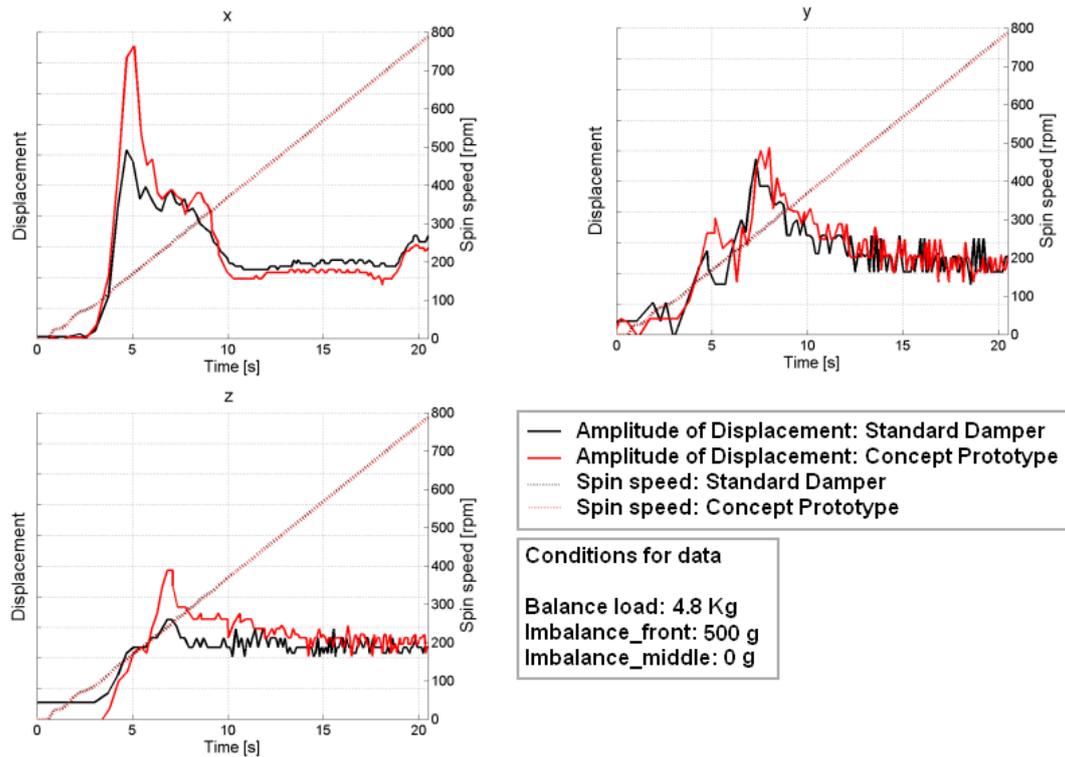


Figure 5.3.2.15 Amplitude of displacement with standard and prototype damper 3

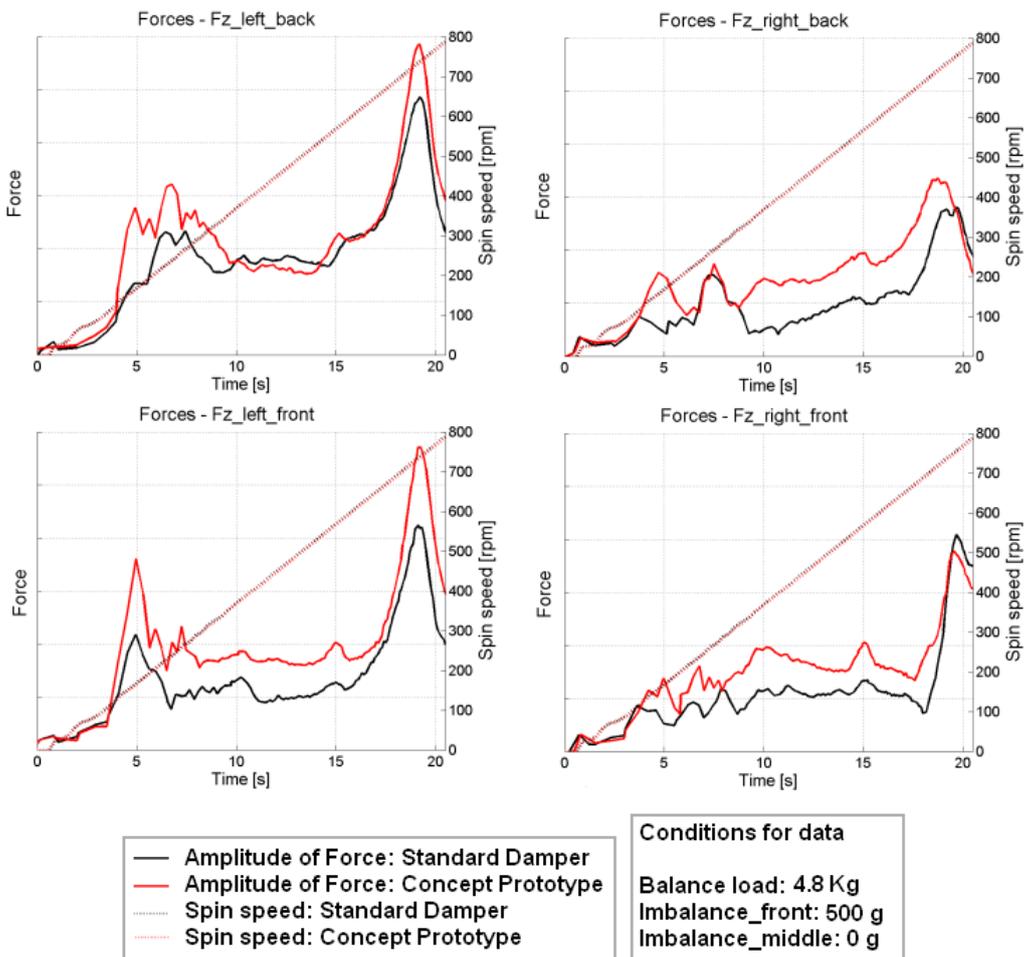


Figure 5.3.2.16 Amplitude of force with standard and prototype damper 3

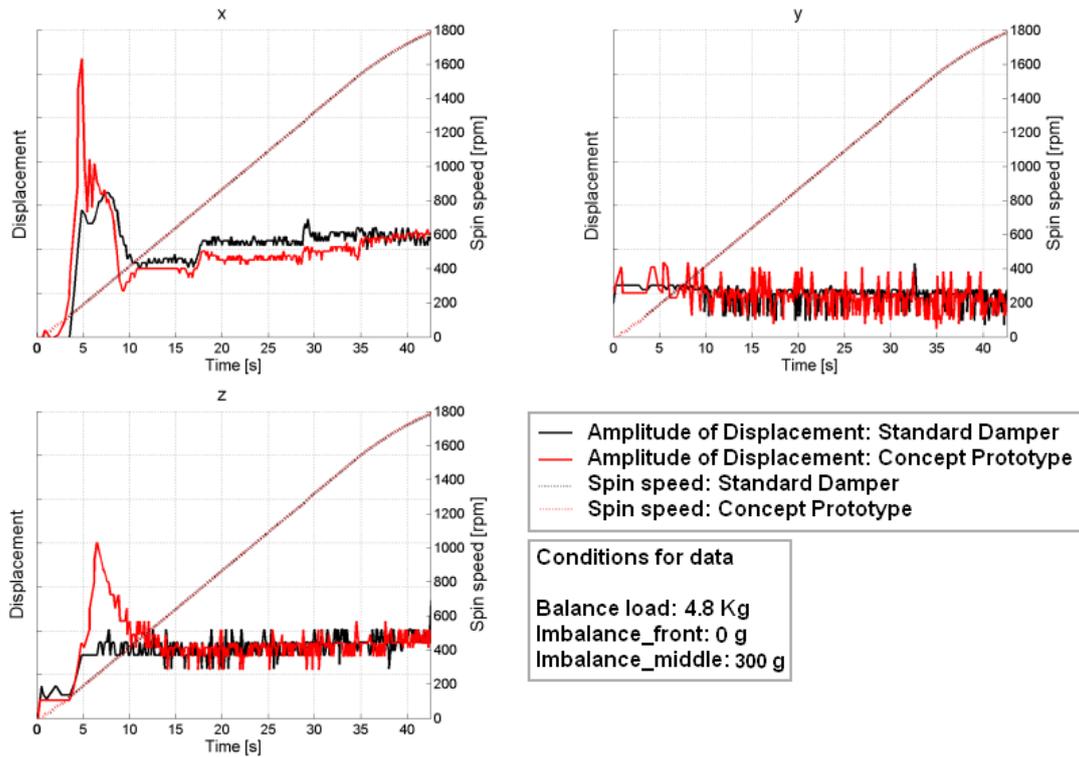


