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PROYECTO FIN DE CARRERA

Vibration Isolation with Skyhook Damping

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Vibration Isolation with Skyhook Damping

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Resumen

Este proyecto se centra en estudiar diferentes métodos de diseño de controladores y en crear un método de diseño adecuado para un sistema de aislamiento activo. La necesidad en un método distinto a los actuales se debe al aumento de la complejidad en el tuneado de los controladores con los métodos clásicos de diseño cuando se tienen dos o más lazos cerrados y a que se está tratando de añadir un lazo adicional a los sistemas de aislamiento mediante acelerómetros. Se hace uso del método H-infinito debido a la posibilidad de este de obtener controladores óptimos sin ser afectado por la complejidad de la planta. El problema de los métodos H-infinito es que el tuneado es difícil de comprender y dominar. Durante el estudio de distintos métodos se ha obtenido una versión de un método H-infinito que es capaz de obtener controladores con un proceso de tuneado que es más fácil de comprender. Este método hace uso de la respuesta en frecuencia deseada para el sistema para obtener el controlador óptimo.

Se ha visto además que aunque en la simulación los resultados en aislamiento obtenido por los controladores sean aceptables una vez aplicado a una máquina física estos no son aceptables. Esto se debe a que es necesario estudiar la máquina física para añadir al tuneado del controlador información sobre incertidumbres para garantizar robustez en la estabilidad o en el rendimiento del controlador.

Abstract

This project is centred to study different control design methods and create a suitable design method for an active isolation system. The study is pushed by the increased complexity in tuning controllers with classical methods for controllers with two or more loops. The H-infinity method is chosen due to the possibility to obtain optimal controllers without being affected by the complexity of the plant. The problem of H-infinity methods is that its tuning process is difficult to understand and master. A version of an H-infinity method is obtained that is able to obtain suitable controllers with a tuning process that has the intuitiveness of classical methods. This method makes use of the desired output of the controlled system to obtain the optimal controller to do so.

Two other methods are used to be compared, a classical loopshaping method and a straightforward approach to the H-infinity methods. The controllers obtained through all three methods are put to the test in a physical machine. The results are unsuccessful. This result in comparison with the simulated tests is due to the small stability margins obtained through the H-infinity method. The conclusion to this work is that in simulated environments the new method is viable while for physical environments there is a need to add uncertainty data to the tuning process.

1 Introducción

1.1 Motivación

Hoy en día se invierte mucho esfuerzo en mejorar los procesos de fabricación de semiconductores. El motivo es la demanda del mercado en obtener chips de mayor densidad que sean más rápidos, baratos y eficientes. La velocidad a la que se producen estas mejoras está descrita por la ley de Moore. Pero a diferencia de las leyes físicas la ley de Moore no es estricta y es más bien una guía que estima el futuro avance de la tecnología de semiconductores.

El proceso de fabricación se divide en cuatro etapas principales: Deposición, Litografía, Ataque químico y alteración de las propiedades eléctricas. En la etapa de litografía es donde el diseño del chip a fabricar se imprime sobre las obleas de silicio. Hoy en día esta etapa supone el cuello de botella para la máxima densidad posible en el chip.

ASML produce máquinas de fotolitografía para la industria de semiconductores. Estas máquinas fotolitográficas están diseñadas para ser capaces de exponer las obleas para obtener conexiones que se encuentran en órdenes de magnitud entorno a los 20 nanómetros en anchura. A estas escalas las máquinas son extremadamente sensibles a las vibraciones. Con el fin de aislar lo más posible de vibraciones y así poder aumentar al máximo el rendimiento de las máquinas se ha creado un sistema de aislamiento activo de vibraciones.

Hasta hace poco el control del aislamiento activo se hacía a través de un solo sensor. Un sensor de posición que ofrecía datos sobre la posición relativa al suelo. Obtener datos de posición relativos limita el máximo grado de aislamiento que se puede conseguir. Por tanto para superar estas limitaciones es necesario incorporar medidas de posición absolutas. El aislamiento mediante medidas absolutas es conocido como “Skyhook Damping”. En el mundo físico no es posible obtener la posición de un objeto sin un punto de referencia pero el movimiento del objeto (velocidad y aceleración) sí es una medida absoluta y está relacionada con la posición.

El mercado de los acelerómetros ha llegado a un punto donde es factible hacer uso de estos sensores para detectar las mínimas aceleraciones que se buscan controlar en las máquinas litográficas.

Sin embargo añadir un segundo tipo de medidas a un sistema de control aumenta la dificultad en la creación de un controlador capaz de optimizar el aislamiento mediante los métodos clásicos de control. Por tanto es necesario investigar la posibilidad de obtener un controlador para sistemas de control más complejos. La opción que se probará en este proyecto es el de los métodos H-infinito.

1.2 Objetivos

El objetivo principal del proyecto será el de obtener un método para crear un controlador para un sistema con dos lazos cerrados, uno por un sensor de posición relativo y otro por un acelerómetro mediante métodos de H-infinito.

El problema con los métodos H-infinito es que el proceso de tuneado consume tiempo y es muy poco intuitivo. Por lo tanto el objetivo de este proyecto es el de crear un método de diseño del controlador mediante H-infinito que haga el tuneado más sencillo y comparar sus resultados con otros métodos más comunes. Los pasos a seguir serán los siguientes:

- Diseño de un controlador con el método loopshaping clásico.
- Diseño de un controlador con un método H-infinito sencillo muy utilizado en la literatura.
- Diseño de un controlador usando un método H-infinito modificado para ser más intuitivo.
- Prueba de los tres controladores en un entorno real para demostrar sus capacidades y comparar los resultados de los controladores obtenidos a través de los diferentes métodos.

1.3 Contenidos de los apartados

El **Capítulo 2** proporcionará conceptos que se utilizarán como base para el proyecto. También mostrará el estado del arte en los sistemas de aislamiento activo y los métodos de diseño de control para sistemas de aislamiento.

El **Capítulo 3** mostrará el proceso de tuneado de los tres métodos de diseño de control utilizados. También se muestran los resultados de los controladores obtenidos y una comparación entre ellos.

El **Capítulo 4** mostrará el sistema de test creado y los resultados obtenidos.

El **Capítulo 5** son las conclusiones finales del proyecto.

1 Introduction

1.1 Motivation

There is a lot of research effort being put onto the improvements in manufacturing processes of semiconductor devices nowadays. The reason is the demand of the market to be able to produce higher density chips which are faster, more efficient and cheaper. The rate at which these improvements occur is known as the Moore's law. But unlike scientific laws Moore's is not a strict one and there is a lot of research effort needed to improve the manufacturing process of chips in order to be able to keep obeying this law.

The manufacturing process can be divided into four main stages: deposition, patterning, removal and modification of electrical properties. It is in the patterning stage where the layout of the manufactured chip is printed onto the wafers. Nowadays this stage is a bottleneck for the maximum density of the chips.

ASML creates photolithographic machines which will print the desired pattern onto the wafers that contain the chips. These photolithographic machines are designed to be able to expose the wafers to create lines that are in the orders of the 20 nanometers in width. These machines are therefore extremely sensitive to vibrations. In order to prevent these vibrations from affecting the machines performance an isolation system is created.

Until recently the active method for vibration isolation that has been used in ASML was based on position sensors. These position sensors give readings of the distance between the ground and the isolated surface which are known as referential readings. With referential readings there is a limitation that affects the damping the isolation system can create. To overcome these limitations another sensor has been added to the isolation system, an accelerometer. Accelerometers give absolute readings. With absolute readings a different type of damping can be created that is called Skyhook damping.

1.2 Objectives

Designing controllers to act in accordance with the readings of several sensors with the classical loopshaping methods is challenging. As the number of control loops increases the number of variables to take into consideration also increases making achieving a good performance harder to obtain. To be able to overcome this problem a different controller design method has to be used. This method will be the H-infinity loopshaping method that is capable of achieving optimal controllers for any number of parallel control loops.

The problem with the H-infinity methods is that the tuning process is time consuming and very unintuitive. Therefore the focus of this project will be that of creating a controller design method using H-infinity that will make the tuning process easier. The steps will be as follow:

- Designing a controller with the classical loopshaping method
- Designing a controller with a straightforward H-infinity method as seen in several research papers
- Designing a controller using a modified H-infinity method with a simpler tuning process.

- Testing all three controllers in a real environment to prove their capacities and compare the results of the controllers obtained through the different methods.

1.3 Chapter contents

Chapter 2 will provide concepts that will be used as the background for the project. It will also show state of the art on active isolation and isolation control design methods.

Chapter 3 will show the tuning process of the three different control design methods. It will also show the results of the obtained controllers and a comparison between the methods

Chapter 4 will show the test setup created and the results obtained with it.

Chapter 5 contains the final conclusions of the project

2 State of the art

2.1 Lithographic machines

The lithographic machine is made to expose the wafers to a light pattern that will define the printed lines in the wafer's dies. The dies are the individual pieces the wafer will be cut into to make each chip. Due to the extreme small sizes of the lines inside the dies the process has to be extremely precise. There are several parts involved in this process; a simplistic representation of the machine is shown in Figure 2-1.

- Light source: Generates the light necessary for the exposure in a controlled frequency and gain.
- Reticle: Contains the pattern that is going to be transferred to the wafers. The size of the reticles is x4 bigger than the pattern that will be printed onto the wafer.
- Lens: Reduces the size of the patterned beam of light and focuses the image on top of the wafer.
- Wafer stage: Positions the wafers under the lens.

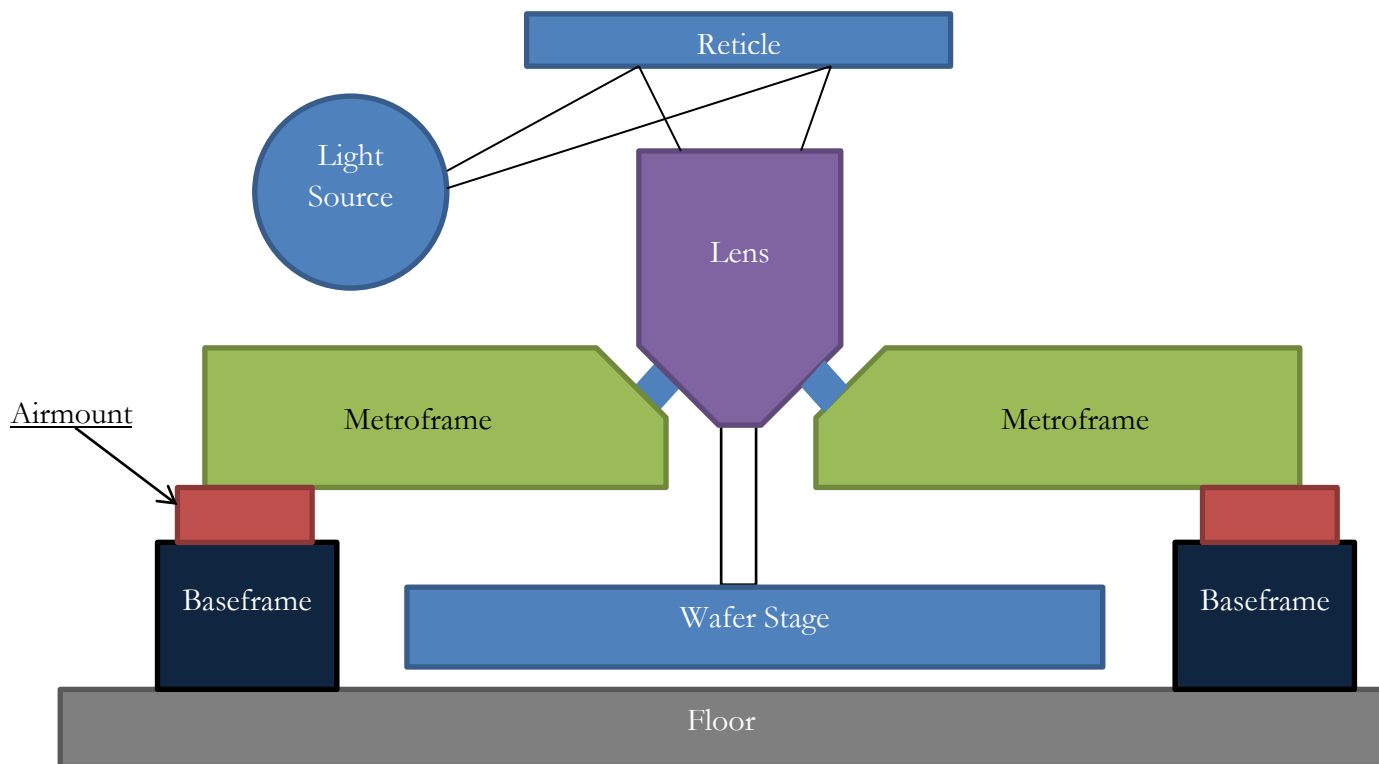


Figure 2-1 - ASML Lithographic machine schema

There are a large number of factors that can be worked on in order to improve the precision of the machine. One of those factors is the isolation of vibration which is the subject of this project. The vibration of the different parts of the machine affects the quality of the image that will be printed to the wafer in different ways: Image fading, position mismatch, among others. To reduce the effect of the vibrations, sensitive parts of the machine are set into a frame that will be isolated from the ground. This frame is called the metroframe and the

isolation of this metroframe will be achieved through the airmounts that are the support on which the frame lies.

2.2 Metroframe

The metroframe is an isolated mass that gives support to a number of components within the machine. Using a frame to support all the components makes it easier to handle as all the vibration control power is focused on keeping just one big mass free of motion.

The metroframe weighs about 2000Kg. It is supported by three mounts (airmounts) which act as a system of springs and dampers. They also contain active Lorentz actuators to allow for active control of the movement of the frame.

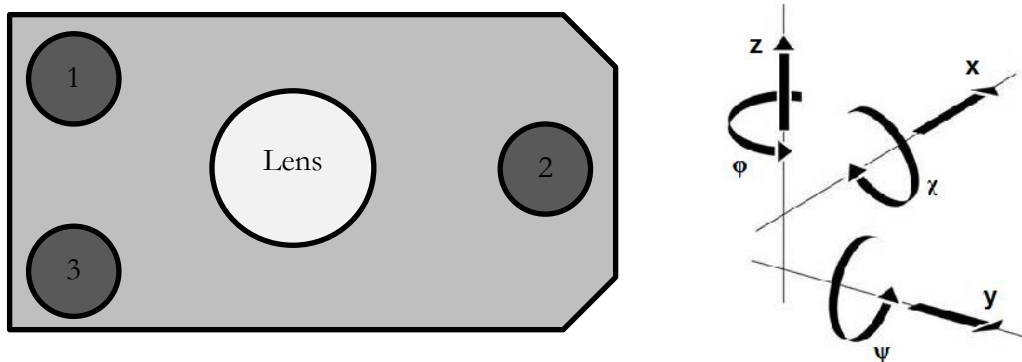


Figure 2-2 - Airmounts position and coordinate system of the metroframe

The location of these airmounts within the metroframe is shown in Figure 2-2. With this configuration the metroframe is capable of moving in six different degrees, three position movements (x, y, z) and three rotational movements (χ, ψ, ϕ). This enables the metroframe to be able to isolate from disturbances in all possible directions.

2.3 Isolation of the metroframe

The different types of disturbances the metroframe has to be isolated from can be divided into two types, disturbances from the ground and disturbances affecting the metroframe directly. To describe how these disturbances affect the metroframe at different frequencies a frequency response diagram will be used. Due to the importance of these frequency responses they will be called transmissibility and compliance. In order to understand what kind of information can be obtained from these plots a simple 1 DOF model as shown in Figure 2-4 will be used to explain them.

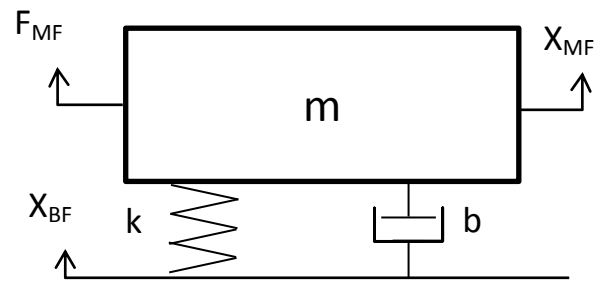


Figure 2-4 - Mass-Spring-Damper schematic representation

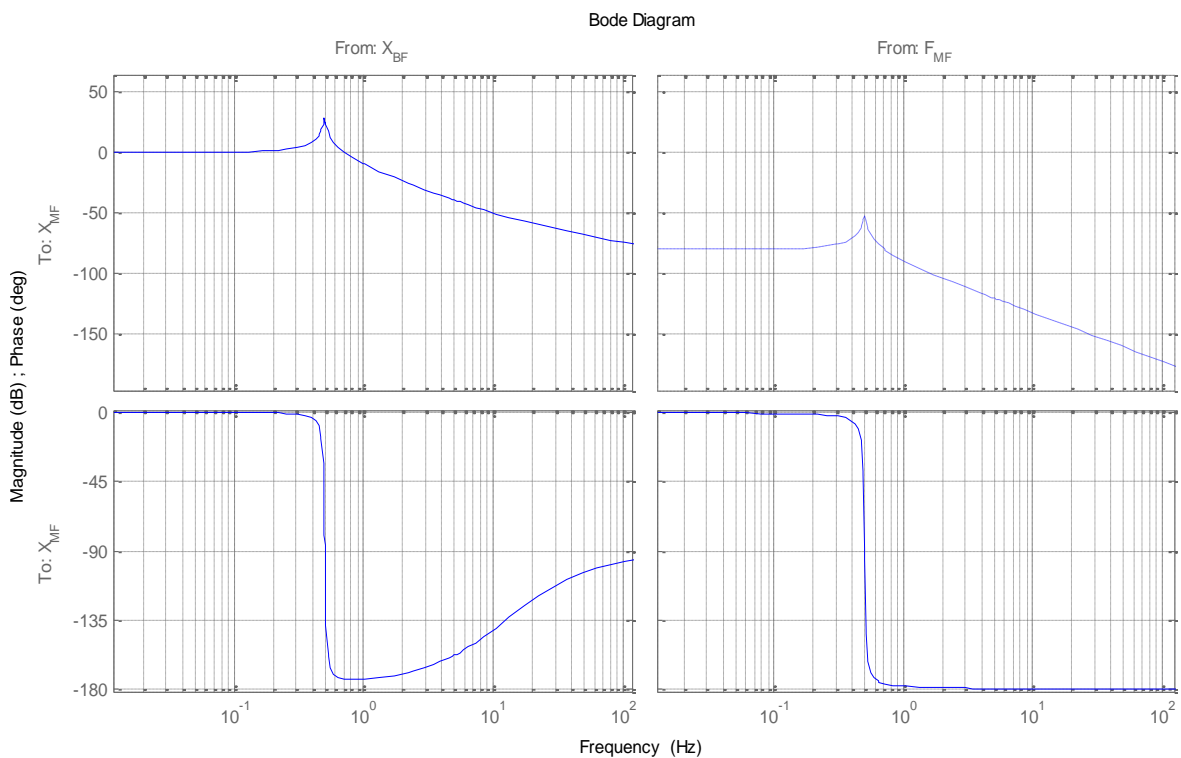


Figure 2-3 - Transmissibility and compliance of a mass-spring damper model

Transmissibility is the transfer function from the position of the base to the position of the metroframe. If the base movement is ' X_{BF} ' and the position of the metroframe is ' X_{MF} ' transmissibility would be described as $H_{tx} = \frac{X_{MF}}{X_{BF}} = \text{provide } s \text{ function}$. (see Figure 2-3) For low frequencies transmissibility starts at 0dB this means that for low frequencies the metroframe will move with the base maintaining the same relative distance between them. After the resonance there is a decoupling of the mass from the ground movement that can be appreciated in the slope. This decrease in magnitude means that the high frequencies of the vibrations are not being fully transmitted to the metroframe. Due to the existence of a damper between the ground and the metroframe the magnitude of the resonance peak is reduced. However it is also a source of vibration transmission to the metroframe, this can be seen in

the reduction of the slope's decrease from being a -2 (-40dB/decade) slope to being a -1 (-20dB/decade) slope for higher frequencies.

Compliance is transfer function from the Force acting on the metroframe to the position of the metroframe. If the force is 'F_{MF}' and the position of the metroframe is 'X_{MF}' compliance would be described as $H_{cm} = \frac{X_{MF}}{F_{MF}} = \text{provide } s \text{ function}$. The form of the compliance's frequency response has a resemblance to the transmissibility however the meaning of results is slightly different (see Figure 2-3). The flat line for low frequencies describes the resistance to force that comes from the springs, thus this line is equal to 1/k (inverse of the spring constant). The slope after the resonance peak is known as the mass line. This is because this line represents the slope (1/m*s²) which comes from the formula F = m*a. This means that for higher frequencies the displacement of the mass is minimized because of the resistance of the mass to change position quickly.

What is expected to be seen in these transfer functions when the isolation of a mass is improved complies with the following criteria, also schematically shown in Figure 2-5 and Figure 2-6

1. Damping of the resonant peak
2. Shifting of the resonant peak to lower frequencies.
3. Faster decoupling of the mass from ground vibrations, equivalent to decreasing the transmissibility's magnitude after the resonant peak.
4. For compliance, reductions in magnitude of the transfer function.