Figure 5.3.2.17 Amplitude of displacement with standard and prototype damper 3

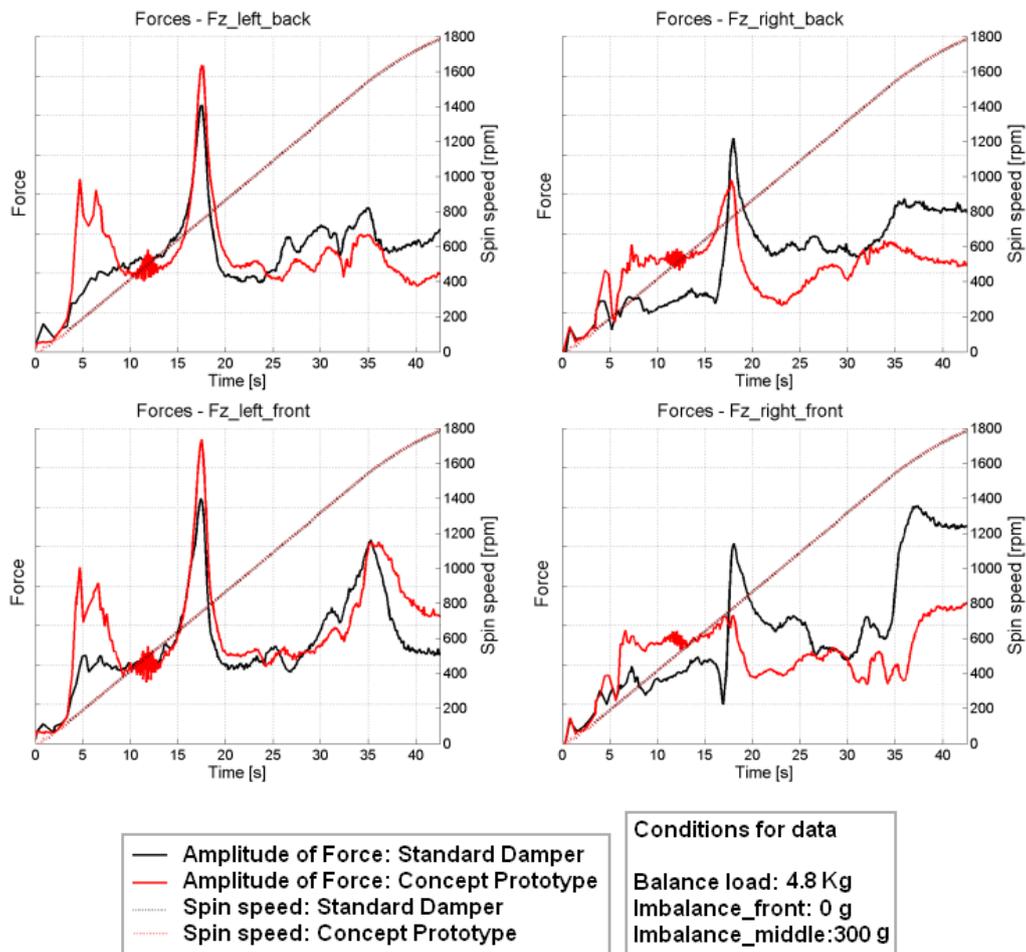


Figure 5.3.2.18 Amplitude of force with standard and prototype damper 3

5.3.3 Conclusions

Several tests were done mounting the standard damper and the prototype damper in the washing machine test rig. In order to do a deep analysis, different amounts of balance and imbalance loads were mounted and tested at two spin speeds range (800 and 1800 rpm). Due to the possibility to adjust the inner parts of the prototype, three configurations were tested: 1, 2 and 3.

The main objective proposed with the new prototype damper was to use its capability of low/high damping behavior to reduce vibrations at all frequencies. It has been shown that as the inner parts were modified, from configuration 1 to 3, the amplitude of displacements was reduced. The results obtained from the new prototype damper show mostly the same behavior regarding displacement as the standard damper at all frequencies. As seen in figures, the highest displacements are located close to the resonant frequency. In spite of that, said high amplitudes of displacements obtained are lower than the admissible regarding the washing machine design.

The best results were obtained from the prototype damper configuration 3, with a reduction of about 40% of force at high frequencies in the case of 300g (spread out imbalance load).

6 Mathematical model

As a final task, a basic mathematical model was developed in order to obtain an expression which modeled the dynamic behavior of the new LCD. For that it was used a parameter optimization script in MATLAB (lsqnonlin) to fit the searched function to the data obtained in the test. The model used was the final prototype model 3.

The standard friction damper was studied previously in [28], and was obtained its mathematical model based on the basic friction behavior, being depending on displacement (x) and velocity (\dot{x}) as show equation (6.1)

$$f(x, \dot{x}) = k \cdot x + c \cdot \dot{x} \quad (6.1)$$

The non-linear behavior of the new LCD depends on displacement (x) and velocity (\dot{x}) too, but is not enough. The main important factor in the new LCD behavior is the internal geometry, which is responsible for the low and high damping effect. The second important factor is related to the internal parts configuration. So, two functions are needed to be included in the new equation: $f1(x)$ and $f2(x)$, representing of internal components behavior.

The friction force of the new damper should be dependent on these two parameters thus it is needed to add more terms in the equation (6.1) to build a complete model. In this case, it was worth grouping the velocity and the functions aforementioned as shows equation (6.2).

$$f(x, \dot{x}) = k \cdot x + \frac{\alpha \cdot \text{sgn}(\dot{x})}{\beta - f1(x)} + \gamma \cdot \dot{x} \cdot f2(x) \quad (6.2)$$

Figure 6.1 shows the results obtained during optimization process. It can be seen that the proposed mathematical model does not fit the measured damper behavior at all frequencies. Only at a frequency of 20 Hz (1200 rpm of spin speed) the modeled curve fits in such a manner that it can be said the mathematical model is valid. In *Figure 6.2* one can see the comparison between the measured and modeled curve at said frequency.