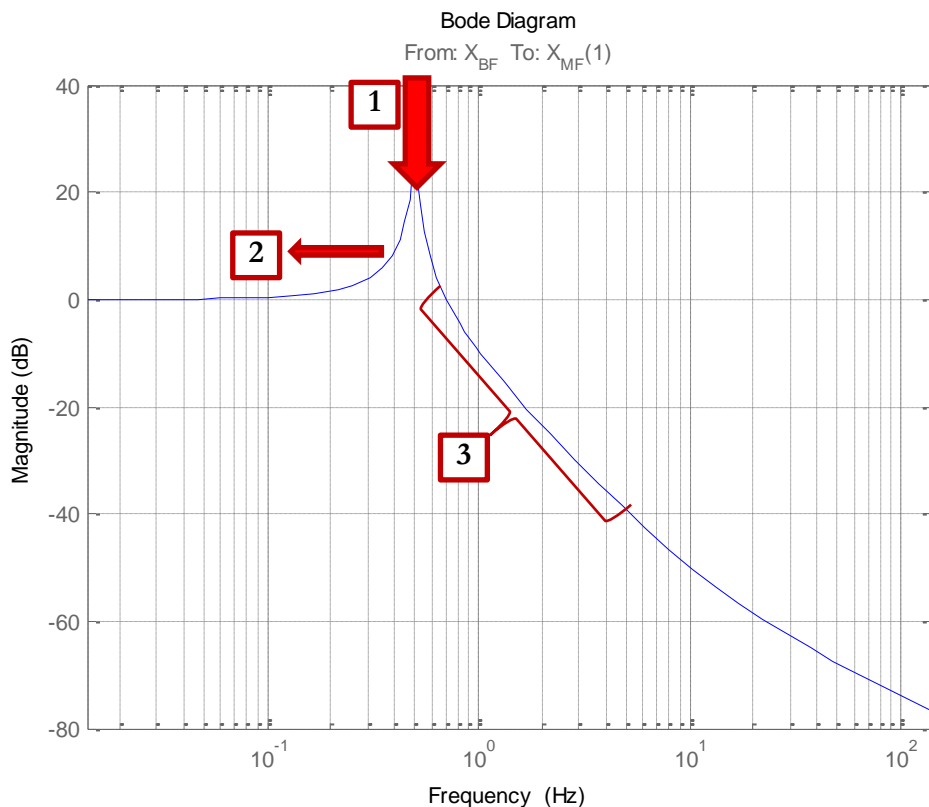


Figure 2-5 - Isolation improvements on transmissibility

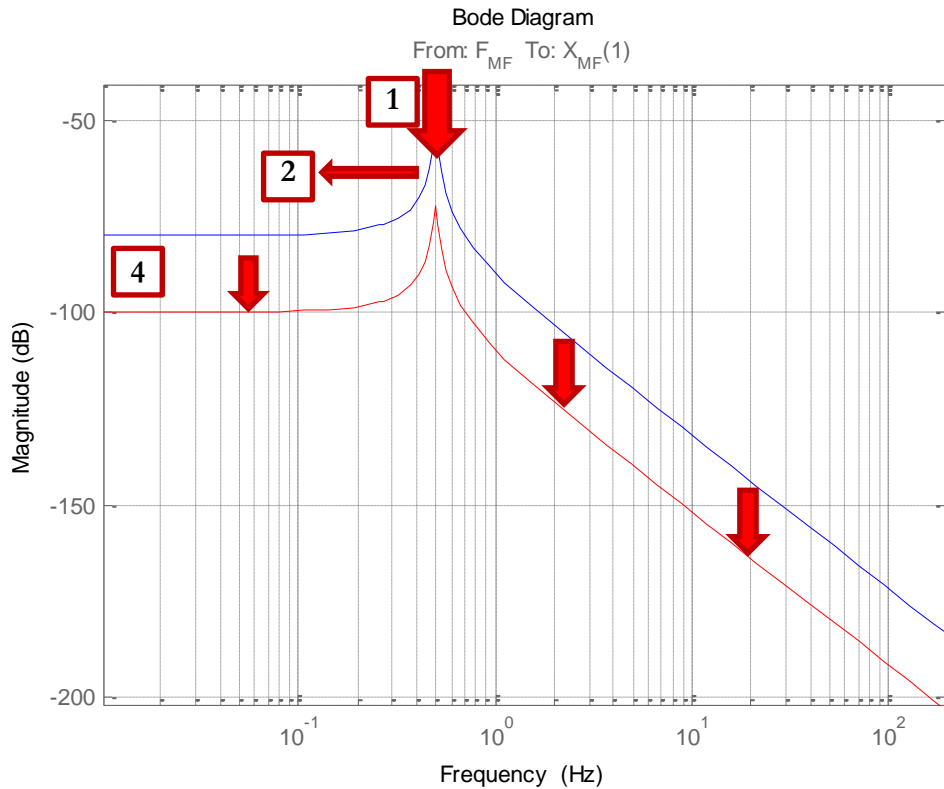


Figure 2-6 - Isolation improvements on compliance

2.3.1 Passive isolation

Passive isolation is the isolation that can be obtained by the use of different configurations of springs and dampers. Therefore the tuning parameters are limited to only the stiffness coefficient ('k') of the spring and the damping coefficient ('d') of the damper. Details on what can be obtained by changing those parameters are shown below. This is done to show the limitations of passive isolation.

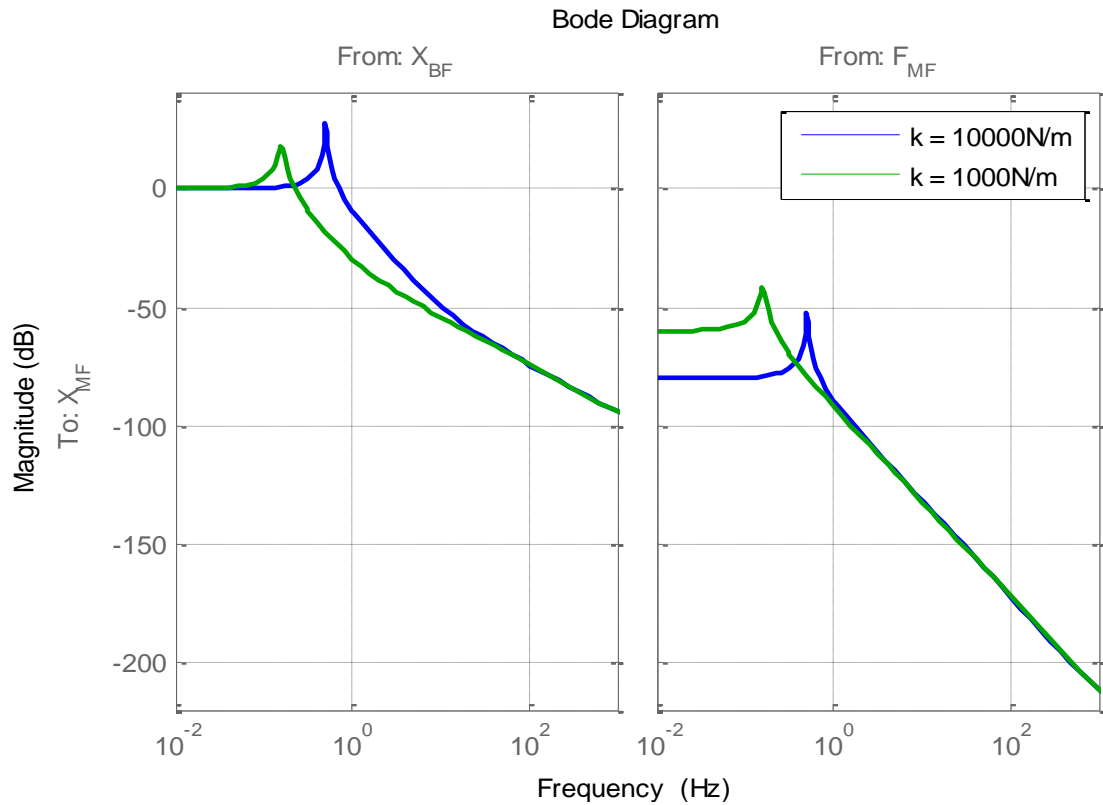


Figure 2-7 - Effect of different stiffness's of the springs in the transmissibility and compliance.

Changing the stiffness of the spring ('k') in Figure 2-4 will affect transmissibility and compliance. For both transfer functions, reducing the stiffness will reduce the frequency at which the mass will start decoupling from the ground. For compliance the constant line $1/k$ will reduce proportionally to the stiffness. The effects of changing the stiffness can be seen in Figure 2-7.

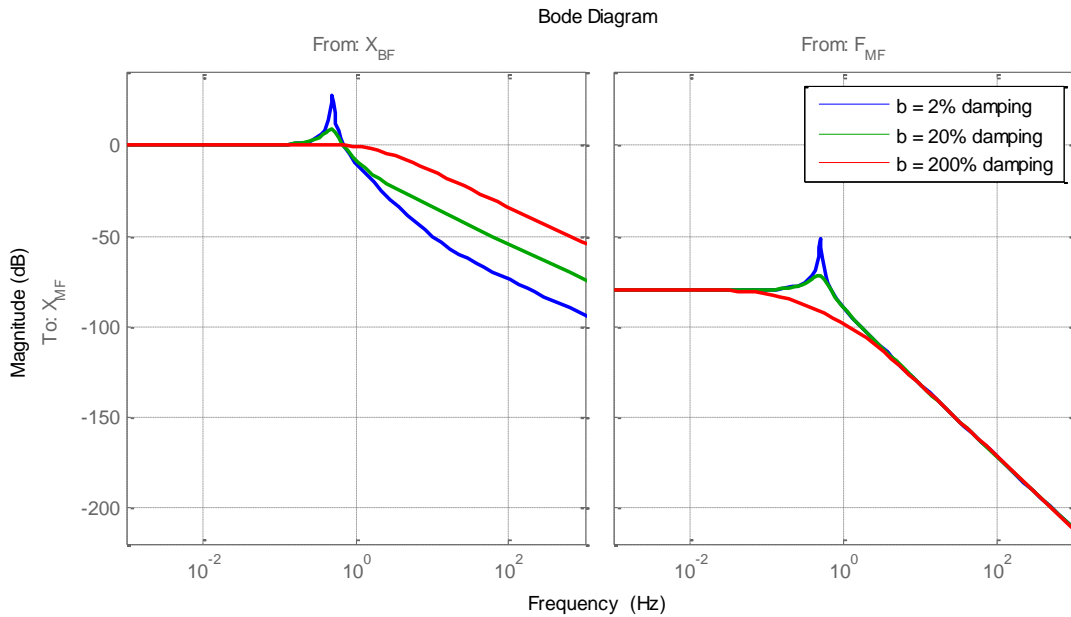


Figure 2-8 - Effect of different damping's to transmissibility and compliance.

Changing the damping coefficient ('b') will reduce the peak in gain of the resonant frequency for both transmissibility and compliance. In the case of transmissibility it also affects the frequency at which the decoupling slope changes from being a -2 slope to a -1 slope influencing transmissibility at high frequencies. The effects can be seen in **Error! Reference source not found.**

Due to the compromises that tuning the springs and dampers imply, passive isolation has limited capabilities to improve on transmissibility and compliance.

2.3.2 Active isolation

Active isolation stands for the techniques that make use of control systems to increase the amount of achievable isolation in a system. The idea behind this technique is to obtain the data on how the system is vibrating and counteract the measured movement in order to increase the vibration isolation of a mass.

The two existing settings for the control loop are feedback and feedforward. As seen in literature the most common setting for vibration control is with a feedback loop as shown in Figure 2-9. Feedforward in vibration control is offered as a complement of the feedback control. Feedforward can help to improve the performance of a system but is limited by the

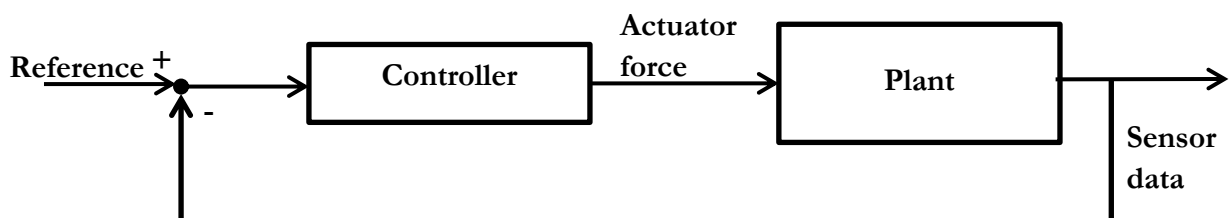


Figure 2-9 - Typical feedback control configuration for plant and controller

precision of the modeling of the plant. (Yasuda, Osaka, & Ikeda, 1996), (Nelson)

The main concerns for active vibration control systems are to maintain the stability and the performance requirements of the system. Being able to do this under uncertain plants (plants that have imprecise variables or unmodeled parts) is called Robust stability and Robust performance respectively.

The first methods on vibration control made use of graphical representations to see the effects of the designed controllers and then use the information to refine its performance. Usually these graphs were shown in the frequency spectrum and made use of this information to manipulate the controllers frequency response by adding different types of filters to the loop. These methods were therefore called as loopshaping methods. (F.Franklin, Powell, & Emami-Naeini), (C.Doyle, A.Francis, & R.Tannenbaum). These loopshaping methods were focused on helping the designer to achieve stability while the performance was left to the designer's ability. Robust stability was not ensured but obtained by adding margins to the stability of the system. A major limitation of these methods is that they are not applicable to MIMO systems.

As processes to be controlled grew in complexity the industry started to look for methods that would help to create controllers that could obtain optimality in these multivariable environments (Skogestad & Postelwaith). One of these methods is known as the H-infinity synthesis method. This method is able to obtain optimal controllers through the application of mathematical algorithms. The H-infinity method will be explained later on in 3.3.

The H-infinity method is not a strict method and different ways of making use of these algorithms have been tried out. For instance, in (Ding C. , Damen, Bosch, & Janssen) a way of combining H-infinity methods in combination with classical controllers is studied. The intention is to create a stable system with the classic controller and add a second controller that would take care of the non-linearity of the system.

There has also been research going on searching for the best configuration of actuators and sensors to obtain the best performance. In (Wal & Heertjes) different configurations of sensors were studied it is shown that the combination of relative position sensors and absolute acceleration gave the best results.

2.4 Plant uncertainties and robustness

The plant models created for the simulations are generally not perfect in their representation of the real plant and the frequency responses differ more or less from one to another. All these divergences are known as uncertainties. These uncertainties can come from very diverse causes: unmodeled modes in the plant, resolution limitations of the sensors, non linearities, etc... To make account of all these different uncertainties they can all be studied as dynamic uncertainties.

Among the dynamic uncertainties these can be divided into two different types, unstructured or structured uncertainties. Unstructured are the uncertainties whose origin is unknown or too disperse and therefore affect the entire system at once. Structured uncertainties are uncertainties that can be narrowed to only one transfer function within a system and therefore studied individually.

Uncertainties can also be described as to where they are applied within the loop: as input multiplicative uncertainties, output multiplicative, additive, etc... From now on additive uncertainty will be used and it is expressed as follows:

$$P_{\Delta} = P + \Delta$$

P_{Δ} symbolizes all the possible plants that the system tries to describe. P is the ideal plant and Δ is the uncertainty of the plant. The property of a controller to stabilize all the possible plants is called robustness.

3 Control design method's study

3.1 Introduction

Three different methods will be used to create a controller for the same plant. The intention is to be able to afterwards compare the designing process of each of these methods. The methods to be compared are the following:

1. Method 1 is the classical loopshaping method.
2. Method 2 is a simple approach to the H-infinity method
3. Method 3 intends to simplify the tuning process of the H-infinity method.

For any of the methods to be tried out, some initial specifications need to be set. These specifications are the model for the simulation environment, the control system structure and the performance goals to be achieved by the controller.

3.1.1 Simulation models

In order to be able to create a control system for the real plant it is necessary in the first place to create a suitable simulation model. The model does not need to be a perfect representation of the real life machine; it actually is beneficial to create a simplified model to make it is easier to study the effect of the controllers.

In this project a simple 1-DOF mass-spring-damper model is going to be used to study the different options of design to create the control system. The equation that describes the motion of the diagram showed in Figure 3-1 is shown below.

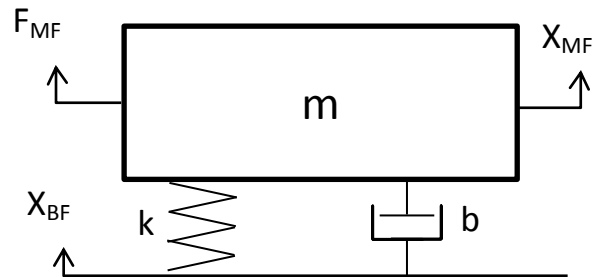


Figure 3-1 - Mass-Spring-Damper schematic representation

$$m\ddot{X}_{MF} = k(X_{BF} - X_{MF}) + b(\dot{X}_{BF} - \dot{X}_{MF}) + F_{MF}$$

It is necessary to implement the equations in state space form. Figure 3-3 shows the schematic block representation drawn from the equation above and from which it is easy to obtain the following state space matrices.

$$\mathbf{x} = \begin{pmatrix} \dot{X}_{MF} \\ X_{MF} \end{pmatrix}; \mathbf{u} = \begin{pmatrix} \dot{X}_{BF} \\ X_{BF} \\ F_{MF} \end{pmatrix}; \mathbf{A} = \begin{pmatrix} -\frac{b}{m} & -\frac{k}{m} \\ 1 & 0 \end{pmatrix}; \mathbf{B} = \begin{pmatrix} \frac{b}{m} & \frac{k}{m} & \frac{1}{m} \\ 0 & 0 & 0 \end{pmatrix}$$

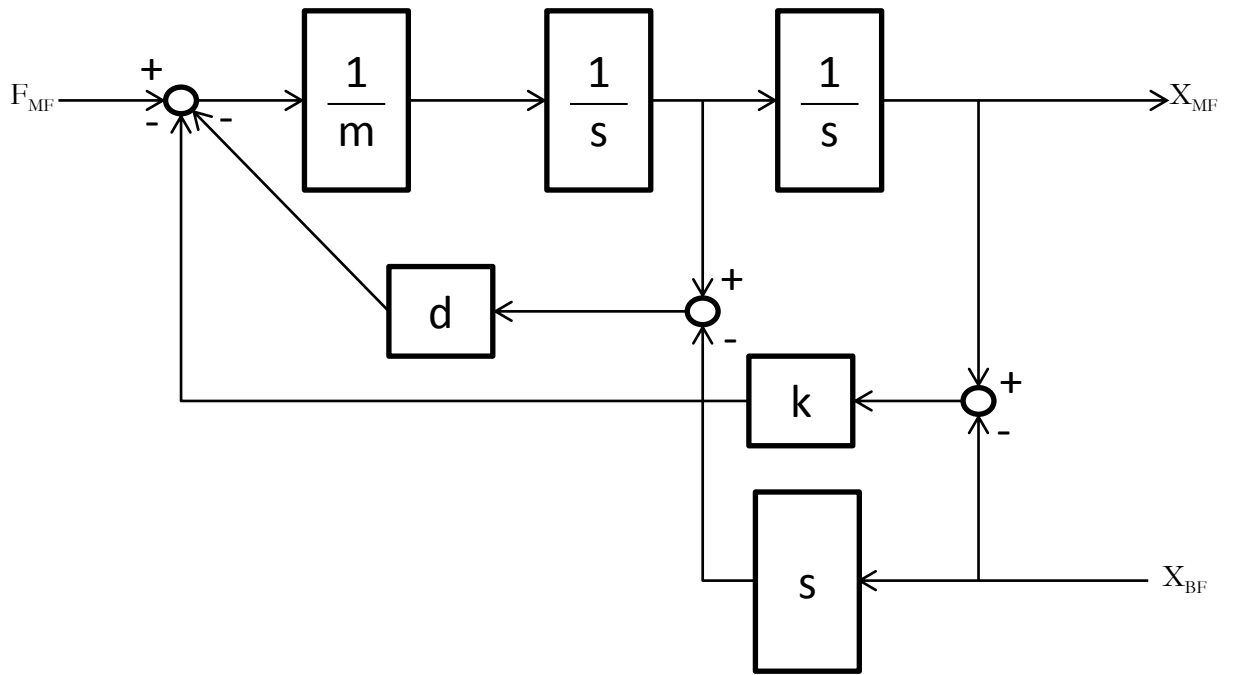


Figure 3-3 - Block schematic of the 1DoF model

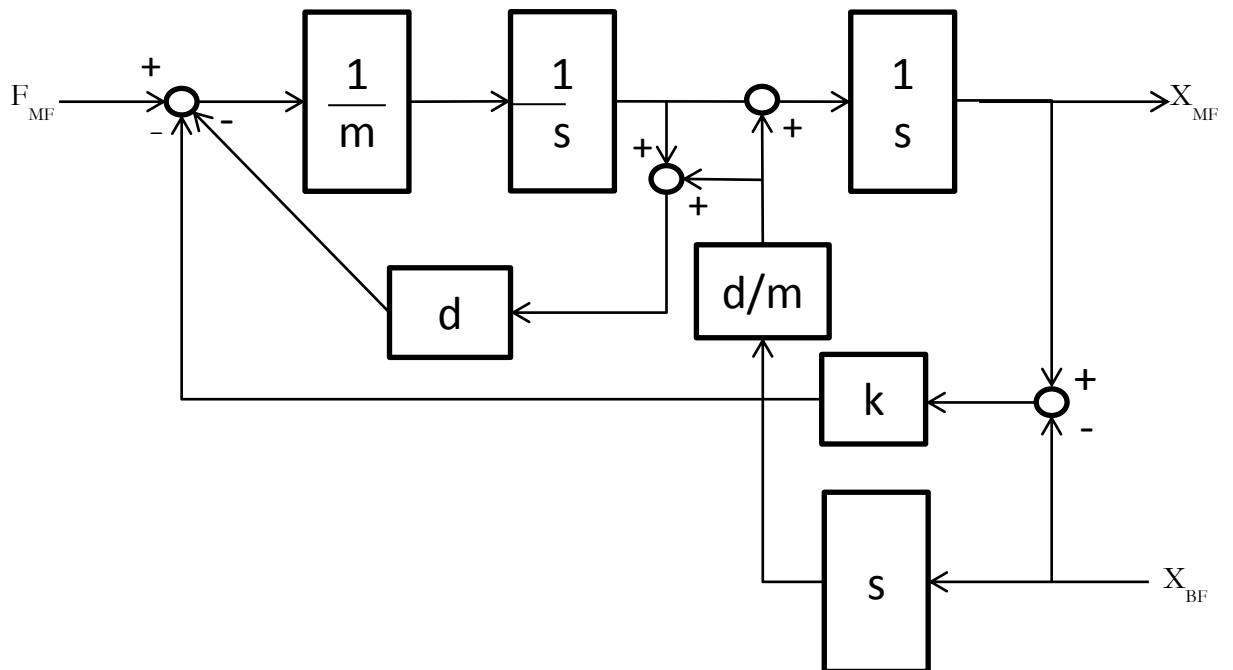


Figure 3-2 - Alternative block schematic on the 1DoF model

This solution would solve the equation, however the input vector 'u' needs to have the base speed. The base speed is not an obtainable variable in our simulation. To make a state space realization that represents the motion of our mass without the variable of speed two different options are available

1. Eliminating the term of base speed in the equation. This is equivalent to moving the damper from its relative to ground position to a skyhook position. However this solution reduces the precision of the model.

2. The second option is to generate a state space representation changing the states.

$\begin{pmatrix} \dot{X}_{MF} \\ \dot{X}_{MF} \end{pmatrix} \rightarrow \begin{pmatrix} X_{MF} \\ \dot{X}_{MF} - \frac{b}{m} X_{BF} \end{pmatrix}$ With this transformation you are allowed to keep the base position as an input and still have the relative damper.

The second methods block schematic is shown in Figure 3-2 which gives the state space representation below:

$$A = \begin{pmatrix} 0 & 1 \\ -\frac{k}{m} & -\frac{b}{m} \end{pmatrix}; B = \begin{pmatrix} \frac{b}{m} & 0 \\ \frac{k}{m} - \frac{b^2}{m^2} & \frac{1}{m} \end{pmatrix}; C = \begin{pmatrix} 1 & 0 \\ 0 & 1 \end{pmatrix}; D = \begin{pmatrix} 0 & 0 \\ \frac{b}{m} & 0 \end{pmatrix}$$

Where:

- m is a mass 1000 kg.
- k is the spring stiffness with a value of 9869.6 N/m which gives a resonant frequency of 0.5Hz
- b is the damping coefficient of the system with a value of 125N*s/m which belongs to a 2% of damping.

The outputs are the position of the metroframe and its speed. The model should give out acceleration as output as the sensor that is going to be used is an accelerometer. The acceleration cannot be obtained directly from the model so the option is to differentiate the speed. Due to limitations in Matlab's computations a perfect differentiator cannot be implemented and a high pass filter with a cut off frequency of 1 kHz is used instead. The reason for choosing 1 kHz is that it falls way beyond the range of frequencies of interest.

Figure 3-4 shows the transfer functions from the model created. The upper two sets of figures of magnitude and phase belong to the transmissibility and compliance of the model. This is also known as the open loop of the plant.

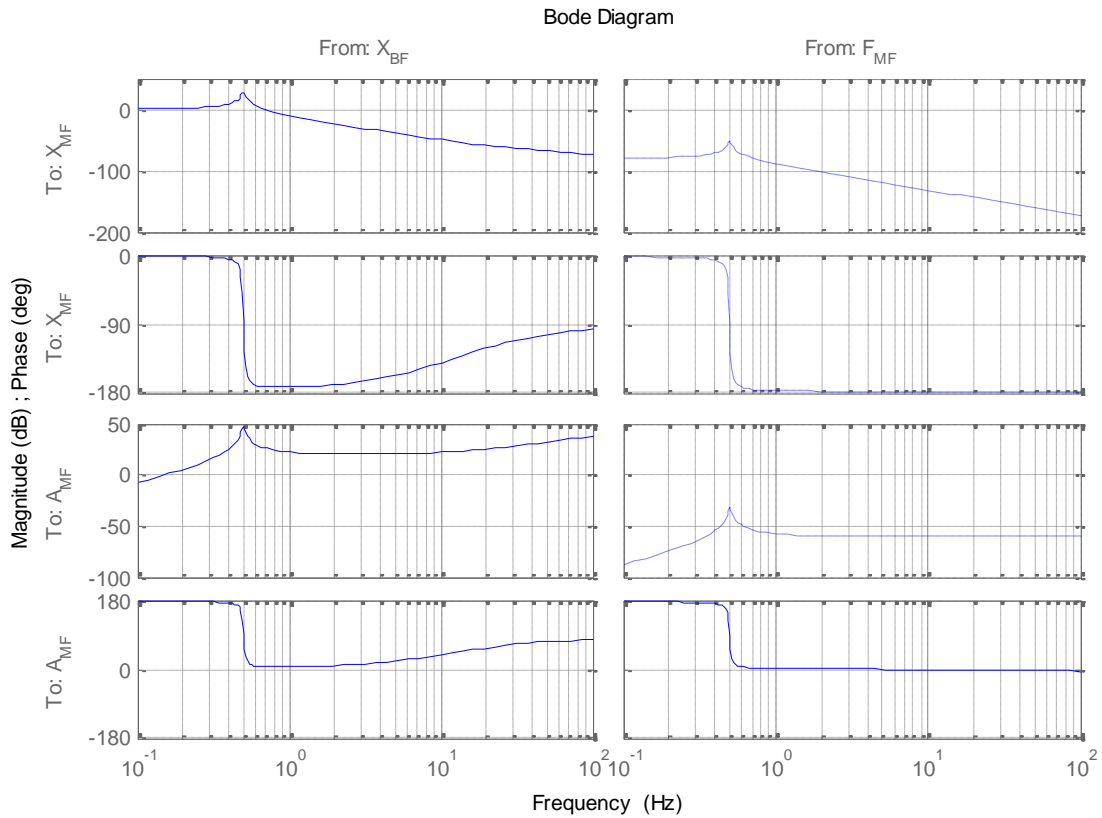


Figure 3-4 – Transfer functions from the mass-spring-damper model.

3.1.2 Control system

The controller to be designed will have two inputs, one from the position sensor and another from an accelerometer. The output of the controller will be force to the actuators that go into the plant. The systems closed loop diagram is shown below.

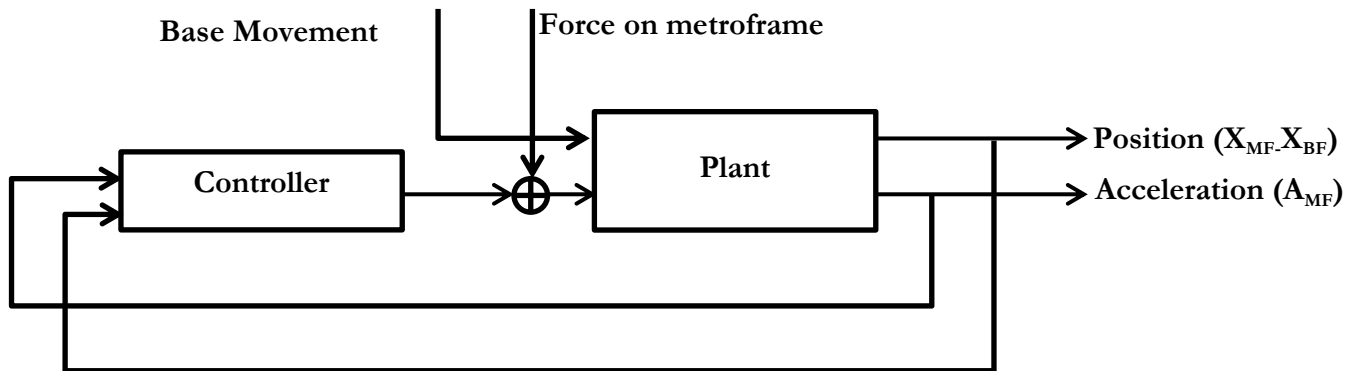


Figure 3-5 - Classic closed loop configuration of plant and controller

3.1.3 Performance requirements

The main goal of the controller is to increase the isolation of a system. Therefore the requirements on performance are the same as those shown in section 2.4 to improve on isolation.

Other requirements that will affect the controller's design are:

1. Reduction of control gain to minimize the effect of sensor noise
2. Adding robustness to the system by not adding gain to the controller at frequencies higher than 100 Hz.

3.1.4 Generic filter description

To be able to easily show the structure of a filter a generic description is going to be made. This generic description is going to be used in the rest of the document. The generic transfer function $F(s)$ for these filters is shown below:

$$F(s) = g * \prod_{i=1}^n \left(\frac{1}{2\pi f_i} s + 1 \right) * \prod_{j=1}^m \frac{1}{\left(\frac{1}{2\pi f_j} s + 1 \right)}$$

- g is the gain of the filter.
- Each f_i represents the frequency at which a zero is placed in the bode plot.
- Each f_j represents the frequency at which a pole is placed in the bode plot.

The format at which filters will be shown further on through the document is the following.

Example filter: $\left\{ \begin{array}{l} \text{Gain}(g) = 10 \\ \text{Zero Frequencies}(f_i) = 5 \text{ Hz} \\ \text{Pole Frequencies}(f_j) = 50 \text{ Hz and } 200 \text{ Hz} \end{array} \right.$

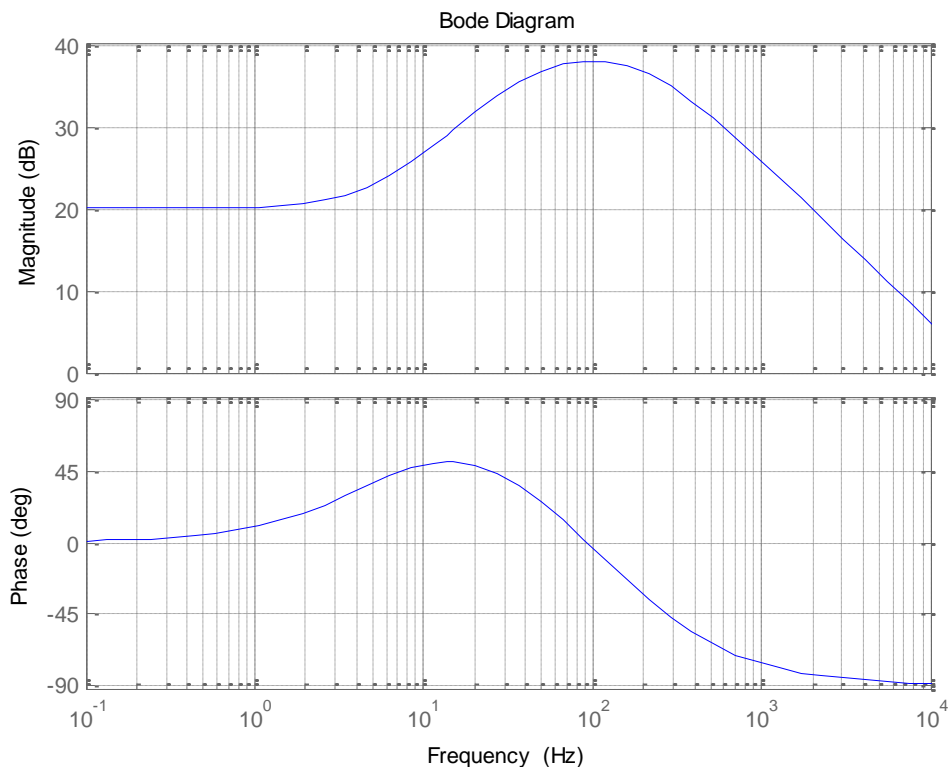


Figure 3-6 - Filter example

In the case of a double pole or double zero the equation used instead of two single poles or zeros is:

$$F(s) = \left(\left(\frac{1}{2\pi f_i} s \right)^2 + \left(\frac{\zeta}{\pi f_i} \right) + 1 \right) \text{ or } \frac{1}{\left(\left(\frac{1}{2\pi f_i} s \right)^2 + \left(\frac{\zeta}{\pi f_i} \right) + 1 \right)}$$

Where the symbol ζ is the damping coefficient and which would be $\frac{1}{2}$ if not stated otherwise.

3.2 Method 1: Classical loopshaping design

Our controller is made out of two different control loops. This means that two controllers have to be designed, one for the acceleration loop and another for the position loop. Each loop has its own characteristics and will be therefore used to fulfill different requirements.

The accelerometer gives absolute readings. This gives the possibility to damp the metroframe without affecting the transmissibility. For this reason the acceleration loop will be used to damp the resonant peak. It will also be used as far as possible to damp at frequencies close to the resonant peak.

The position loop gives relative data, adding gain to this loop has the same effects as increasing the damping. Therefore adding gain to frequencies over 0.5 Hz must be avoided so to not affect transmissibility. The position loop will be used to add integral action to the controller to prevent from low frequency noises that can displace the metroframe.

The design process will be the following:

1. First the acceleration loop will be tuned on performance and stability.
2. Afterwards the position loop.
3. The performance and the stability of both controllers working in parallel will be tested. If the system is either unstable or the performance is not considered good enough the steps will be repeated.
4. Noise parameters will be added. The performance on noise cancellation will be checked, in the case of unsatisfactory results the steps will be repeated.

With this method the best performance is going to be achieved through the iteration of the steps in a trial and error manner.

The next sections will explain the thought process behind the decisions and show the final controller for each loop.

3.2.1 Acceleration loop design

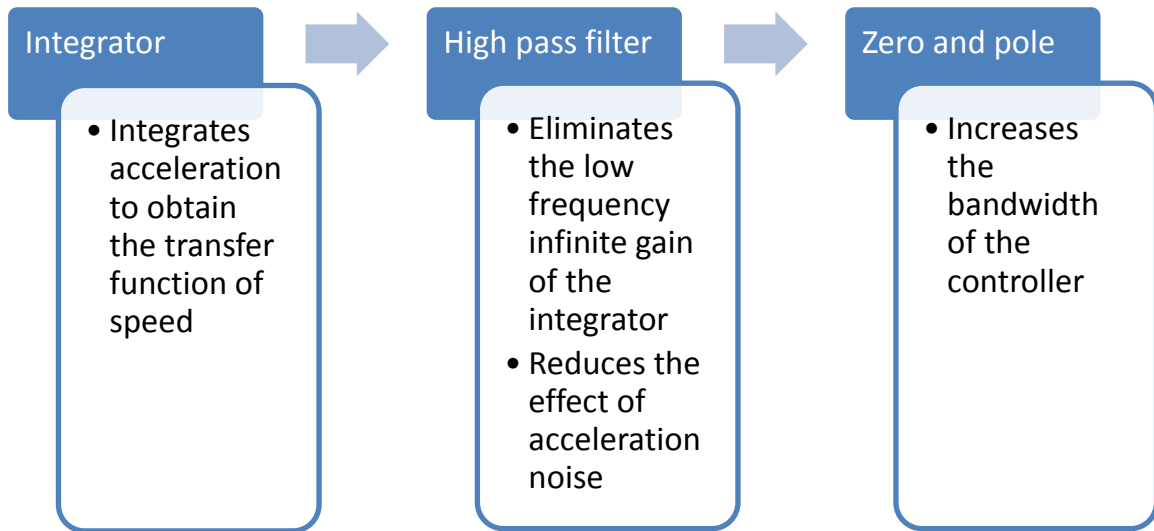


Table 1 - Design process used for the acceleration loop controller

$$\text{Acceleration loop controller: } \left\{ \begin{array}{l} \text{Gain}(g) = \frac{10000}{0.6\pi} \\ \text{Zero Frequencies}(f_i) = 1 \text{ Hz} \\ \text{Pole Frequencies}(f_j) = 0.3 \text{ Hz and } 10 \text{ Hz} \end{array} \right.$$

Accelerometers offer acceleration data but for the damping of the resonance peak and its surroundings speed has better frequency characteristics as can be seen in Figure 3-8. Using speed can also be related to having the feedback loop act as a damper.

To obtain the transfer function of speed is the reason to start the design of the controller with an integrator. To avoid infinite gain on dc a high pass filter is added. This high pass filter would not be necessary in the ideal case but is essential when noise is added. In our situation it is helpful to have the high pass cut frequency at the highest frequency possible to reduce the effect of the accelerometer's noise.

To increase the bandwidth at which the acceleration loop has influence on the closed loop a zero and a pole are added. The final result in the open loop of the controller and plant can be seen in Figure 3-10.

The nyquist stability criterion (Figure 3-9) shows us that this controller is stable. Going back to the open loop bode plot it becomes obvious that the loop obtains its stability because the phase never passes through the 180° .

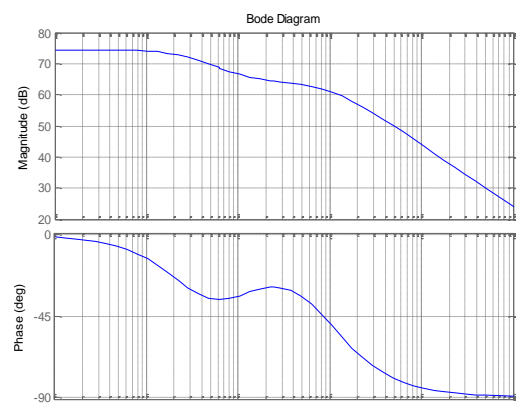


Figure 3-7 - Bode plot of Controller's freq. response

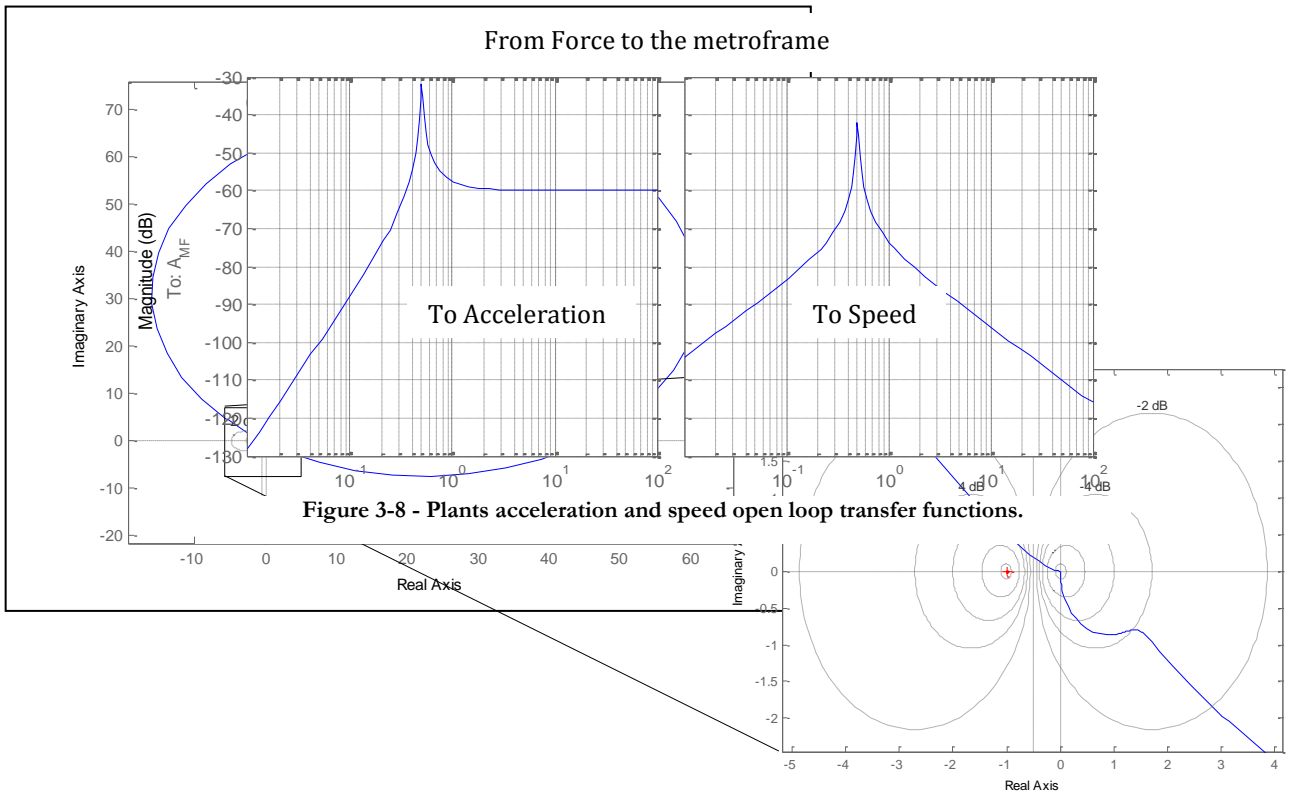
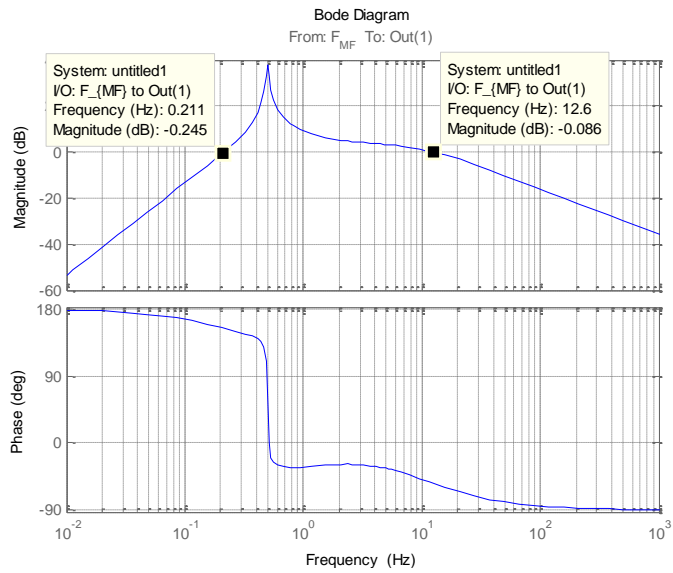


Figure 3-9 - Open Loop Nyquist plot.



3.2.2 Position loop design

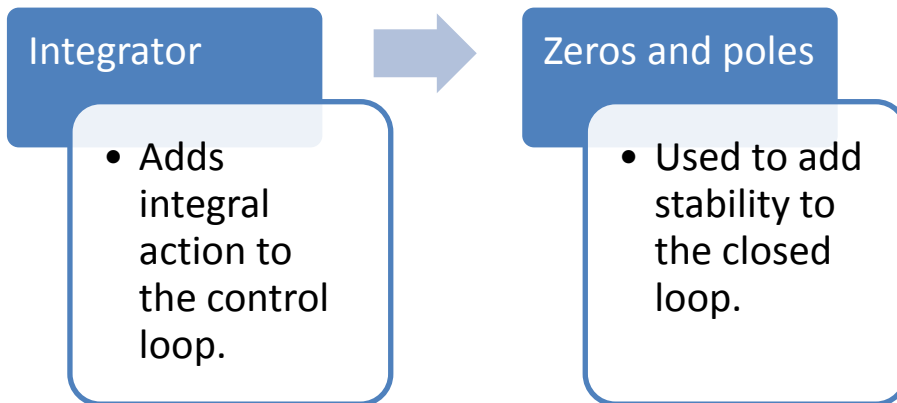


Table 2 - Design process used for the position loop controller

$$\text{Position loop controller: } \begin{cases} \text{Gain}(g) = 1000 \\ \text{Zero Frequencies}(f_i) = \text{two at } 0.2 \text{ Hz} \\ \text{Pole Frequencies}(f_j) = 0 \text{ Hz and two at } 1.5 \text{ Hz} \end{cases}$$

The position loop has to add integral action to the controller, therefore starting with an integrator is the natural choice. It also decreases gain towards high frequencies so having an integrator is also desirable in this sense. This could be the final form of the controller; however, in this state the closed loop is potentially unstable as shown in Figure 3-11. In the Nyquist plot shows how adding a little bit of gain to the loop would make the graph encircle the -1 value in the real axis (red cross) making the closed loop unstable. For this reason some corrections to the controller have to be added to make it more robust.

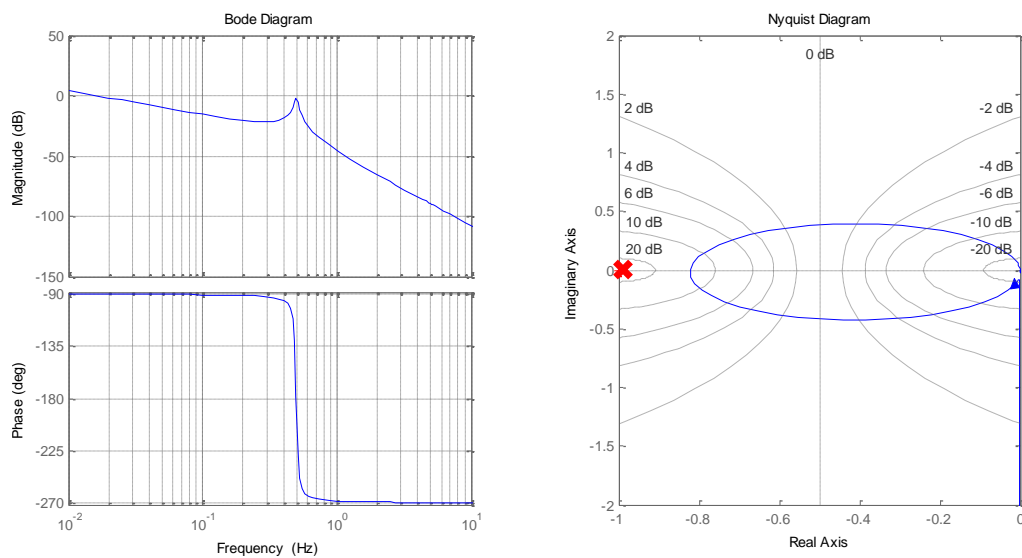


Figure 3-11 - Open Loop Bode and Nyquist plot with only the integral as controller.

The instability comes from the plants shift of -180 degrees at 0.5 Hz from the -90 degree line that makes the phase go through the 180^0 line. Therefore a good correction would be adding a phase shift of 90 degrees at 0.5 Hz. This may be achieved adding a zero before the 0.5 Hz to add phase, also a pole will be added at a symmetric distance to recover the -1 slope. To accelerate the addition of phase and to narrow the band between the pole and zero we will double the amount of poles and zeros.

Figure 3-12 is the final controller in the open loop.

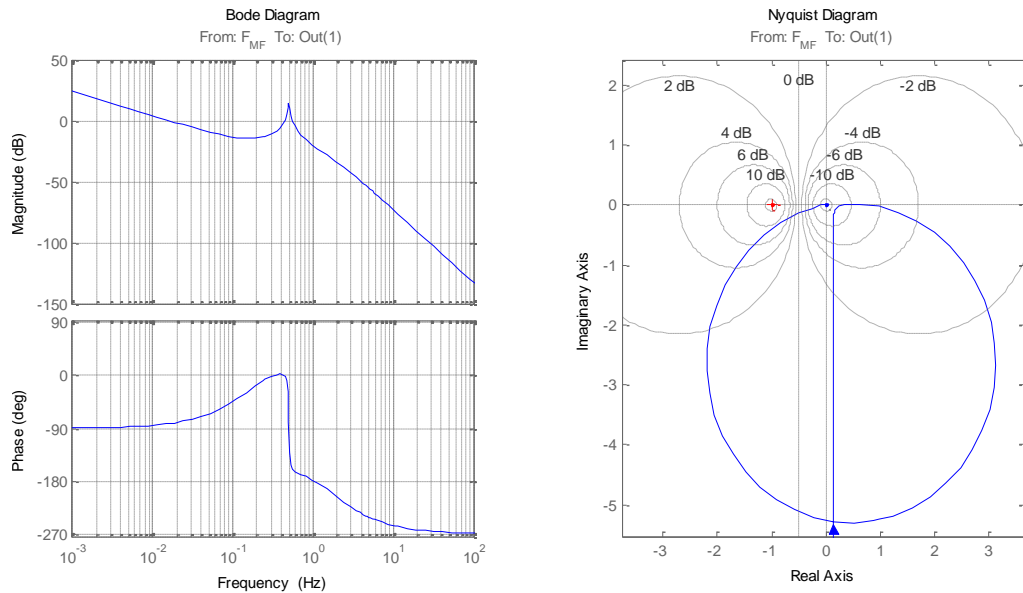


Figure 3-12 - Open Loop Bode and Nyquist plot.