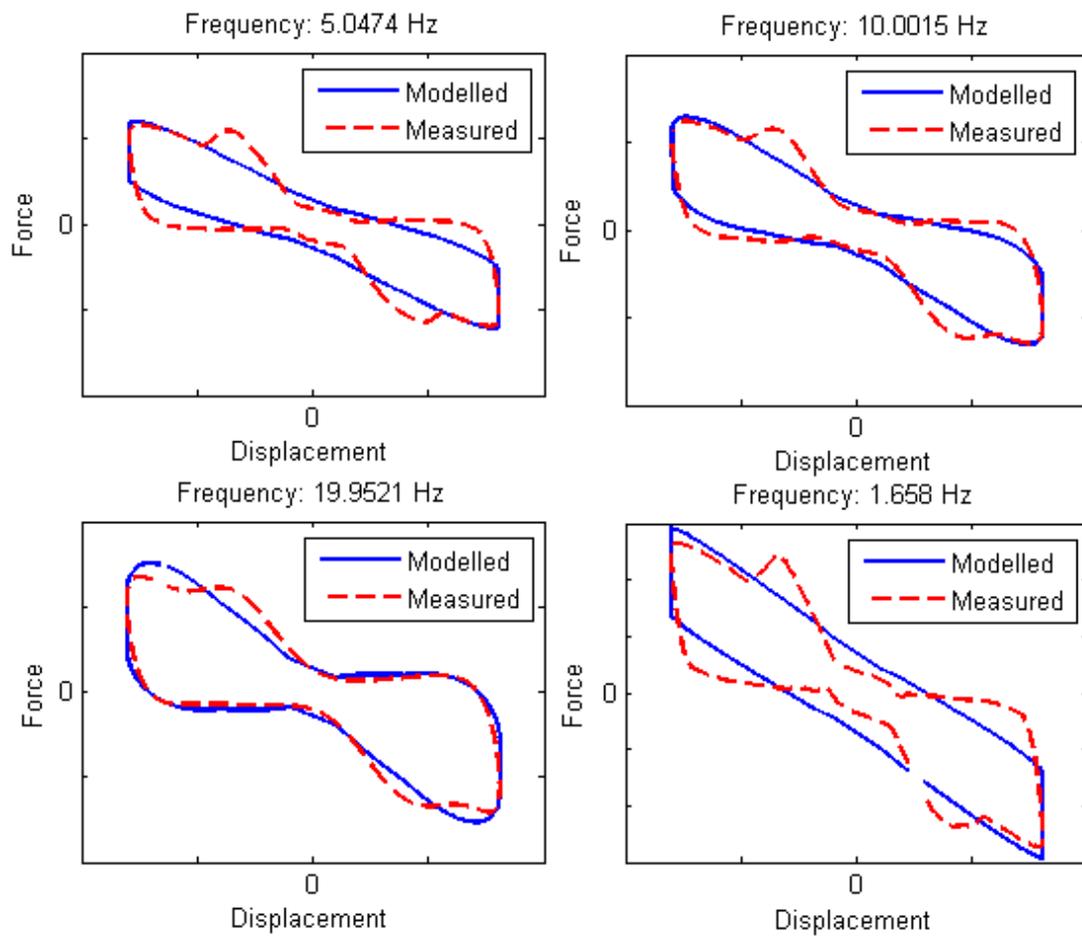


Figure 6.1 Optimization results

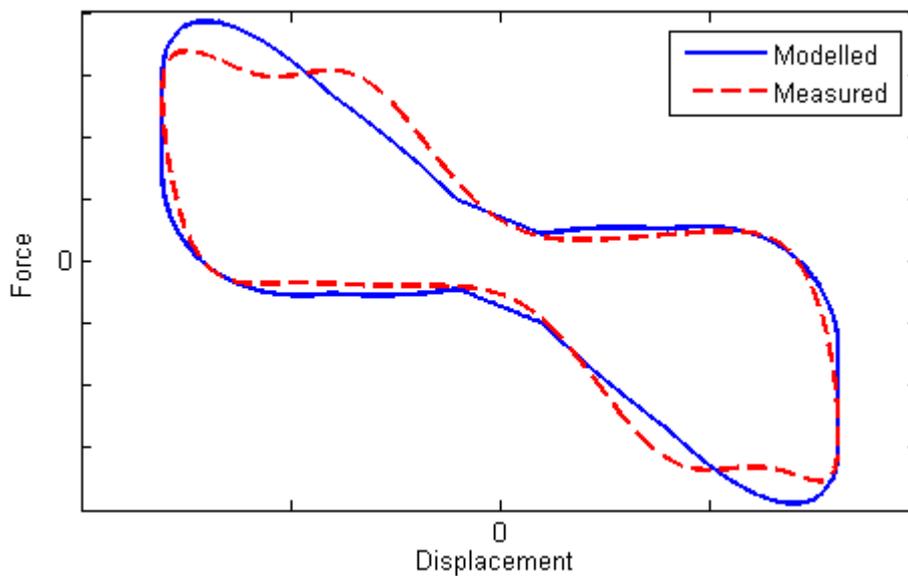


Figure 6.2 Comparison damping behavior: measured and modeled

The values of the parameters which fit better the modeled curve to the original measured are shown in the following table.

Table 6.1 Optimized values of the parameters

| k | α | β | γ |
|---------|----------|---------|------------|
| -5.3338 | 479.9996 | -5.0053 | -0.0094102 |

The results obtained from the optimization scrip were:

Number of iterations: 50

Number of function evaluations: 401

Sum of squared residuals at solution: $9.7731e+06$

Execution time: 12.875 s

Where:

$$|F^{sim}(\Delta, \xi) - F^{exp}(\Delta, t)|^2 \rightarrow min = 9.7731e^6 \quad (6.3)$$

$$|F^{sim}(\Delta, \xi) - F^{exp}(\Delta, t)|_{MAX}^2 = 6.9598e^3 \quad (6.4)$$

$$|F^{sim}(\Delta, \xi) - F^{exp}(\Delta, t)|_{MIN}^2 = 7.49e^{-7} \quad (6.5)$$

Where ξ represent the parameters k, α, β and γ .