3.2.3 Full controller stability analysis

The full controller is the sum of both, the AccController and the PosController. To be able to study the stability of the system we are going to use the following configuration (Figure 3-13)

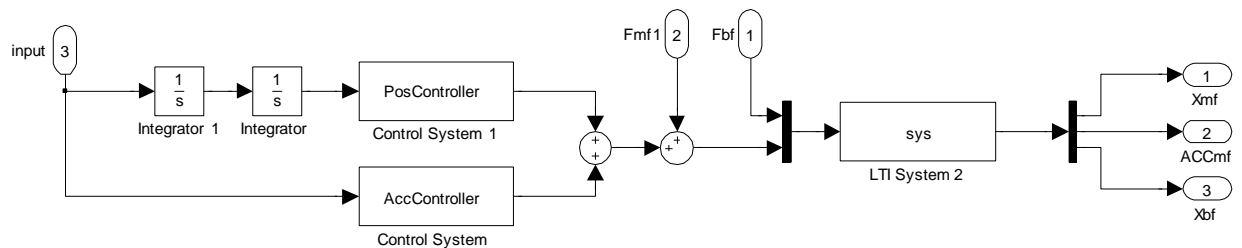


Figure 3-13 - Full controller open loop configuration.

This configuration in an open loop for both controllers is possible because the inputs of the controllers are acceleration and position which can be related with each other by the formula $position = \iint acceleration dt$. This seem incorrect as the position seems to be considered absolute and this is not the real data used in the controller. However as it is only used to study the stability of the system the acceleration and position can be taken as generics.

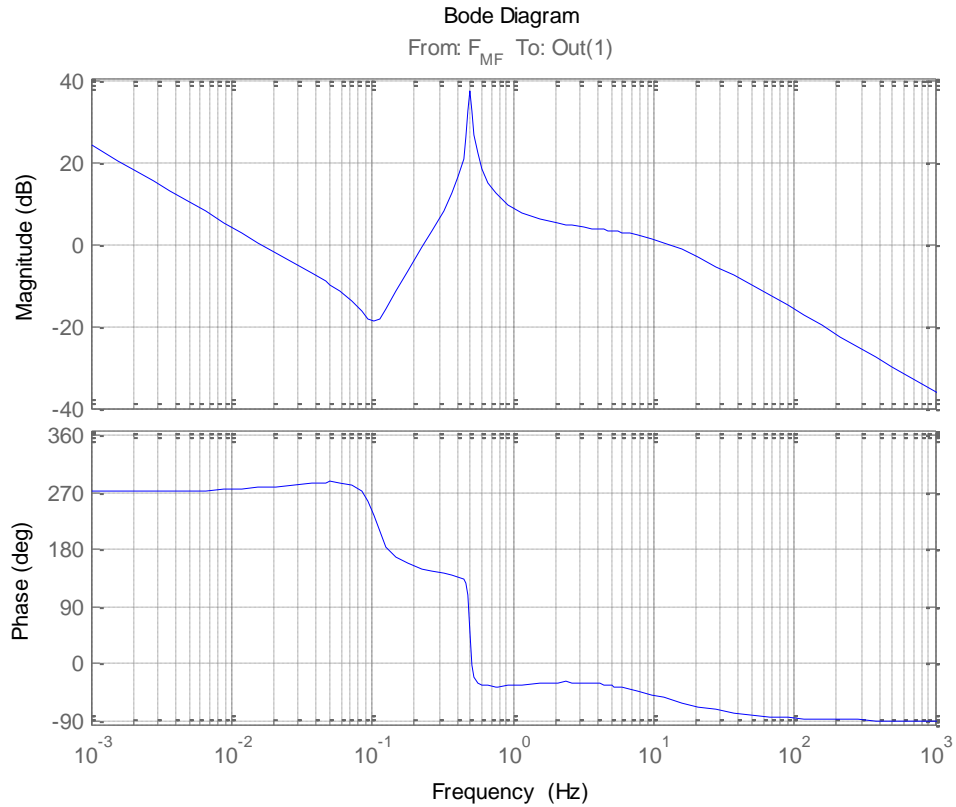


Figure 3-14 - Full controller open loop Bode plot.

The frequency response bode plot and Nyquist diagram obtained from the system in Figure 3-13 are shown in Figure 3-14.

The open loop bode plot shows the features in the controller, the integral action at low frequencies, the peak damping and the broader bandwidth damping. In the nyquist plot (Figure 3-15), we see how the system is stable. The gain margin that is achieved is around the 20 dB.

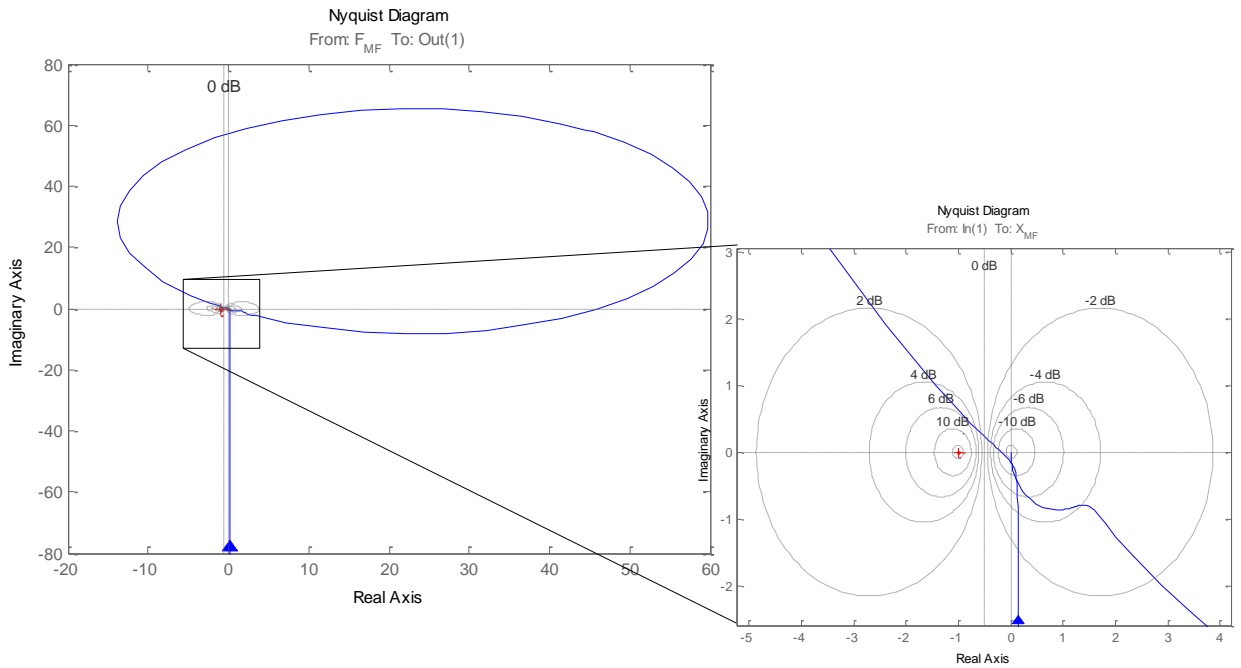


Figure 3-15 - Open Loop Nyquist plot.

3.2.4 Controller's Results

Finally, the systems performance will be tested to see how good it covers the design specifications. In Figure 3-17 the closed loop transmissibility and compliance plots (green) are shown on top of the plants open loop plots (blue) to show the improvements achieved.

The peak has been correctly damped and shifted towards lower frequencies, in this case to 0.2 Hz. The damping on higher frequencies is an improvement in the isolation from disturbances. The integral action from the position controller can be seen in the compliance as a decoupling from the Force also at low frequencies starting at 0.01 Hz.

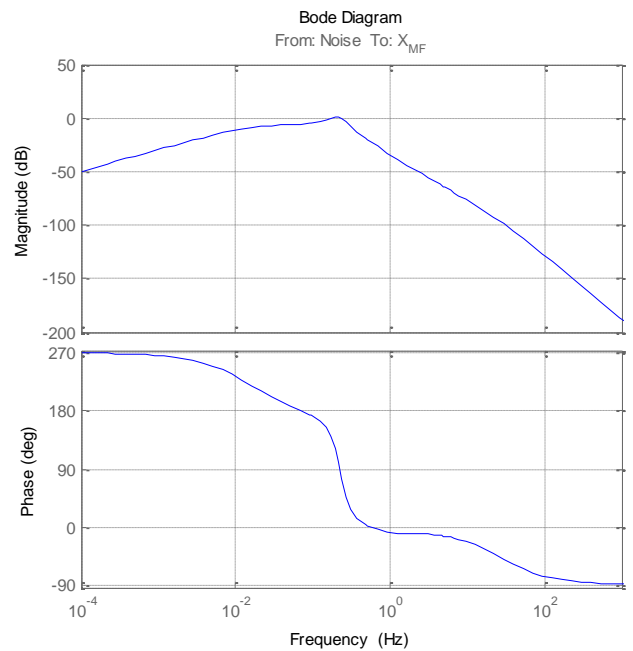


Figure 3-16 - Noise-Displacement frequency bode plots

Figure 3-16 shows the gain of the noise at different frequencies from the sensor to the position of the metroframe. Logically the biggest gain in the noise will be around the

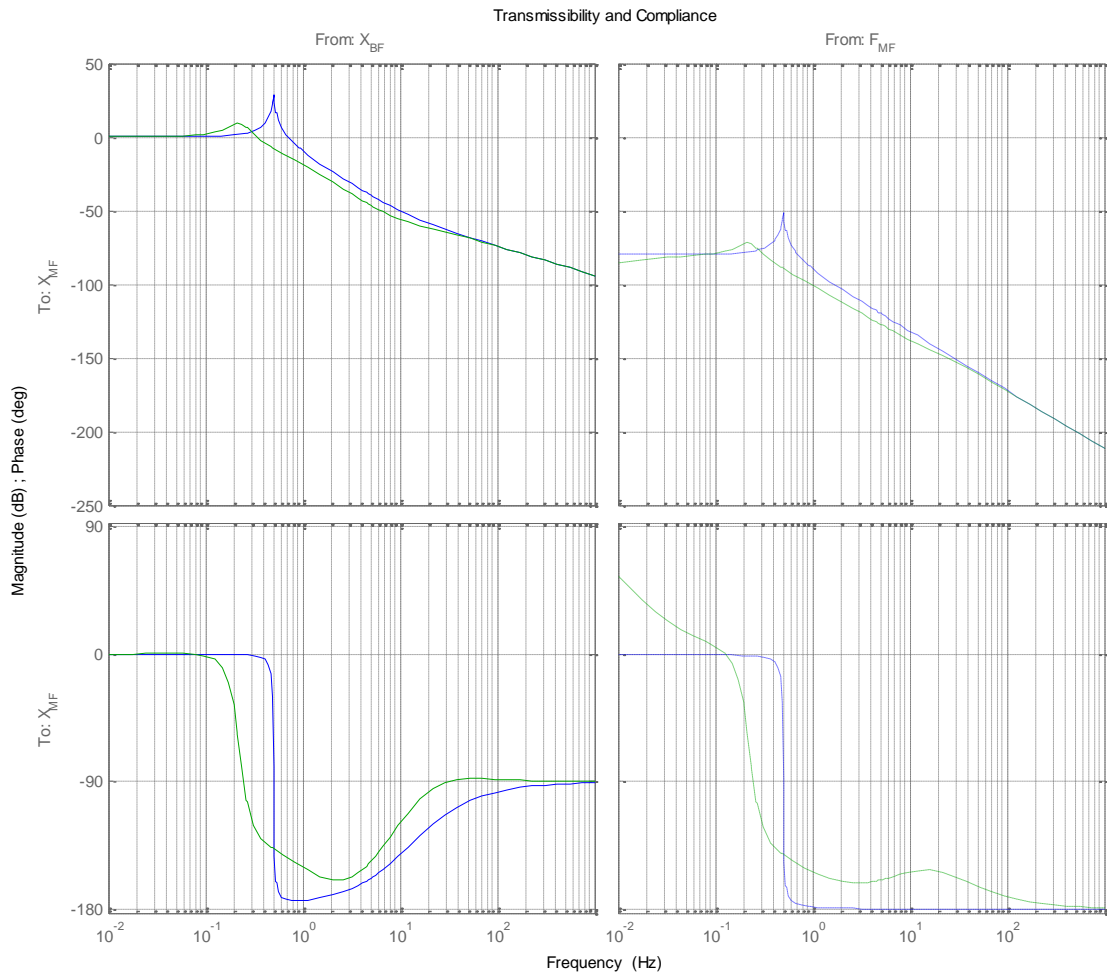


Figure 3-17 - Transmissibility and compliance of the classic loopshaping method controller

resonance where the controllers action is at its highest.

3.2.5 Conclusions

This method is highly intuitive and the relation between the results and the tuning process itself is very strongly related. This gives a feel of control over the entire process. However, reaching a balance on performance is not straightforward. In order to achieve the results just described the number of iterations were very high, especially when the noise of the sensors was added to the design because the number of variables to be taken into account were doubled.

3.3 H-infinity Theory

The design process in the H-infinity methods is very different from the classical loopshaping method seen previously. The way this method obtains its controllers and where the tuning process lies will be shown in the following sections.

3.3.1 H-infinity norm

In the H-infinity method the controller is obtained through optimization algorithms. These algorithms generate controllers that preserve internal stability while minimizing the H-infinity norm of the system.

$$\|H(s)\|_{\infty} = \sup_{\omega \in \mathbb{R}} \bar{\sigma}(H(j\omega))$$

The H-infinity norm seen above is the norm applied to a certain input output system, where the norm is telling us the maximum gain of the frequency responses of the singular values of the system i.e. the SISO interpretation of this norm is the maximum gain in the magnitude of the frequency response of the system.

3.3.2 H-infinity design algorithms

The H-infinity algorithms make use of the following configuration of plant and controller system shown in Figure 3-18, where the different signals are:

- w: Systems inputs
- z: Systems outputs
- u: Controllers output (actuators)
- v: Controllers input (measurements)

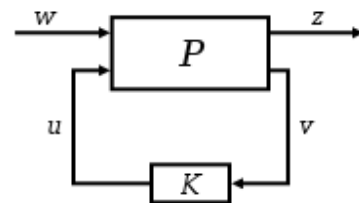


Figure 3-18 - H-infinity plant plus controller configuration

The actual way that the H infinity controller design algorithms generally work is the following. For the configuration of plant and controller shown in Figure 3-18 an optimum achievable H-infinity value exists for the closed loop transfers ($w \rightarrow z$). This value is unknown by the algorithm so it will try to approximate this optimal value, thus not really obtaining an optimal controller but a good approximation. This minimum H-infinity value obtained by the algorithm will be called gamma (γ). This gamma will be obtained by iteratively giving a smaller value to gamma and studying the viability in terms of stability and performance till a minimum is reached. Once the value is obtained the algorithm will proceed to synthesize one of the infinite number of possible controllers for the achieved gamma.

3.3.3 Augmented plant

As the controller is synthesized by the H-infinity algorithm the only way to add specifications to that controller is by modifying the plant. This modification of the plant is done by adding weighting functions to the inputs and outputs of the system as shown in Figure 3-19. The blocks added (W_w and W_z) are matrices of filters for all inputs and outputs that give the possibility to modify the closed loop transfer functions.

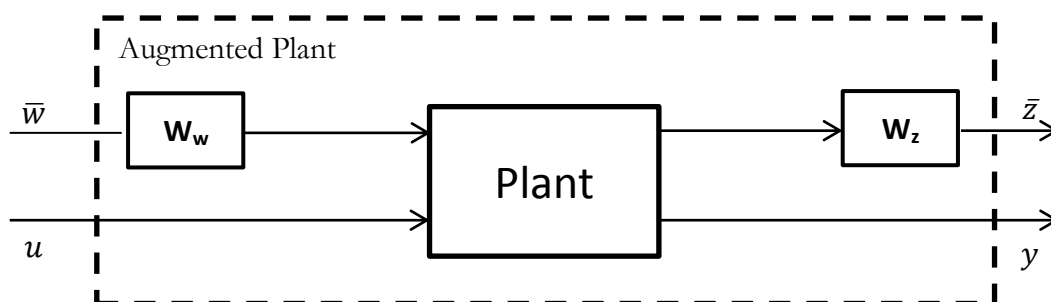


Figure 3-19 - Augmented plant scheme

Adding gain to different frequencies in the weights will indicate that there has to be a tighter control on those frequencies. It is equally important to specify frequencies where the minimization is not necessary so to concentrate the control action on the bands of interest, this is done by lower gains on the weights.

There will be a weighting function for each input to the plant, this is generally used to characterize the inputs frequency response. Characterizing the inputs helps to optimize the controller as there will be more control action at frequencies where the input has more gain. The outputs from the plant and the frequencies at which they will have more or less gain are given by the design requirements. For instance, if the acceleration of a system has to be minimized at high frequencies the output z from the plant will be its acceleration and the weight will be a high pass filter.

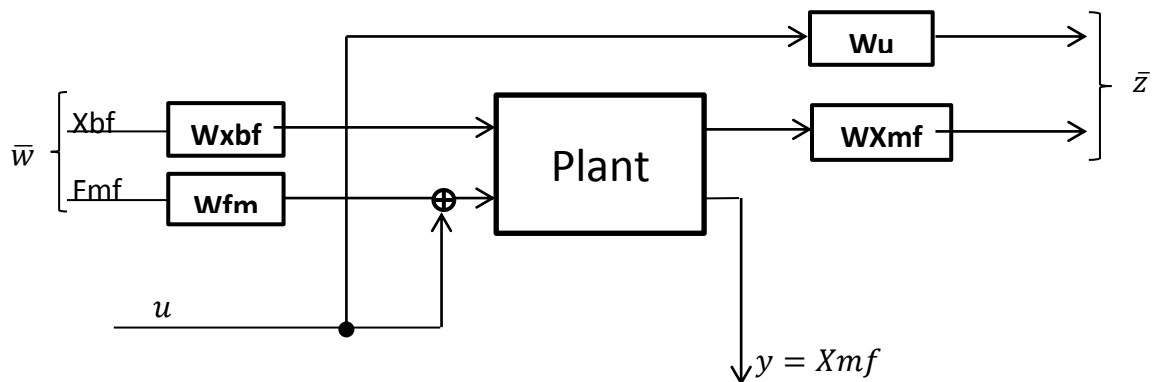


Figure 3-20 - Augmented plant setup for simplified example

As an example on how the H-infinity algorithm works we will design a controller for the simple 1 mass-spring-damper system described in section 3.1.1. As part of the example a theoretical absolute position sensor will be used to make a SISO controller, this choice is made to get simpler results. The inputs to the augmented plant will be the same as the plant's (X_{bf} , F_{mf}) and as outputs the position of the metroframe and the controller's output (X_{mf} , u). All the weights will be nothing but gains to simplify the example. The schema of this augmented plant setup is shown in Figure 3-20

What is expected from this setup is the following. Minimizing the transfers from the inputs to the metroframe's position can be achieved by adding gain to the controller. But as also the transfers from the inputs to the controller's output are being minimized a compromise must be achieved.

Figure 3-21 shows the loop without any controller (open loop) and on top the closed loop

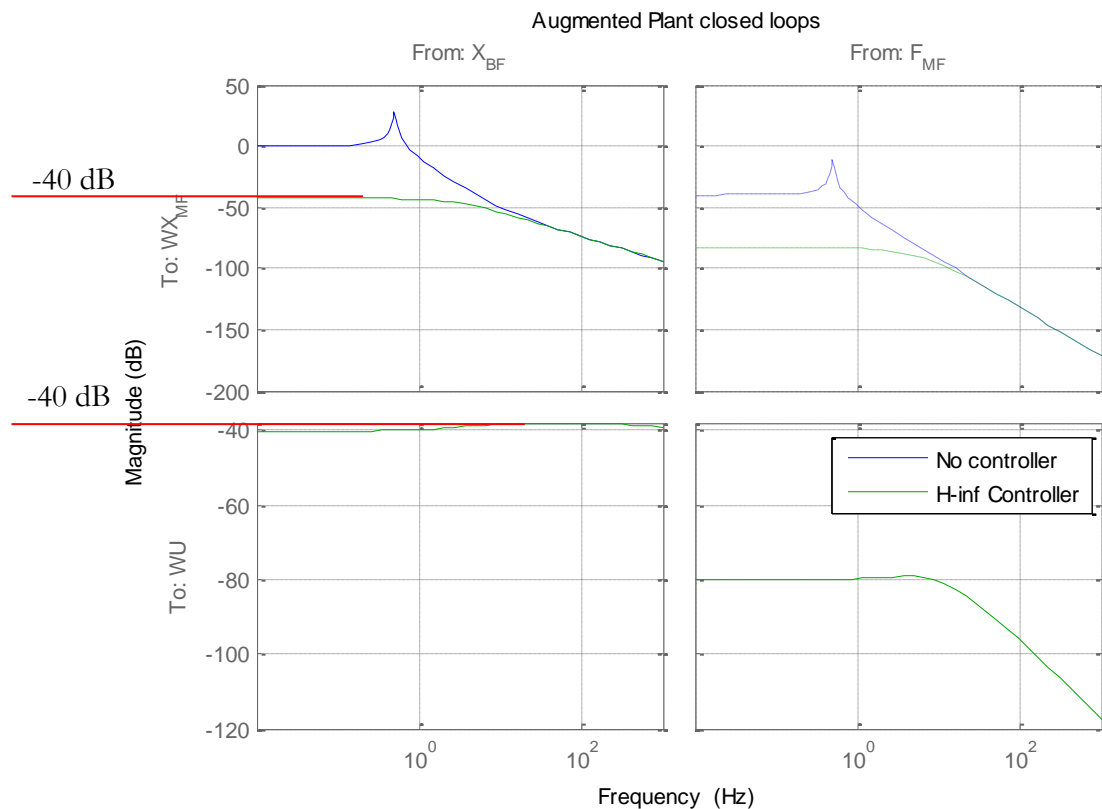


Figure 3-21 - Augmented plant's open Loop and H-infinity controller's closed loop comparison.

with the H-infinity controller for every input to output of the augmented plant. This comparison will let us see what the H-infinity algorithm tries to do with the transfer functions of the augmented plant.

What happens is that the algorithm via means of the controller tries to minimize the transfer's magnitude as much as possible. The stopping point is when adding more gain to the controller will make the H-infinity norm increase its value instead of decreasing it. This is seen in our example in the two left transfers, where adding gain would make the output to WU be bigger in magnitude than the output to Wxmf.

On top of this minimization process which searches for the best performance the algorithm will make sure that the system maintains its stability. A small detail to be noted, results for vibration control would be excellent if the ideal absolute sensors as used for the example existed.

3.3.4 Controllability and observability

Two properties that need to be fulfilled by the augmented plant in the design of a suitable H-infinity controller are controllability and observability (Skogestad & Postelwaith). These properties are defined as follows

- Controllability – *A system is considered controllable if and only if the system states can be varied by changing the systems input.*

- Observability – *A system is considered observable if and only if the initial state can be determined from the observed system's output through an interval of time for all the internal states.*

Both these conditions have to be met by the augmented plant from the controller's point of view, in order to be able to generate a stabilizing controller. The following analytical procedures can be used.

For both the controllability and observability the matrices from the state space representation will be used. Formerly $\begin{cases} \dot{x} = Ax + Bu \\ y = Cx + u \end{cases}$ where:

- x is the state vector
- y is the output vector
- A is the states matrix
- B is the input matrix
- C is the output matrix
- D is the feedforward matrix.

The A, B, C, D matrices enable us to easily characterize any model with two equations independently from the amount of partial differential equations it may be composed of.

To determine controllability we will need to obtain the controllability matrix. And the system will be considered controllable when the controllability matrix is full rank. The controllability matrix is defined as follows, where A and B are the A, B matrices of the state-space description, p is the number of inputs and q the number of states.

$$\zeta = [B \quad AB \quad A^2B \quad \dots \quad A^{p-1}B] \in R^{p \times pq}$$

Observability has an analog analysis to controllability. In this case the observability matrix will be created. The system will be considered observable if the matrix is full rank. C and A are the C, A matrices of the state space description and p is the number of inputs.

$$Q = \begin{bmatrix} C \\ CA \\ CA^2 \\ \vdots \\ CA^{p-1} \end{bmatrix}$$

Observability and controllability are achieved for all augmented plants designed in the following sections.

3.3.5 Robust stability analysis.

Once a controller has been designed it is necessary to check for robust stability. The small gain theorem provides tools to do so, stated as following:

Let $W_1, W_2 \in \text{RH}_\infty$, $P_\Delta = P_0 + \Delta$ for $\Delta \in \text{RH}_\infty$ and K be a stabilizing controller for P_0 . Then K is robustly stabilizing for all $\Delta \in 1/\gamma \beta \text{RH}_\infty$ if and only if $\|W_2 K S_0 W_1\|_\infty < \gamma$

Where γ is the H-infinity value from 3.3.2 and RH_∞ is the H infinity space which consists of all proper and real rational stable transfer matrices.

What this property says is that for additive uncertainties a system that is nominally stable with a certain maximum gain (γ) will also be robustly stable if the maximum gain of the disturbed plant falls beneath 1. This theorem allows guaranteeing robust stability for unstructured uncertainties. The problem with this kind of statement is that because of the unstructured nature of the uncertainty the margin given to provide the robustness can be too conservative.

Because we know that the gamma value is the maximum gain of the system at just one frequency we can make this statement more precise by extending it to more frequencies. In a SISO system this can be seen as shown in the following example.

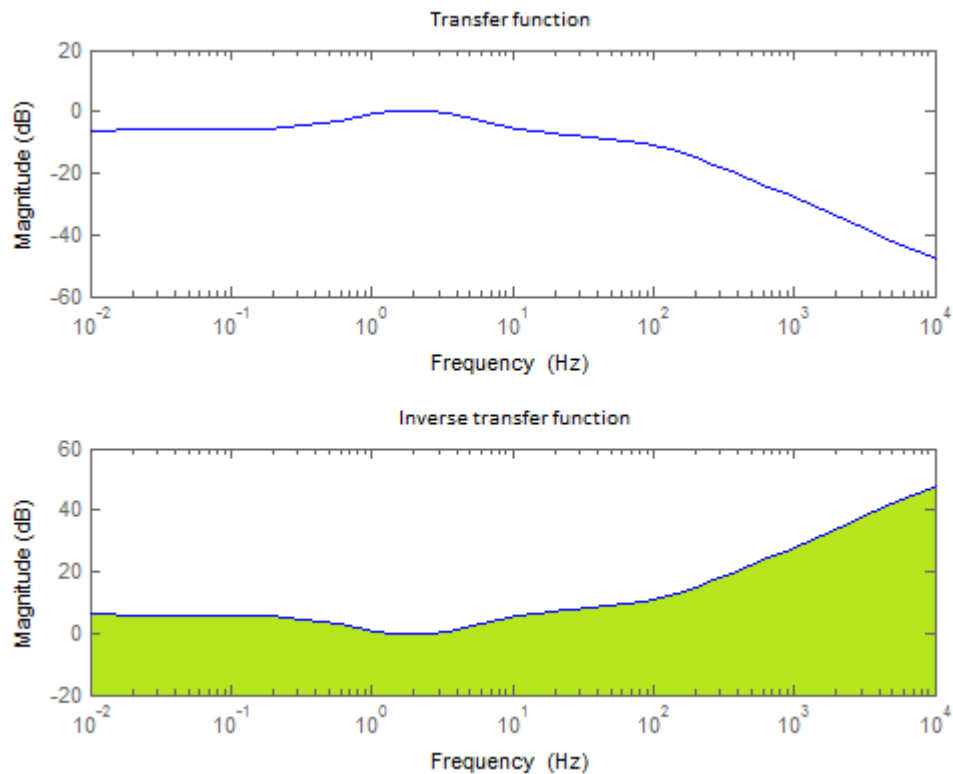


Figure 3-22 - Example of allowable uncertainty in a SISO system

The zone marked in green would be were the uncertainty of this random transfer function would be allowed to be to keep robust stability in the system.

In the case of MIMO systems the visualization is not possible as the maximum gain of the full system doesn't

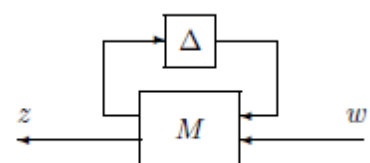


Figure 3-23 - Uncertainties in the LFT configuration.

have a graphical visualization. However in the disposition of Figure 3-23 (where M is the closed loop of plant and controller) if the uncertainty has a structured nature it can be “pulled out” of the system and visualized in the same fashion as the SISO systems.

3.4 Method 2: Minimization of metroframe’s movement design

To use the metroframe’s absolute position to be minimized by the controller as seen in Figure 2-1. This approach has been used in (Wal & Heertjes) as base controller design method to elaborate its study. In (Nakashima, Tsujino, & Fuji, 1996), (Pantazi, Sebastian, Pozidis, & Eleftheriou, 2005) and (Watanabe & al., 1996) similar approaches are used, the idea is making use of the optimal capabilities of the H-infinity method to minimize one of the variables in the system. This seems like the obvious approach to H-infinity and that is why it has been

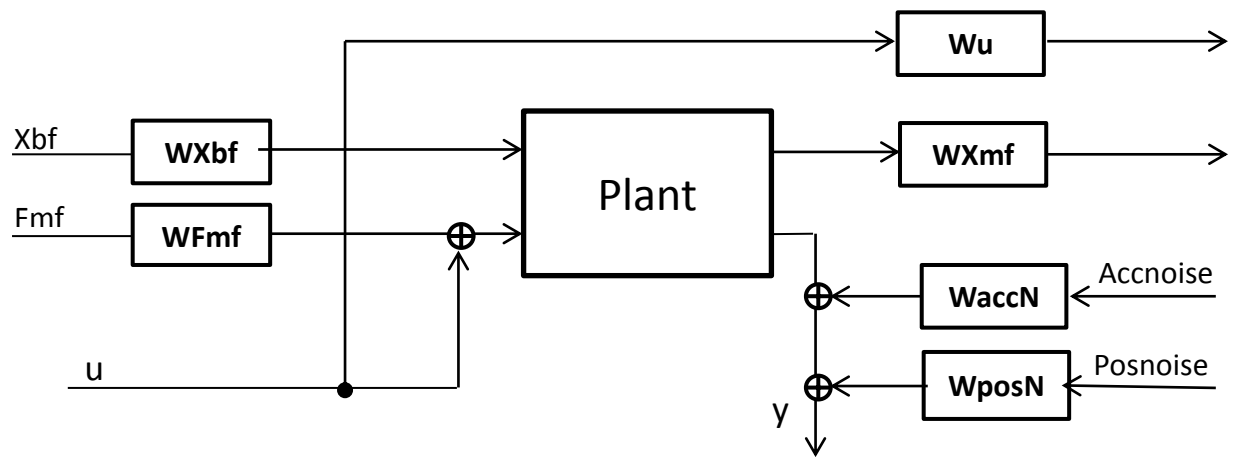


Figure 3-24 - Augmented plant for minimization of the metroframe

extensively used in applications in search of alternative methods of control.

The effect of changing the different weightings is the following

- **WXmf:** Filters the frequency response of the position of the metroframe. Adding gain to this weight will increase the control action at the selected frequencies. This weight is the base for adding design requisites to the system in this configuration.
- **Wu:** Affects the magnitude of the controllers. Adding gain to this weight reduces the control action. Helps to set boundaries in frequency to limit the action of the controllers in regions of less interest.
- **WXbf and WFmf:** Related directly to the transmissibility and compliance. These weights will be shaped to represent the actual behaviour of the inputs baseframe movement and force to the metroframe.
- **WposN and WaccN:** They will describe the real behaviour of the noises of the sensors and reduce the controllers output where the noise is bigger in magnitude.

3.4.1 Weights used

The following weights have been applied.

Output weights:

- W_{xmf} : $\begin{cases} Gain(g) = 10^{-7} \\ Zero\ Frequencies(f_i) = 0.05Hz\ and\ double\ 0.5Hz \\ Pole\ Frequencies(f_j) = 10Hz\ and\ double\ 100Hz \end{cases}$

Our interest is centred from the resonance frequency until around 100 Hz. The passive response of the plant decreases with a -2 slope from its resonance frequency at 0.5 Hz. This means that in the closed loop this slope is also present reducing the importance for the H-infinity algorithm of higher frequencies at a rate of -40 dB per decade. To even out the weight for all the band of interest a +2 slope will be added from 0.5 Hz to a 100 Hz evening out the frequency response for our region of interest, the result can be seen in Figure 3-25. On top of this a high pass filter will be added. This filter will be used to emphasize the region of interest building a difference in gain between low frequencies (less interesting) and high frequencies (more interesting).

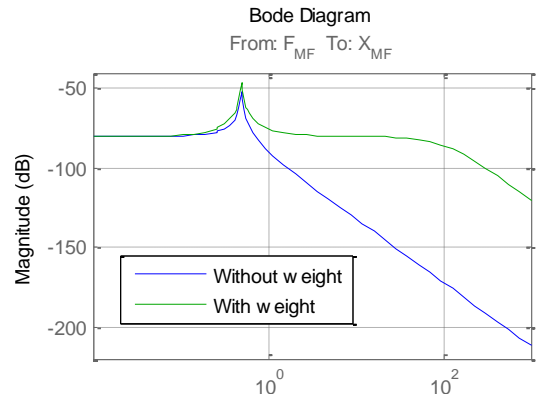


Figure 3-25 - Effect of evening out compliance's -2 slope

- W_u : $\begin{cases} Gain(g) = 10^{-12} \\ Zero\ Frequencies(f_i) = 0Hz \\ Pole\ Frequencies(f_j) = 250Hz \end{cases}$

This weight will be a first order high pass filter that will deal with unwanted control action at high frequencies. The cut off frequency will be at 250 Hz. The gain of this weight has been chosen through trial and error to obtain the desired results.

The input weights:

- W_{xbf} and W_{Fmf} : $\{ Gain(g) = 10^{-5}\ and\ 10\ respectively$

Both these weights are just gains that relate to the inputs magnitudes 10 micrometers for the baseframe's position movement and 10 Newton for the Force to the metroframe.

- $W_{PosNoise}$: $\{ Gain(g) = 10^{-6}$
Just a gain in order to replicate white noise.

- \bullet **WAccNoise:**

$$\begin{cases} \text{Gain}(g) = 10^{-3} \\ \text{Zero Frequencies}(f_i) = 5e^{-4}\text{Hz}, 5e^{-2}\text{Hz}, 5\text{Hz} \\ \text{Pole Frequencies}(f_j) = 5e^{-5}\text{Hz}, 5e^{-3}\text{Hz}, 5e^{-1}\text{Hz} \end{cases}$$

The noise of the accelerometer is a $1/\sqrt{f}$ ramp that decreases until 5Hz.

3.4.2 Results

In Figure 3-26 you can see the obtained transmissibility and compliance.

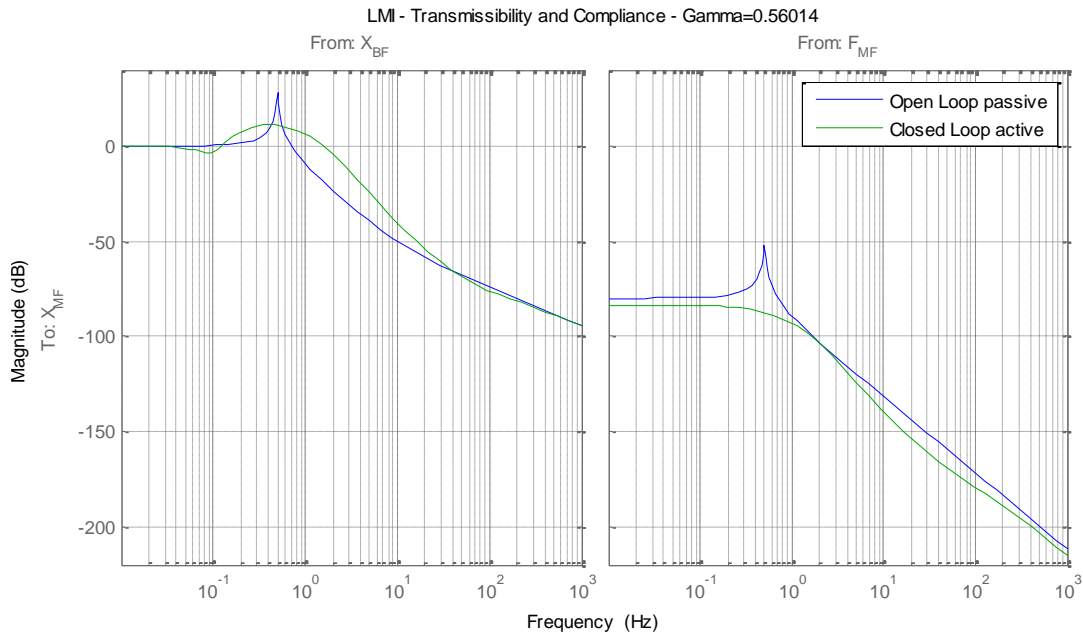


Figure 3-26 - Obtained transmissibility and compliance for the minimization of position approach

The transmissibility is worse in the closed loop than what the passive system offers and the compliance doesn't show a big improvement. The result for transmissibility is obtained because the algorithm decides that adding gain to the position loop - even though transmissibility is worsened - is worth to reduce the H-infinity norm of the system.

The next figure (Figure 3-27) shows the open loop of the final implementation of controller and plant. What is worth noticing here is the bandwidth of the system, this is determined by

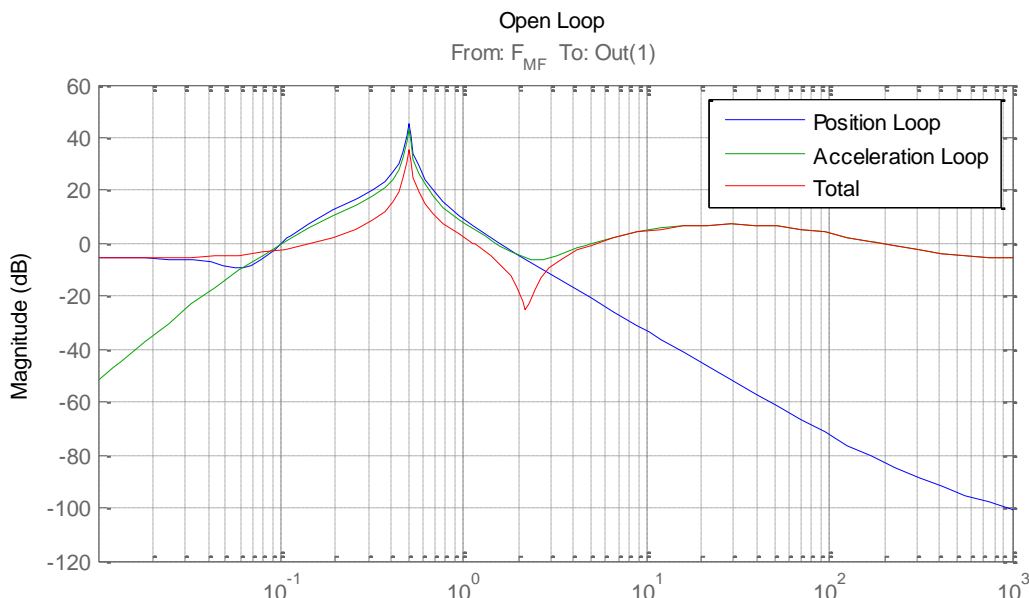


Figure 3-27 - Open loop for the minimization of position approach

the last frequency where the open loop falls below the 0dB, in the shown figure this happens at 200Hz. This bandwidth is result of the weight on the controller output (W_w), without this weight the bandwidth goes up to 10kHz.

Robust stability analysis

The controlled system with the controller obtained by this method is robustly stable if the uncertainties obey the following rules.

The maximum allowable uncertainty gain for structured uncertainties for each input and output is shown in Figure 3-28

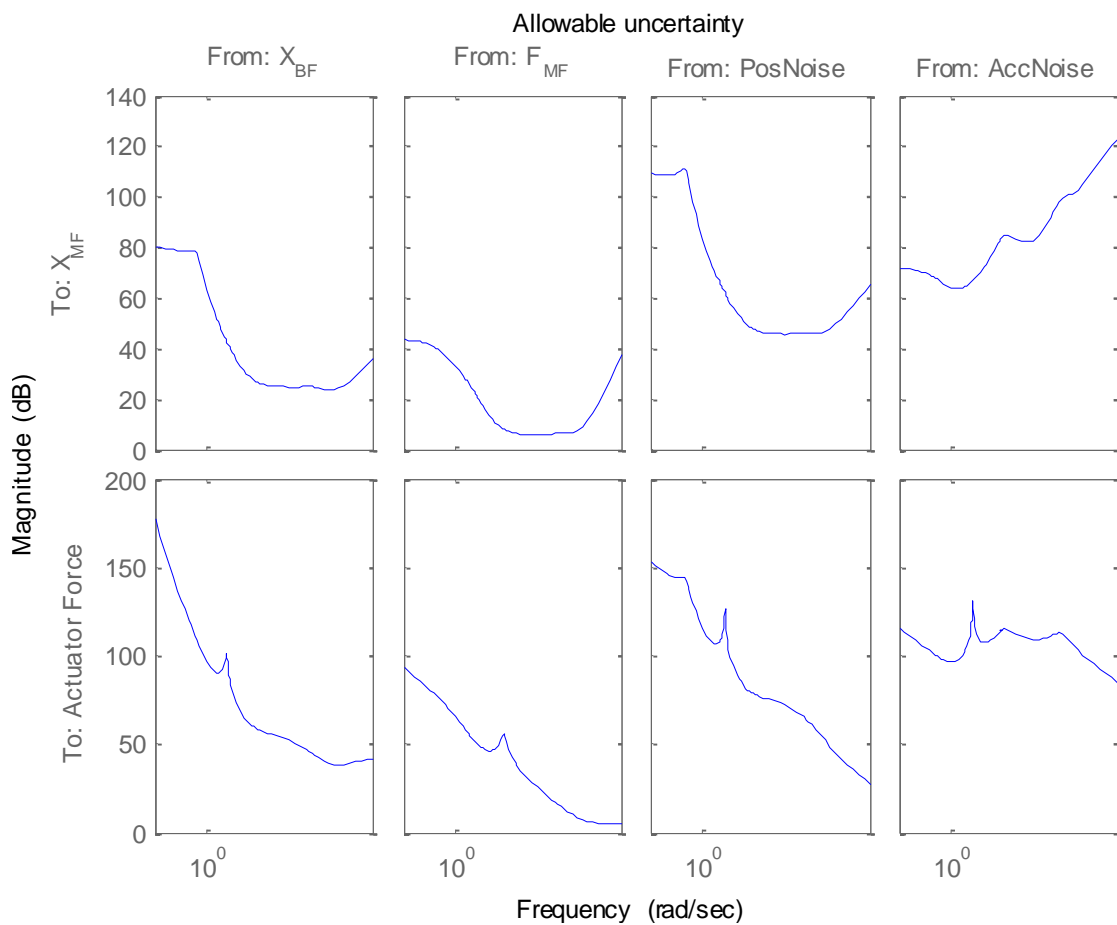


Figure 3-28 – Structured uncertainty limit for each input and output of the system.

The minimum gain for any input to output response at each frequency is the limit of the maximum allowable unstructured uncertainty. This is shown in Figure 3-29. The maximum H-infinity norm for the uncertainty must be lower than $1/\gamma < 1.7852$

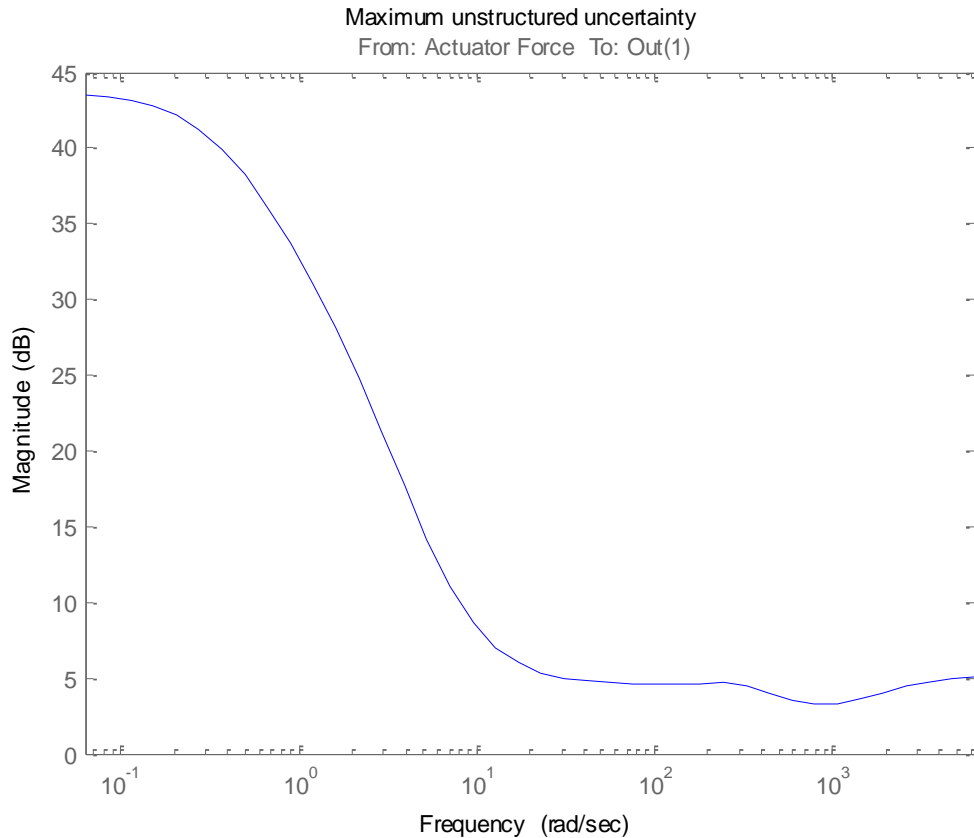


Figure 3-29 - Unstructured uncertainty limit of the system

3.4.3 Conclusions

The results on transmissibility and compliance are worse with this method compared to the classic loopshaping method. Even the passive system delivers better results for transmissibility than this controller. However, the tuning process was not so exhaustive, mainly because of the lack of information on the frequency responses of the inputs. Having this information would help to optimize the output. The exact reason is that without more precise information the disturbances are assumed to be very broadband and therefore ask for control power at high frequencies affecting therefore the output in the band of interest.