Conclusions

A basic mathematical model was developed in the last stage of this work. Said model can be considered as an initial part of a better and more complicated mathematical development which fits the behavior of the novel low-cost damper at a wider frequencies range.

7 Conclusions and recommendations

A novel Low-Cost Damper (LCD) has been developed, manufactured and tested during this master thesis for its application in washing machines suspension system. From the beginning to the end during twenty weeks it has been focused passing through several stages and analysis until the final solution proposed. Initially a benchmarking of all devices followed by a complete Quality Function Deployment (QFD) analysis were done in order to focus to the best configuration and technology applicable to develop this new damper. Friction damping joined to a low-high mechanically action was the way forward. In addition, several sorts of available dampers were studied, obtaining an enriching knowledge. Once selected the best proposed concept, a deep study was done in order to adjust its behavior to the washing machine suspension system requirements.

The final prototype developed was manufactured and several tests were performed using the test rig for damper classification. The tests were satisfactory in order to figure out the prototype behavior at different conditions and using four models by varying the internal parts. It has been also possible to demonstrate the “butterfly behavior” predicted in the first design stage and done a deep analysis to know how affect in said behavior the entire concept.

As an intermediate result, the LCD model 3 was the best configuration to make the prototype capable to be introduced into washing machines suspension system. Once selected the final prototype, three more were manufactured and tested in the washing machine test rig, being mounted as a four strut configuration and compared with the standard friction dampers. Said tests showed the applicability of the new LCD as new damper in washing machines suspension system, sighting in this case certain features that can make it promising. In general the new LCD offers a good response at high frequencies with a reduction of the forces, about 40% for a specific load condition. At low frequencies appear the not expected results according to the amount of displacement comparing with the standard damper configuration. As the internal parts of the prototype were modified from configuration 1 to 3, said behavior was improved. In spite of that, said high amplitudes of displacements obtained were lower than the admissible regarding the washing machine design.

The fact that the LCD prototype was developed using parts from the standard friction damper is clearly influential in the final results, due to the possibility of adaptation to the new conditions, such as low-high damping transition, amount of damping and connection between parts. All the internal parts can be redesigned to be able to fit better together and thus get closer to the desired behavior, avoiding undesirable dynamic effects. This is referred to the extra force peaks shown in the new damper behavior.

A simple mathematical model of the prototype was built and its parameters were adapted as far as possible using optimization algorithms. Obviously it is needed to take into account that a deeper study would offer a better mathematical model.

As final recommendation, an optimization of the washing machine suspension system with the proposed dampers can be done as future work. Thus, an optimization of the position of the damper fixed points can be done in order to improve the washing machine suspension systems behavior.

8 References

- [1] M. Fenechea, *Consideration about suspension systems*, Fascicle of management and technological engineering, Vol.VI, 2007, p.420-422.
- [2] Halcyonics, the anti-vibration specialists. Online January 2009,
http://www.halcyonics.de/en/technology/popup/popup_technology_Active_vibration_isolation.php
- [3] J.D. Carlson, *Controlling vibration with magnetorheological fluid damping*, 2002.
- [4] Continental air damping systems. Online January 2009,
http://www.conti-online.com/generator/www/com/en/continental/portal/themes/press_services/press_releases/fairs_events/pr_2006_10_10_intermot/gt_pr_intermot_cads_en.html
- [5] X. Song, *Design of adaptive vibration control systems with application to magnetorheological dampers*, ETD-111299-170915, 1999.
- [6] Deicon, Dynamics & Control, *Vibration Isolation Mounting*
- [7] G. Obstfelder, *Hydraulic vibration-damper with variable damping characteristic*, WO, 1989, 012183.
- [8] S. Yoshikawa, A. Bogue and B. Degon, *Commercial application of passive and active piezoelectric vibration control*, *Applications of Ferroelectrics* IEEE, 1998, p.293-294.
- [9] N. Abdullah, M. Jailani and R. Zakia, *Compromising vehicle handling and passenger ride comfort using ER-damper*, *Mechanika*, 2005, Vol.4, p.50-55.
- [10] N.Harris, Henri P, *Electrorheological dampers and semi-active structural control*, IEEE, 1995, P.3528-3533.
- [11] L.Bitman, Y.T. Choi, S.B. Choi and N.M. Werely, *Electrorheological damper analysis using an Eyring-plastic model*, *Smart Mater Struct*, 2004, Vol.14, p.237-246.
- [12] J.D. Carlson, *Low-cost MR fluid sponge devices*, *Journal of intelligent material systems and structures*, Vol.10, 1999, p.598-594.
- [13] R. Ferlicca, *Friction damper for washing machines or the like*, Milan (IT), Jul. 24, 2001. US 6,264,014 B1.
- [14] T. Peuker, A. Pelezer and M. Weder, *Damper*, Suspa Holding GmbH, Altdorf (DE). Nov. 4, 2008. US 7,445,098 B2.
- [15] J.W. Chang, J.W. Kim, Y.H. Klim, D.W. Kim and K.C. Woo, *Damper for washing machine*, (KR). Nov. 25, 2004. US 231,374 A1
- [16] M. Ehrnsberger and M. Rössner, *Frictional Damper for washing machines with spinning cycle*, Suspa Compart Aktiengesellschaft. Altdorf, (DE). Oct. 5, 1999. US 5,964,105.
- [17] Carlson,J.,David, *On-off damper system, especially for washing machine*, (US). Oct. 13, 2005. WO 95,820 A1.

- [18] H.J. Bauer, H-P Bauer, D. Mayer and L. Stadelman, *Frictional vibration damper*, Suspa Compart Aktiengesellschaft, (DE). Jan. 14, 1992. US 5,080,204.
- [19] G-R Park and C-S Jun, *Combination type damping and washing machine*, LG Electronics Inc., Seoul (KR). Oct. 18, 2005. US 6,955,248 B2.
- [20] J.D. Carlson, *Magnetically actuated motion control device*, Lord Corporation. Apr. 30, 2002. US 6,378,671 B1.
- [21] M. Ehrnsberger, D. Mayer and H. Siegner, *Frictional damper, in particular for spinner-type washing machines*, Suspa Compart Aktiengesellschaft, Altdorf, (DE). Aug. 27, 1996. US 5,549,182.
- [22] Y-H Kim, J-W Kim, D-W Kim, S-C Park, S-M Jeon and G-R Park, *Damper of drum type washing machine*. LG Electronics Inc., Seoul (KR). Apr. 19, 2007. US 7,204,104 B2.
- [23] S.K. Park and G.R. Park, *Damper in a washing machine and fabricating method*, LG Electronics Inc., Seoul (KR). Oct. 16, 2007. US 7,281,614 B2.
- [24] Y-H Kim, J-W Kim, and D-W Kim, *Damper and washing machines*, (KR). Aug. 5, 2004. US 148,980 A1.
- [25] W.T. Thomson and M.D. Dahleh, *Theory of Vibration with applications*, Prentice hall, Upper saddle River, NJ07458. University of California at Santa Barbara, pp. 67-70.
- [26] Thomas Nygård, *Modeling and Optimization of Washing Machine Vibration Dynamics During Spinning*, Chalmers University of Technology, 2009.
- [27] Thomas Nygård, J.Sandgren, V.Berbyuk, A.Bertilsson: *Vibration Control of Washing achine with Magnetorheological Damper*, Proceedings of the 8th International Conference on Motion and Vibration Control (MOVIC 2006), August 27-30, 2006, KAIST, Daejeon, Korea
- [28] Thomas Nygård, V.Berbyuk: *Dynamics of Washing Machines: MBS Modeling and Experimental Validation*. Proceedings of the Multybody Dynamics 2009, ECCOMAS Thematic Conference, June 25-28, 2007, Politecnico di Milano, Milano, Italy.
- [29] Thomas Nygård, V.Berbyuk, A.Sahlén: *Modelling and Optimization of Washing Machine Vibration Dynamics*. Proceedings of the 9th International Conference on Motion and Vibration Control (MOVIC 2008), September 15-18, 2008, Technische Universität München.