3.5 Method 3: Shaping with desired compliance and transmissibility design

In this approach the idea is to minimize the difference between the transmissibility and compliance of the plant and a shape in frequency of transmissibility and compliance. The idea of using the desired transfer function came from (Ding, Damen, & Bosch, Robust Vibration Isolation by Frequency-Shaped Sliding Surface Control with Floor Velocity/Acceleration Measurement, 2012) where desired transmissibility and compliance are used in the sliding surface design. The advantage of this approach at first sight is the ease with which you would be able to introduce desired behaviours to the machine. The configuration of this augmented

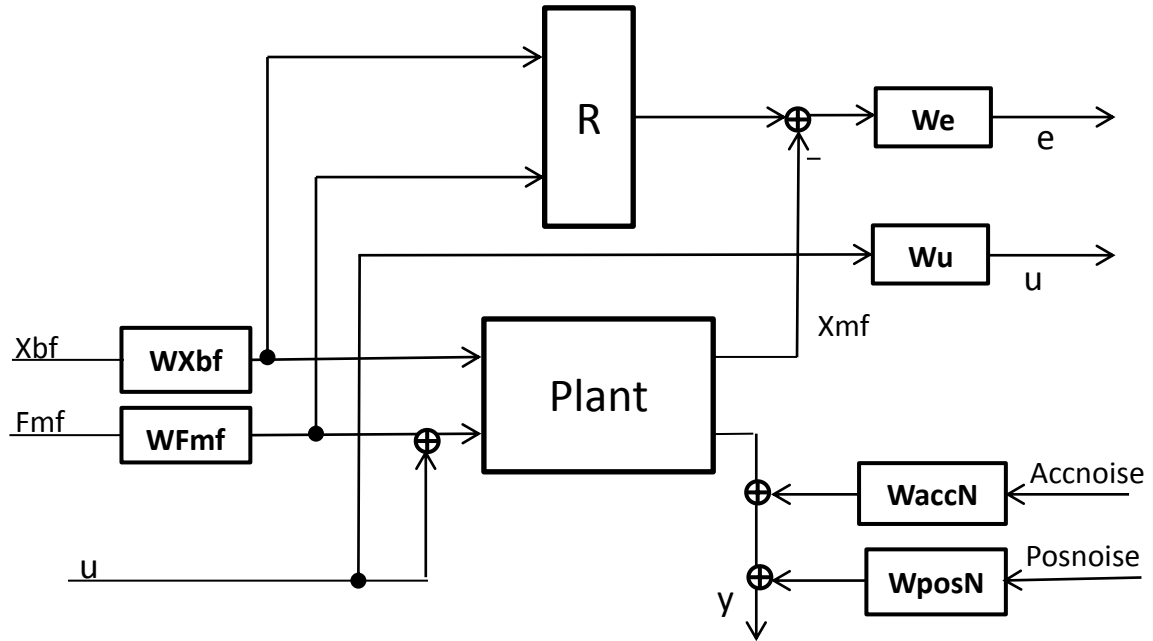


Figure 3-30 - Augmented plant for the shaping desired transfers method.

plant is shown below.

There are only two outputs that are going to be minimized. The controllers output “u” for robustness and an error value “e”. In our configuration “e” belongs to the error between the metroframe’s position coming from the plant and the same variable processed by the filters simulating the desired transmissibility and compliance. The different actions that modifying the different weightings produce are the following.

- **We:** Filters the output on the error between desired and actual transmissibility and compliance. This is the only performance output so this weight will be designed on performance.
- **Wu:** Affects the magnitude of the controllers. . Adding gain to this weight reduces the control action. Helps to set boundaries in frequency to limit the action of the controllers in regions of less interest. i.e. High frequencies for undesired dynamics.
- **WXbf and WFmf:** Related directly to the transmissibility and compliance. These weights will be shaped to represent the actual behaviour of the inputs Baseframe movement and force to the metroframe.
- **WposN and WaccN:** They will describe the real behaviour of the noises of the sensors and reduce the controllers output where the noise is bigger in magnitude.

3.5.1 Desired transmissibility and compliance

The desired transmissibility and compliance can be seen in Figure 3-31.

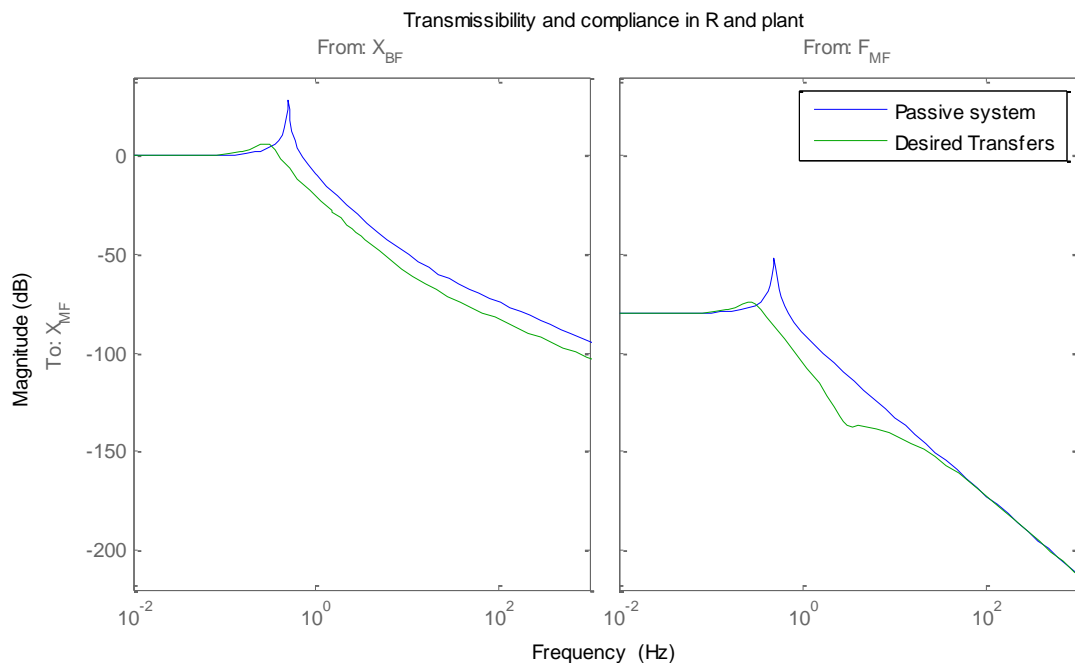


Figure 3-31 - Desired transmissibility and compliance

For both transmissibility and compliance the resonant peak is shifted from 0.5Hz to 0.3Hz.

For transmissibility the desired plant differs from the passive system till infinite high frequencies, this would mean to have a controller that has loop gain until infinitely high frequencies. This is not a reasonable assumption as the control for high frequencies has to be avoided. However, it is important to realize that these desired transmissibilities and compliances are just demands for the controller and that they will be approximated by the controller's action within its limits and not necessarily followed precisely.

The demands for compliance are: preserved stiffness represented by the horizontal line at low frequencies and reduced effect of the force for the frequencies above the resonance peak.

3.5.2 Weights used

Output weights:

- $\mathbf{W}_e: \begin{cases} \text{Gain}(g) = 10^{-3.7} \\ \text{Zero Frequencies}(f_i) = \text{double } 0.5\text{Hz} \\ \text{Pole Frequencies}(f_j) = 12.5\text{Hz and } 100\text{Hz} \end{cases}$

The reason for adding this growing slope for the output error is represented in Figure 3-32. The difference between the desired transmissibility and compliance may seem equal in decibels, but is not in the absolute scale. The H-infinity norm is calculated for the absolute values. Therefore this weight is added to compensate for the natural

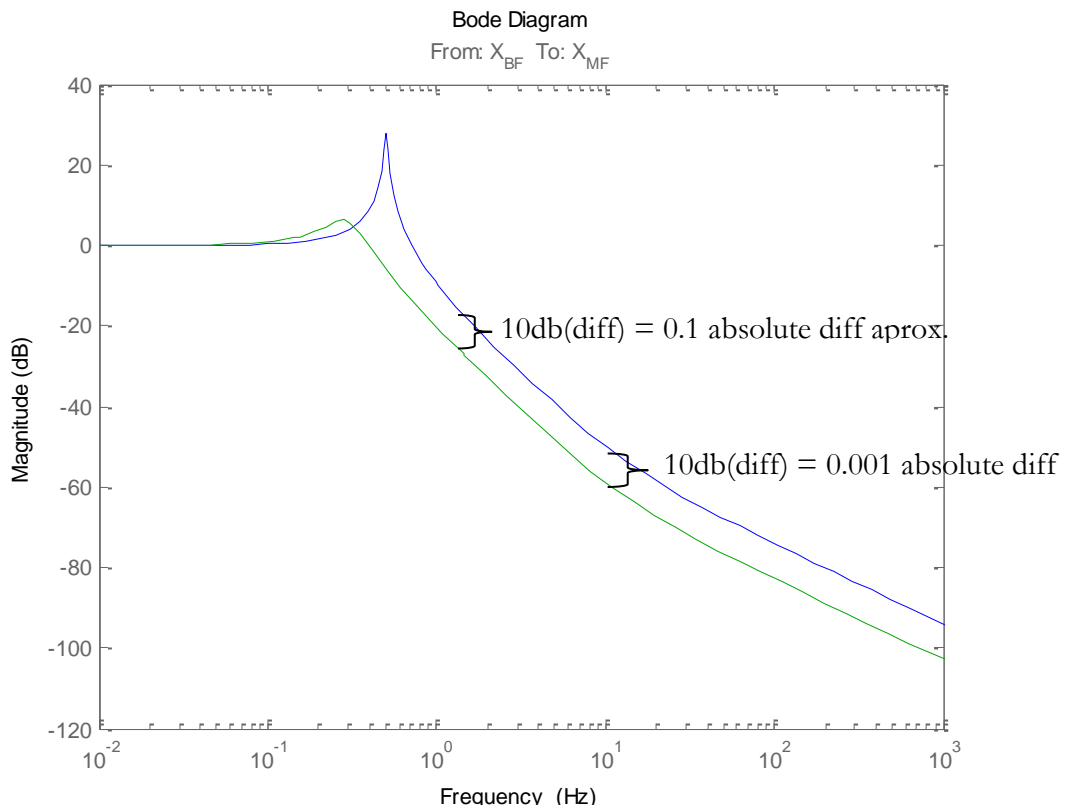


Figure 3-32 - Absolute difference versus logarithmic difference in bode plot.

decreasing slope of the system and so evening the difference from 0.5Hz up to 100Hz that is where our region of interest lies.

- $\mathbf{W}_u: \begin{cases} \text{Gain}(g) = 10^{-14} \\ \text{Zero Frequencies}(f_i) = \text{double } 0\text{Hz} \\ \text{Pole Frequencies}(f_j) = \text{double } 250\text{Hz} \end{cases}$

This weight will be a second order high pass filter that will deal with unwanted control action at high frequencies. The cut off frequency will be at 250 Hz. The gain is chosen by tuning.

The input weights:

- \mathbf{W}_{Xbf} and \mathbf{W}_{Fmf} : $\{ \text{Gain}(g) = 10^{-5} \text{ and } 10 \text{ respectively} \}$
Both these weights are just gains that relate to the inputs magnitudes 10 micrometers for the baseframes position and 10 Newton for the Force to the metroframe.
- \mathbf{W}_{posN} : $\{ \text{Gain}(g) = 10^{-6} \}$
Just a gain with the idea to replicate white noise.

- $\mathbf{WaccN} : \begin{cases} \text{Gain}(g) = 10^{-3} \\ \text{Zero Frequencies}(f_i) = 5e^{-4}\text{Hz}, 5e^{-2}\text{Hz}, 5\text{Hz} \\ \text{Pole Frequencies}(f_j) = 5e^{-5}\text{Hz}, 5e^{-3}\text{Hz}, 5e^{-1}\text{Hz} \end{cases}$

The noise of the accelerometer is a $1/\sqrt{f}$ ramp that starts at 5Hz.

3.5.3 Results

The following figures show the results obtained with this method. Figure 3-33 shows the obtained transmissibility and compliance in comparison with the passive and the desired plots. The controlled response shows a good approximation to the desired transfers. For compliance the obtained result almost mimics the desired plant's behaviour. For transmissibility the response is not exactly the same as the desired response but it is still a big improvement. The resonant peak has been shifted and damped correctly while there is still a reduction in the

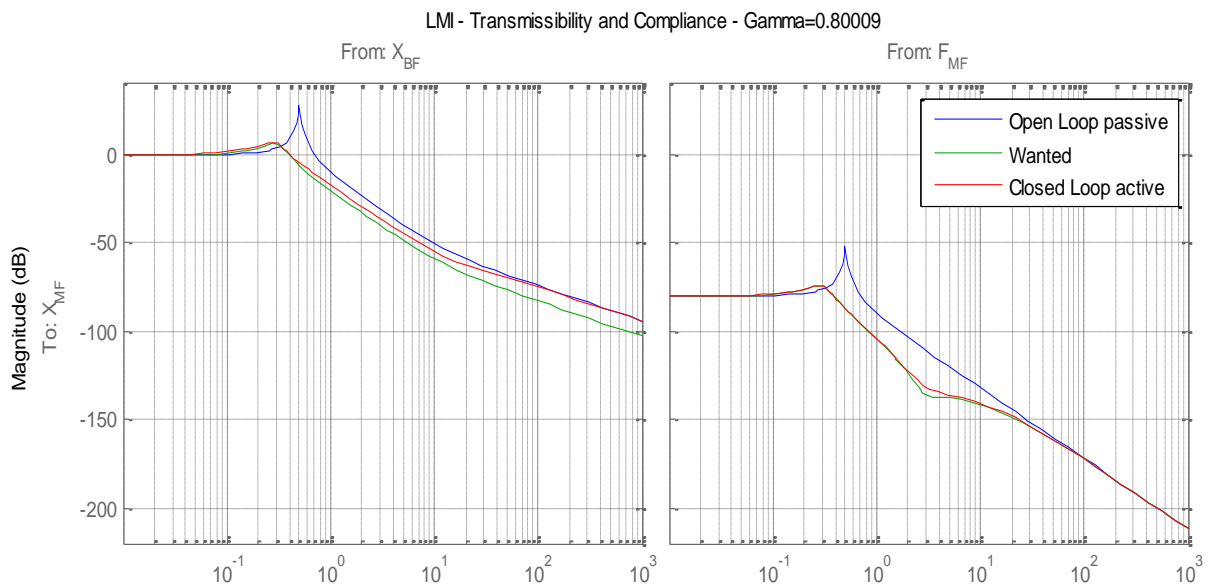


Figure 3-33 - Transmissibility and Compliance from the desired plant approach

magnitude for frequencies up to 100Hz.

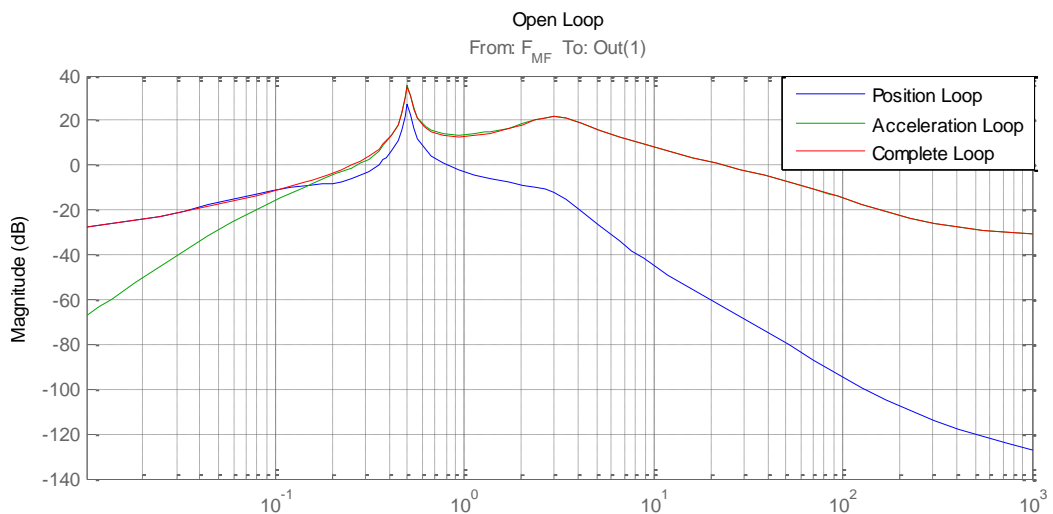


Figure 3-34 - Open loop for the desired transfers approach.

Figure 3-34 contains the open loop response of the controller plus the plant. The different superposed plots are: the response of the position loop, the response of the acceleration loop and the response of the full controller with both loops acting together. The figure shows among other things how for low frequencies the main control power is delivered through the position loop and how this shifts to the acceleration loop for higher frequencies. This shift is explained in the following way: The noise in the accelerometer grows towards low frequencies. Therefore the controller reduces the gain for the acceleration loop and makes use of the gain in the position controller. As this noise decreases, the preferred loop will be that of the absolute values. In addition, adding gain to the position loop for higher frequencies would affect the transmissibility making the controller reduce the gain for this loop and leaving the controller only dependent on the acceleration loop for the high frequencies.

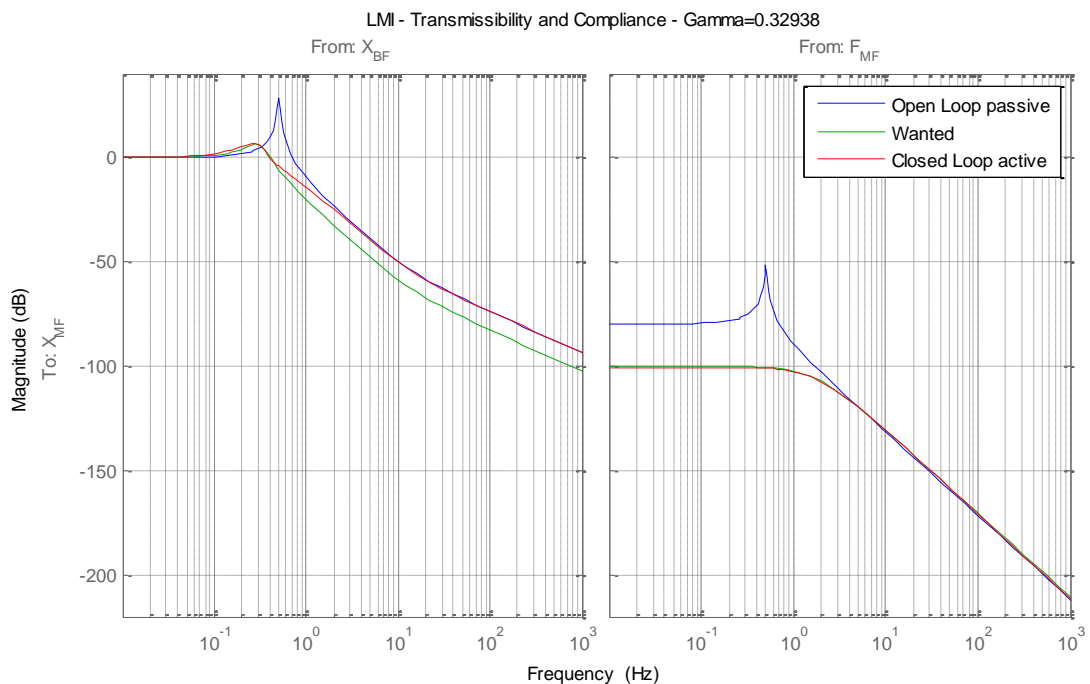


Figure 3-35 - Transmissibility and compliance with alternative desired transfers

The Figure 3-35 shows a test made with different desired transfer functions to prove consistency in the results. The transmissibility has been kept the same, and the compliance has been changed to ask for a higher stiffness and damping of the resonance peak.

It's visible that the results are similar to the previous one, with the closed loop approximating the desired transfers even though the demands have changed so much.

Robust stability analysis

The controlled system with the controller obtained by this method is robustly stable if the uncertainties comply with the following boundaries. The limit for any of the possible structured uncertainties should be.

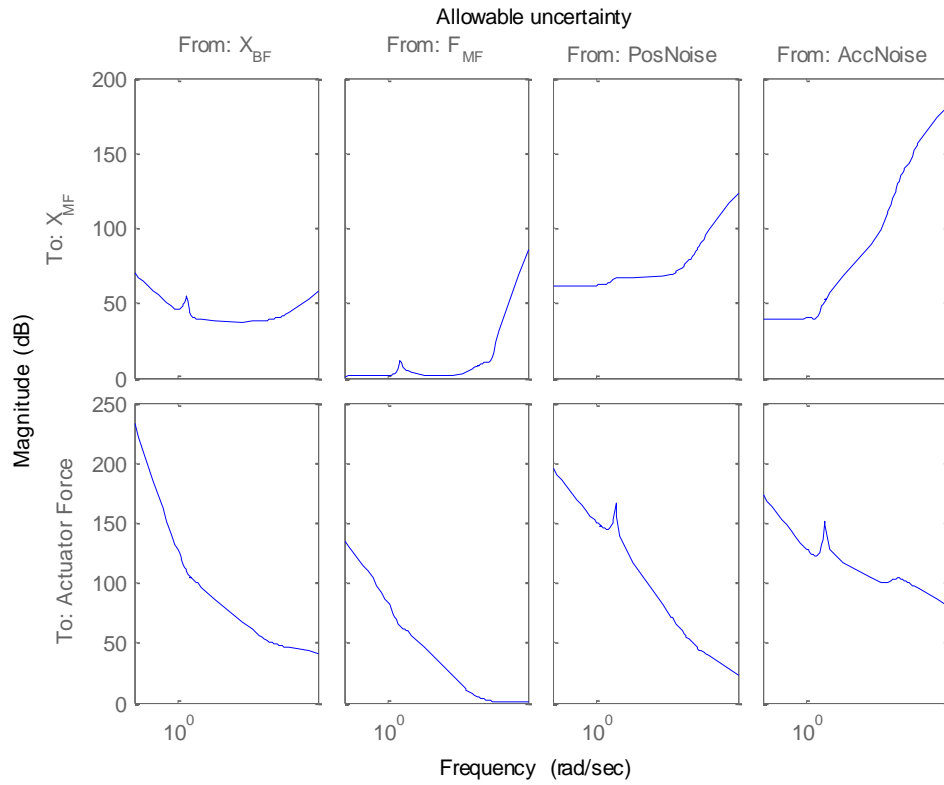


Figure 3-36 - Structured uncertainty limit for each input and output of the system.

For the case of an unstructured uncertainty the uncertainty should be lower in magnitude than the limit shown in . The maximum H-infinity norm for the uncertainty must be lower than $1/\gamma < 1.249$

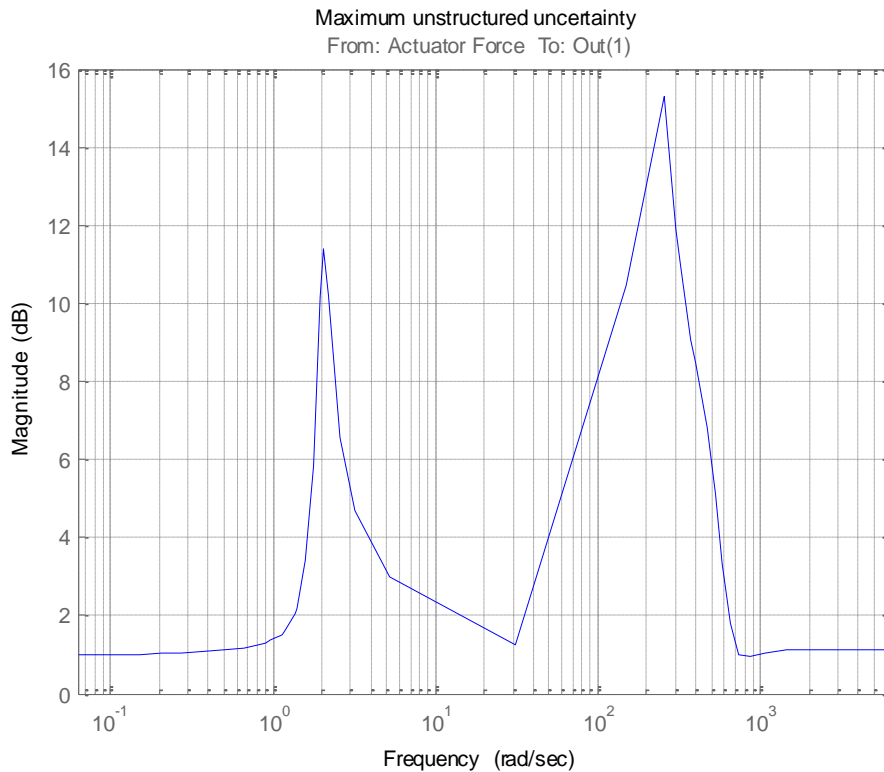


Figure 3-37 - Unstructure uncertainty limit for the controlled system.

3.5.4 Conclusions

This method allows us to easily shape the transmissibility and compliance of our system. The results are comparable to the classical loopshaping method. And the tuning process is very easy compared to the previous h-infinity method. The need of previous information on the plants behaviour is not so high and this affects the tuning process making it far more manageable.

One drawback on this method is that the optimality advantages of H-infinity are lost. The fact of predesigning the transfer functions creates pre-established limitations to the isolation capabilities of the system, more or less in the same fashion as with the classical methods.

3.6 Conclusions

The new method created for controller design has showed us that it is possible to adapt the optimal H-infinity method to become easier to tune without any loss in performance in comparison to the classical loopshaping method. This method also gives the great advantage of being scalable to MIMO while the classical method isn't.

Compared to the straightforward approach of the H-infinity method the new method needs a lot less previous information on the behaviour of the plant leaving it more flexible to adapt to unknown environments. But in both cases the results could improve a lot if more information was provided, information on the frequency spectrums of the inputs or on known uncertainties.

4 Real time implementation

In order to take the controllers out of their simulated environments a test setup is created. This test setup will serve to see the behaviour of the designed controllers in the physical machine. These tests will also provide information on how to improve the design techniques to better fit the real environments.

4.1 Test setup

4.1.1 Hardware

There are three airmounts sustaining the metroframe's mass as shown in section 2.2. Each airmount has a pneumatic system and two Lorentz actuators. The pneumatic system acts in the vertical direction only while the Lorentz actuators are set one in the vertical direction and the other in the horizontal direction. Positions and directions of the horizontal actuators on the metroframe are shown in the schematic below (Figure 4-1).

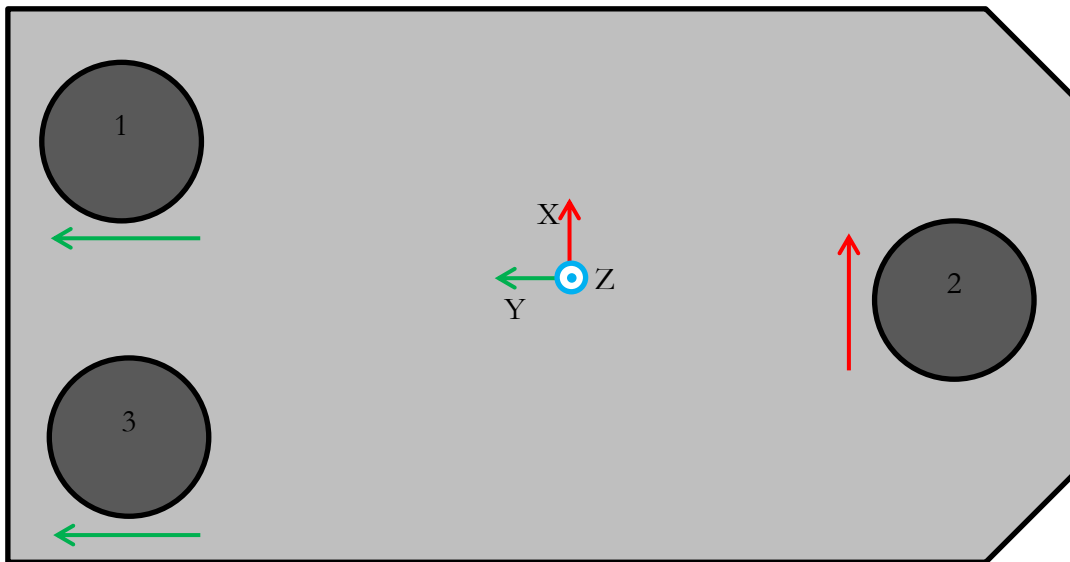


Figure 4-1 –Directions of Lorentz actuators and position sensors in the metroframe in the horizontal plane.

The position sensors are also placed in each airmount. There are 6 sensors, three vertical position sensors to measure the z direction in each airmount and three horizontal position sensors one in each airmount and in the same direction as the actuators, as shown in Figure 4-1. The data from these sensors is processed and transformed to obtain the position coordinates around the center of gravity. The acceleration sensor will be added externally and will be set in the center of gravity of the machine. The accelerometer used is the PCB393B05.

To test the controllers it is necessary to have a means to connect the pc with the sensors and actuators. For this purpose a dSpace system will be used. dSpace is a technology that enables a connection between real machines directly to a Simulink model. In the following sections the test setup in hardware and software will be explained.

The hardware is divided into four main parts

- The machine with its sensors and actuators.
- The electric circuitry known as Mbac-Dicr
- The dSpace system that controls the machine
- The PC, to program and control the dSpace system and to retrieve test data.

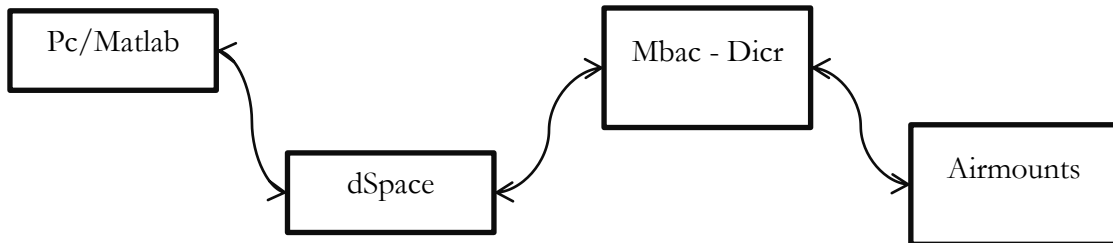


Figure 4-2 - Connections within the test setup

The Mbac-Dicr is the circuitry that connects the dSpace system to the machines components (sensors, actuators, pneumatic valves, etc...). It is responsible for adapting the signals from and to dSpace of the sensors and actuators. It is also responsible for powering the different components. In the case of the accelerometer the link between the sensor and dSpace is made independently from the mbac through the signal conditioner PCB480C02.

The system that will be used is a modular dSpace system consisting of a processor board (ds1005) and an I/O board with two modules one for inputs (ds2002) and another for outputs (ds2102). The way dSpace systems work is the following:

1. Create a model in Simulink using dSpace blocks that model the inputs and outputs of the I/O boards.
2. Import the model to the dSpace processor board and let it run in real time with the machine

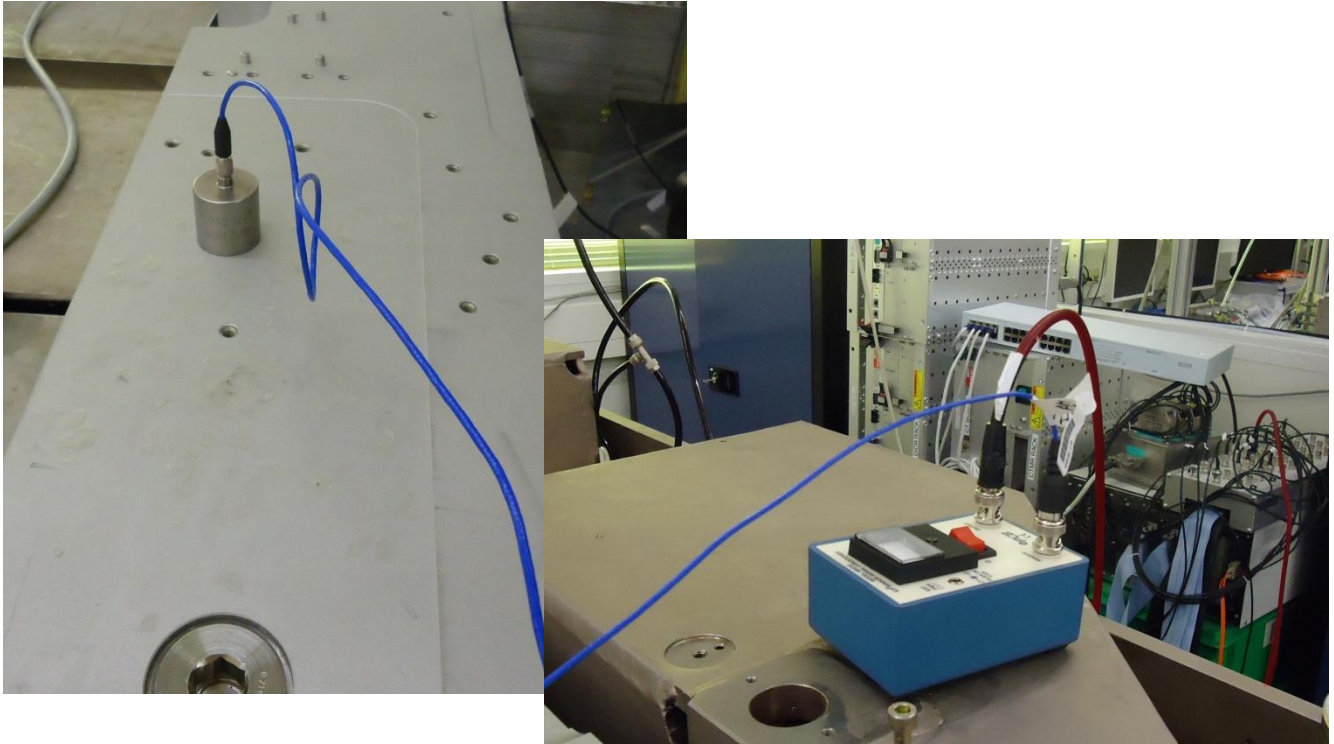


Figure 4-3 - Accelerometer and signal conditioner

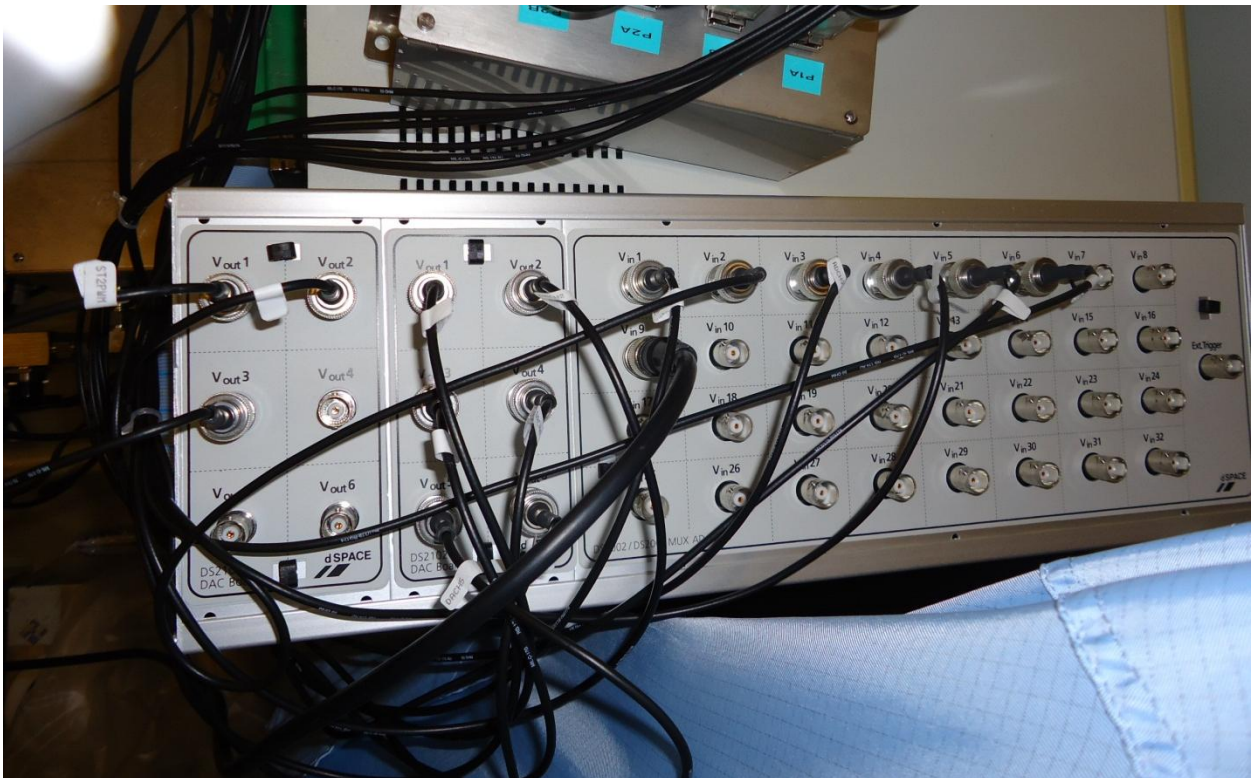


Figure 4-4 - Input and output board of the dSpace system

The dSpace system is the link between the hardware and the software so from the point the signals reach the dSpace system it all runs in software.

4.1.2 Software

In order to test the designed controllers a simulink model is created to close the loop between the sensors and the machine. The model consists of the following parts.

- Controller module
- Pneumatic controllers
- Lorentz controllers

Controller module

The controller module initializes the system. This means lifting the mass from the ground and

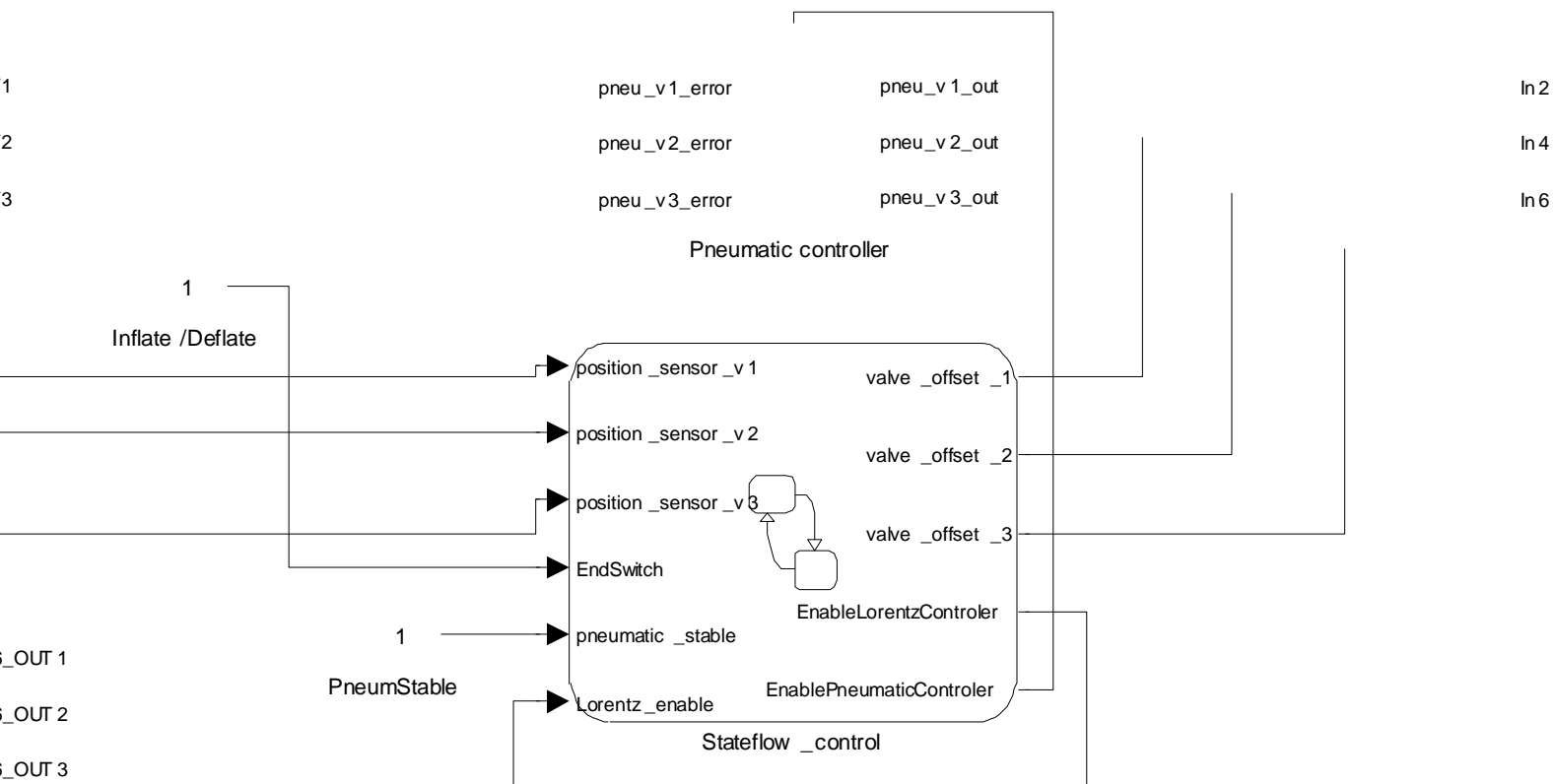


Figure 4-5 - Stateflow chart for the initialization of the system

Setpoint _x	error_x	ctrl_in_x	ctrl_out_x
Measured _x			
Setpoint _y	error_y	ctrl_in_y	ctrl_out_y
Measured _y			
Setpoint _rz	error_rz	ctrl_in_z	ctrl_out_z
Measured _rz			
Setpoint _z	error_z	ctrl_in_rx	ctrl_out_rx
Measured _z		ctrl_in_ry	ctrl_out_ry
Setpoint _rx			
Measured _rx	error_rx		0
Setpoint _ry		1	Constant
Measured _ry	error_ry	Control_Out_Enable	Switch
error_measure logical			

The three controllers working for the pneumatic system work in parallel. One controller for each airmount 's vertical direction. These controllers are in charge of positioning the mass at the correct height. They operate at very low frequencies leaving the active isolation to the Lorentz actuators. The information they receive is directly the position coming from the sensors in each airmount.

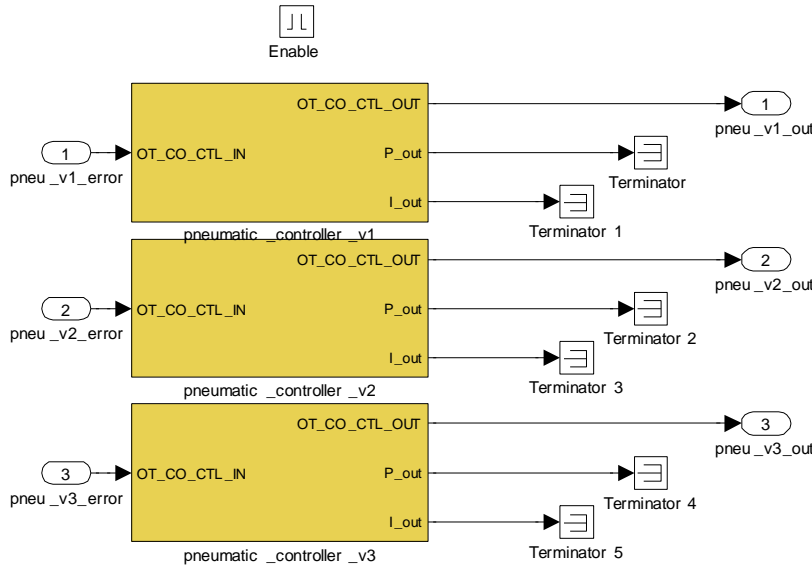


Figure 4-6 - Pneumatic controllers on the Simulink model

From the six controllers for the Lorentz actuators each controller operates on one of the coordinates that the machine can move on, these are the three displacement coordinates (x,y,z) and the three rotational coordinates (χ , ψ , ϕ). In order to use information of the position of the metroframe in its center of gravity a transformation matrix needs to be used to translate the information coming from the position sensors at each airmount to the desired position.

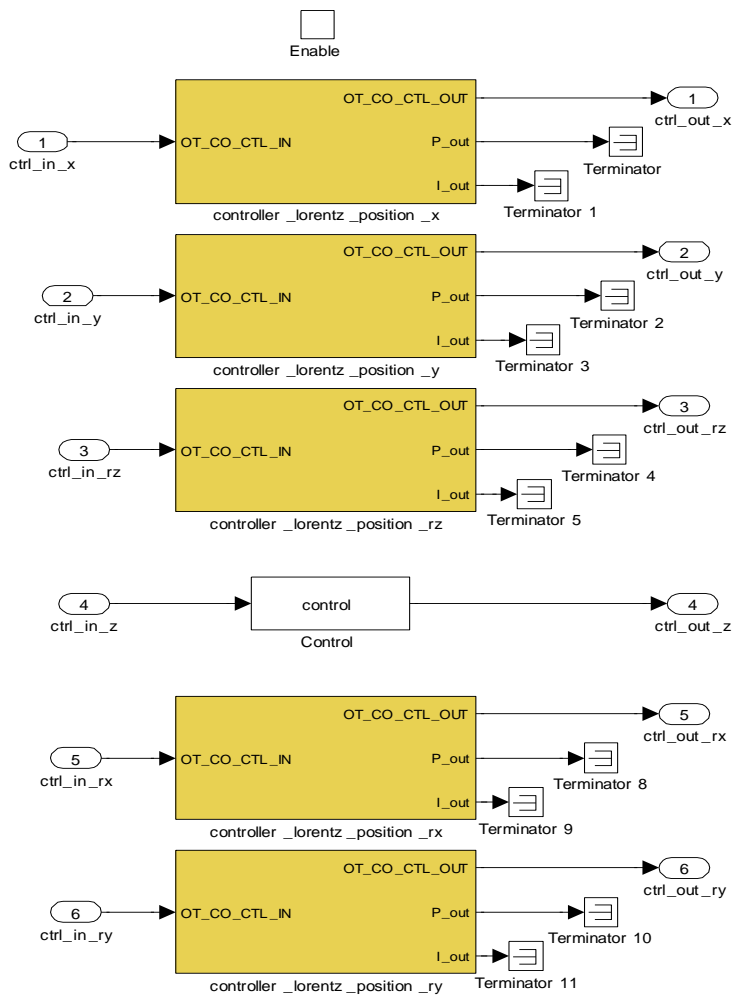


Figure 4-7 - Lorentz controller blocks on the Simulink model

4.2 Plant identification

To study the effect of the controllers on the machine it is necessary to identify the plants frequency response. In the created test setup it is not possible to obtain directly an open loop response for the plant. This is because the metroframe without the control loop is too unsteady to give reliable results. Therefore open loop of the plant has to be obtained by measuring different transfer functions from the closed loop system and calculating the open loop from them.

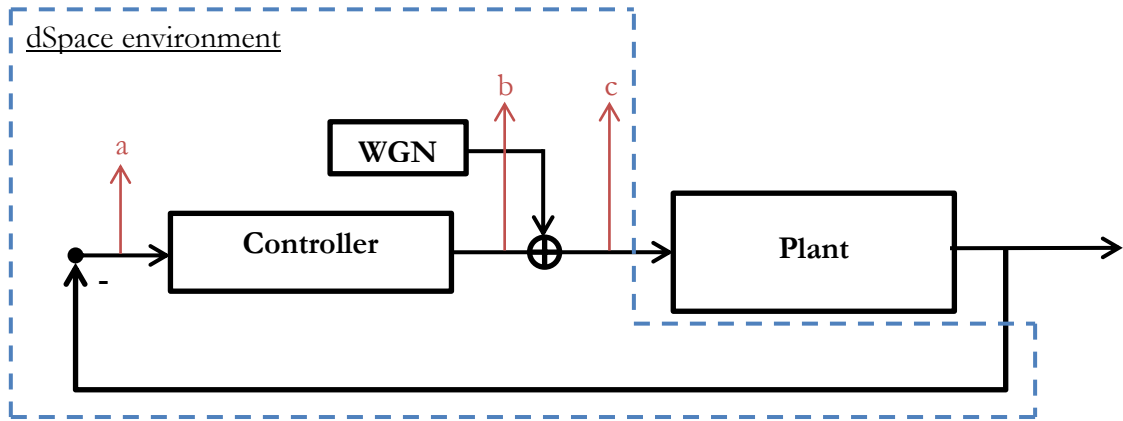


Figure 4-8 – Gaussian noise added to the closed loop within the dspace environment

To obtain these closed loop transfer functions, white Gaussian noise will be added to the loop as shown in the schematic in Figure 4-8. The transfer functions that can be obtained are the frequency responses from the input (wgn) to the three outputs shown in the schematic (a,b,c). The transfer functions obtained are the following:

$$\frac{a}{n} = \frac{-P}{1 + P \cdot C} = \text{PS (process sensitivity)}$$

$$\frac{b}{n} = \frac{-P \cdot C}{1 + P \cdot C}$$

$$\frac{c}{n} = \frac{1}{1 + P \cdot C} = S \text{ (sensitivity)}$$

*P = plants transfer function; C = controller transfer function; n = noise input.

To obtain P the following can be done

$$-\frac{PS}{S} = -\frac{a}{c} = P$$

The obtained transfer function for the plant is the response from the force of the actuators to the position of the metroframe. This is the compliance of the plant. The transmissibility cannot be obtained because it is not possible to add noise to the floor vibrations.

In the case of the frequency response from the force to the acceleration of the metroframe it is possible to directly read the information from the accelerometer in an open loop. This is because the metroframe is already controlled by the position control loop and the accelerometer is set independent from the loop.

To obtain the plants frequency response a measurement of 15 minutes has been made with a white Gaussian noise of 5N amplitude in the Lorentz actuator.

The process shown in Figure 4-9 has been used to obtain the frequency response. The measured signal is divided into pieces, in this case in 15 parts of 60 seconds each. Each signal is filtered and then its Fourier transform is calculated. Finally the frequency responses of each piece are merged together. The result is a frequency response where the external noises are reduced. This process is used to calculate all the frequency responses needed to calculate the plant (a,b,c and n) as shown before.

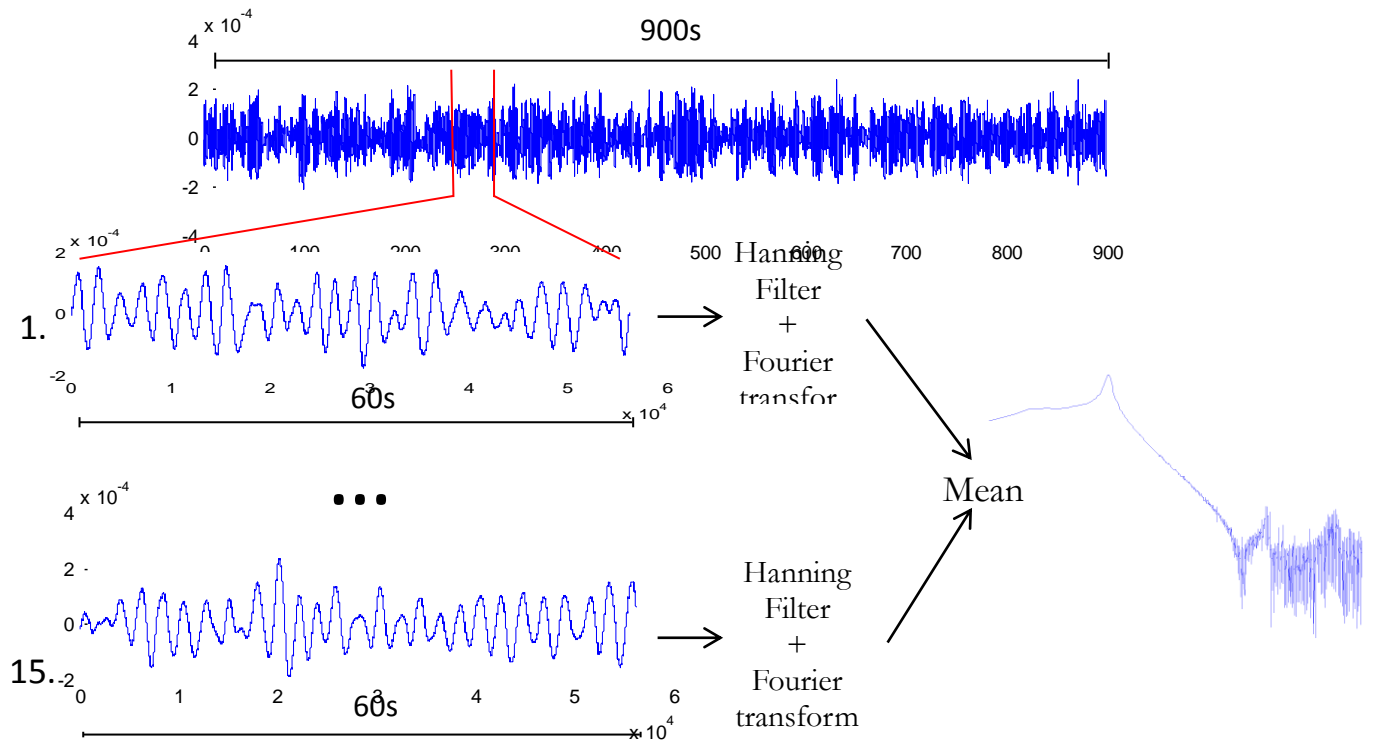


Figure 4-9 - Process used to obtain cleaner frequency responses from measured data

4.2.1 Non-linearity's and limitations in the setup

In the real setup there are some nonlinearity's that were not present in the simulation environment and may affect the output of the plant's identification. These non-linearity's are saturation levels of the actuators, maximum metroframe displacements, sensor errors and sampling rates.

- *Actuator saturation levels*

The saturation levels of the actuators limit the maximum amount of newton of force the actuators can generate. These saturation levels are not only limited by the capacity of the actuators but its limit is set even lower by the software. This is done to avoid excessive force output that could overheat the Lorentz actuators. The saturation levels are set to be $\pm 10\text{N}$ for the vertical actuators of each airmount.

In the simulated environment there was no limitation on the amount of force the controllers could ask the actuators to apply. Thus this will bring problems when applying the created controllers to the real time environment.

- *Metroframe's end stops*

The metroframe has limited movement in all directions. The maximum displacement is around $\pm 2\text{cm}$ around the centre for the vertical direction. This limits the amount of input noise that is used for the identification.

In the simulations the displacement was around the micrometers so this limitation should not affect the controlled system.

- *Sensor errors*

The accelerometer sensor used has an unexpected offset. This offset does not affect the identification. However, this offset although small affects the controllers behaviour. The effects on the output of the controllers from this offset have been tested in the simulated environment. The offset is seen by the controllers as a great acceleration at the start producing a peak as an initial response. After this the output is steady and the offset is no longer a problem.

There are two possible solutions for this problem. One would be to connect the controllers to the sensors without closing the loop until the controllers output is stabilized. And the other would be to calculate the offset and manually eliminate it from the accelerometer's signal by subtraction. Because the offset is steady in between measurements the latter solution has been used.

- *Sampling rates*

In the design controllers have been assumed to be in continuous time. In the real time implementation because there is a need to digitalize the information sampling rates exist. These sampling rates limit the maximum frequencies at which the controllers can operate.

The sampling rate of the machine is 1250 Hz. This frequency lies higher than the bandwidth of interest of our design. However, in the case of the H-infinity methods the controllers have poles and zeros lying beyond this frequency. These high frequency poles and zeros generate unexistent peaks in the frequency responses.

In order to prevent strange behaviours coming from the undersampling of the high frequencies in the machine when run through the controllers, the poles from the controllers with frequencies higher than 100Hz have been eliminated. The results of this reduction are shown in Figure 4-10.

This reduction brings no changes to the magnitude or the phase of the frequency response for frequencies lower than 100 Hz.

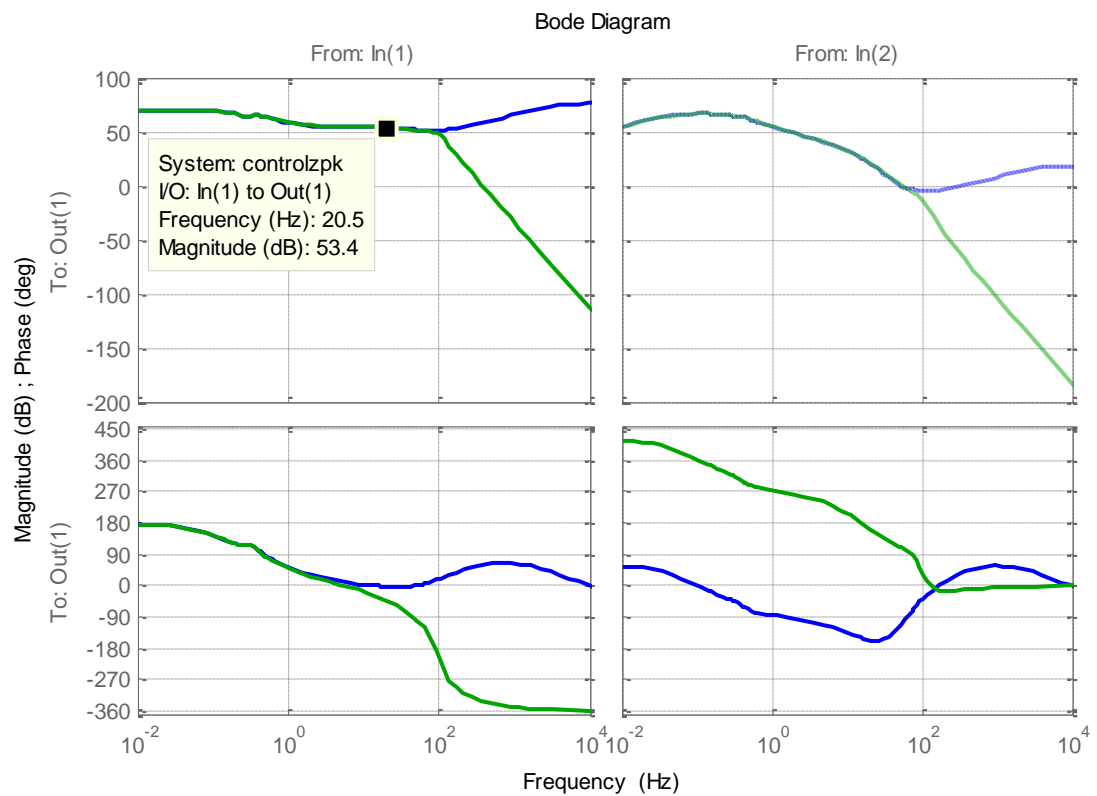


Figure 4-10 - Results from high frequency pole elimination in H-infinity controllers

4.2.2 Identification results

The magnitude, phase and coherence of the frequency responses are shown in Figure 4-11 and Figure 4-12.

The obtained frequency responses of the plant of the real time machine are shown in. The responses are linear for the frequencies between the 0.1 Hz and 10 Hz. Above and below these frequencies the coherence is reduced for the following reasons.

- For the low frequencies this comes from the size of the samples used to obtain the frequency response. The bigger the data string of the sample the better resolution at lower frequencies you can get. The limit is set by $1/(\text{data length})$ in this case $1/45 = 0.0222$ Hz
- For the higher frequencies this is due to mechanics that produce nonlinear behaviours.
- In the case of the acceleration, the increasing magnitude at low frequencies comes from the noise from the accelerometer.

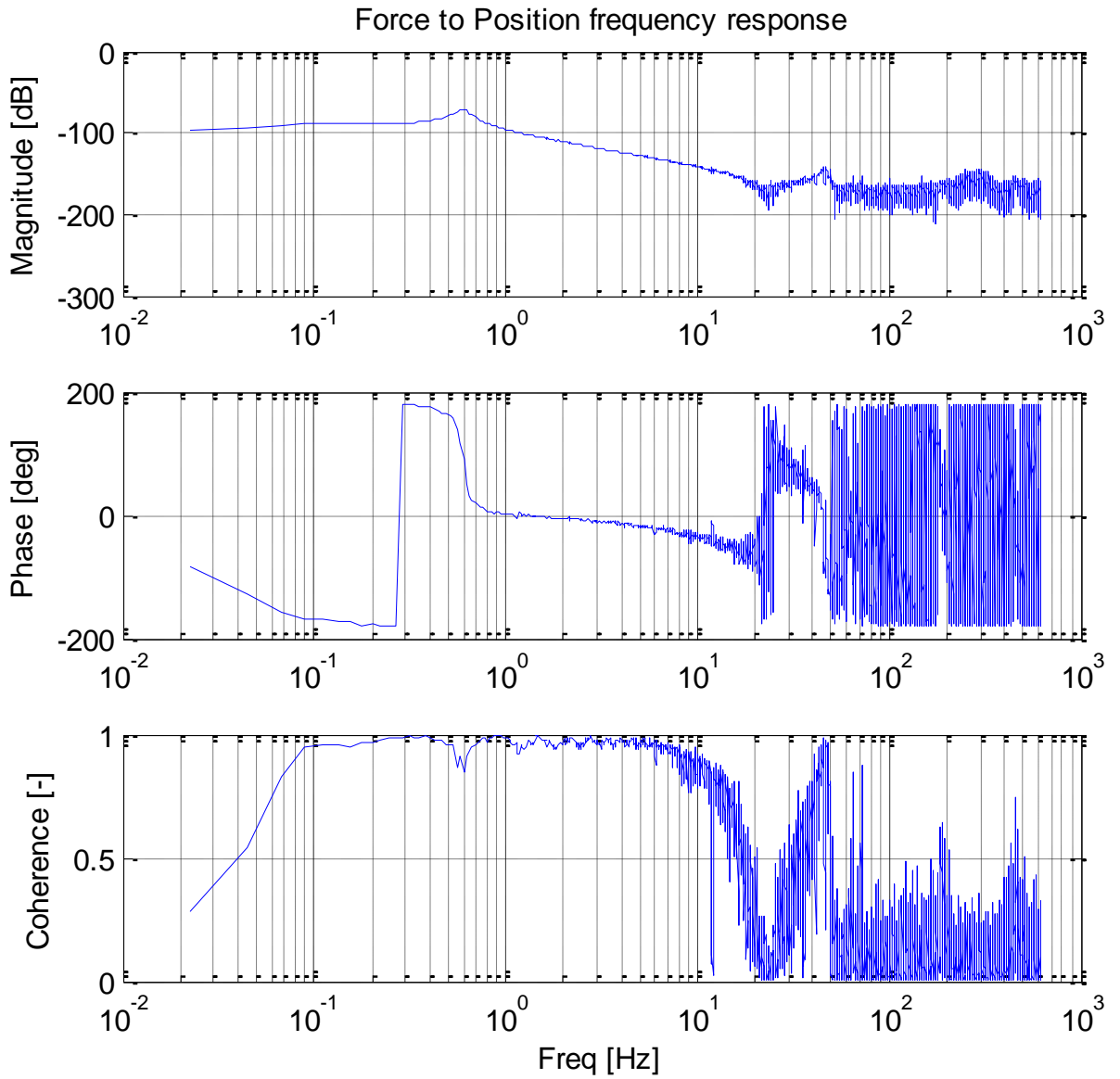


Figure 4-11 - Measured plant frequency response from Force to Position

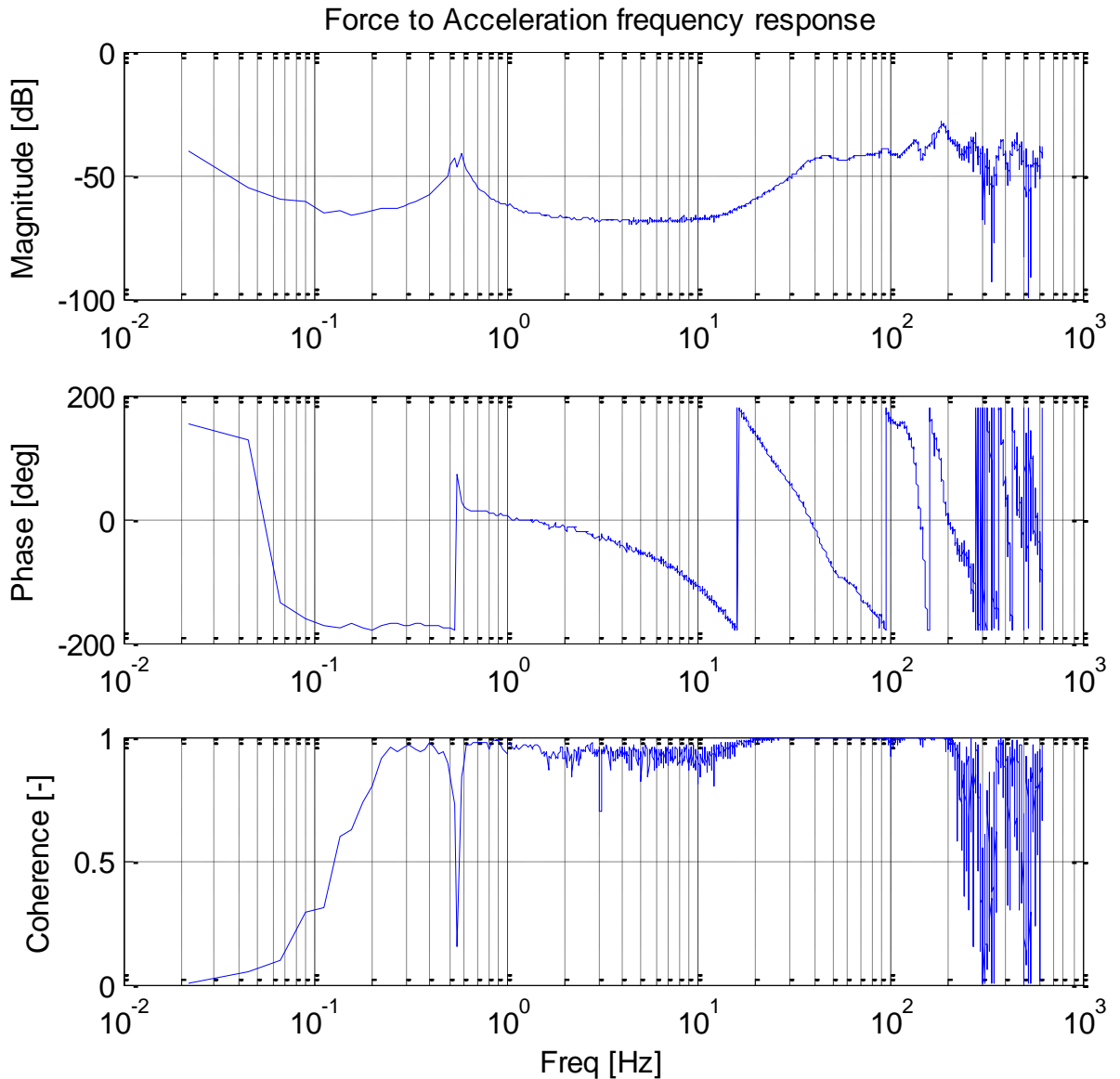


Figure 4-12 - Measured plant frequency response from force to acceleration

It is important to know that the coordinate transformations are not perfect. There is coupled controller action on the z direction coming from the other 5 controllers. This means that the identified plant in the z direction is slightly improved from what the real passive plant should look like. A comparison between the identified plant and the simulated model will help understand the possible modelling errors made and show how significant they may be.

The comparison with the modelled plant is shown in Figure 4-13. The differences between these two plants are the following:

- The resonance peak is a little shifted, from 0.5Hz in the model to 0.6 Hz in the real time plant.
- This shift means that the stiffness in the real time plant is higher than in the simulation. This is also visible in the low frequencies of the compliance (Force to position)

- There is a big mass decoupling at around 40Hz that has not been modelled for the simulation.
- When comparing the phases it is apparent that they are very different. This is problematic for the stability of the controlled system as the phase makes constant shifts of 360 degrees whereas in the simulation it keeps constant.

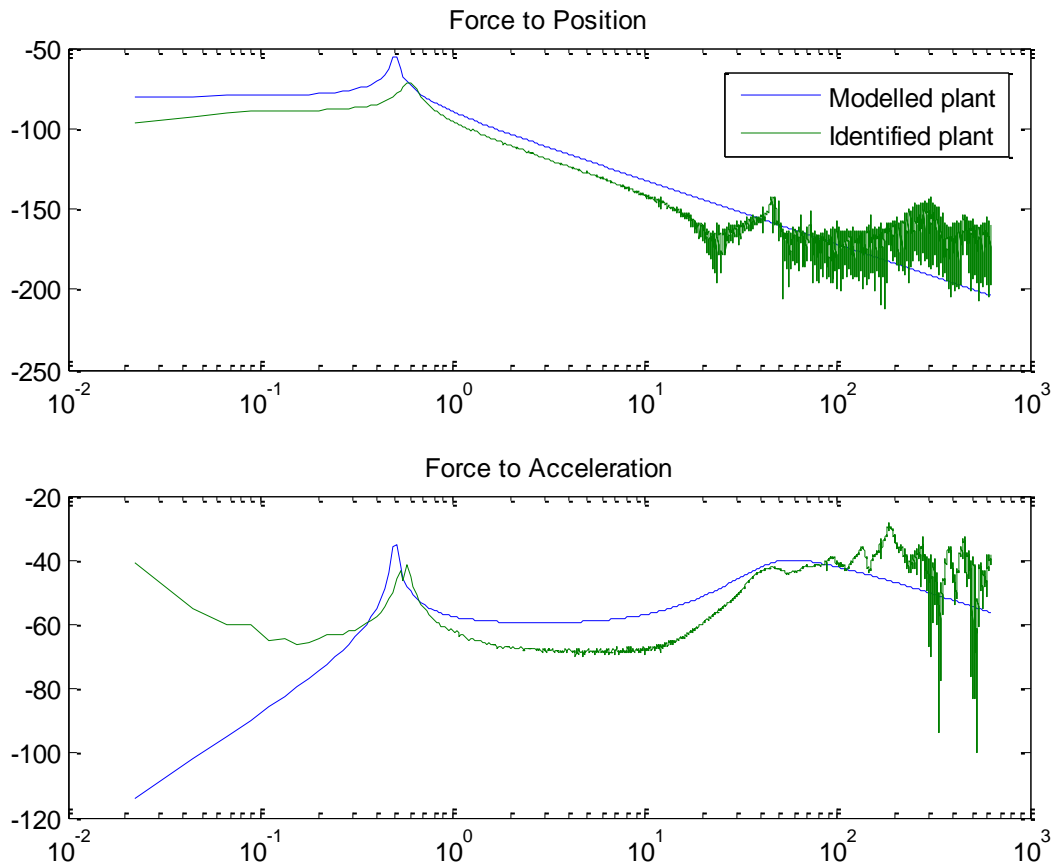


Figure 4-13 - Modelled vs identified plant frequency response magnitude

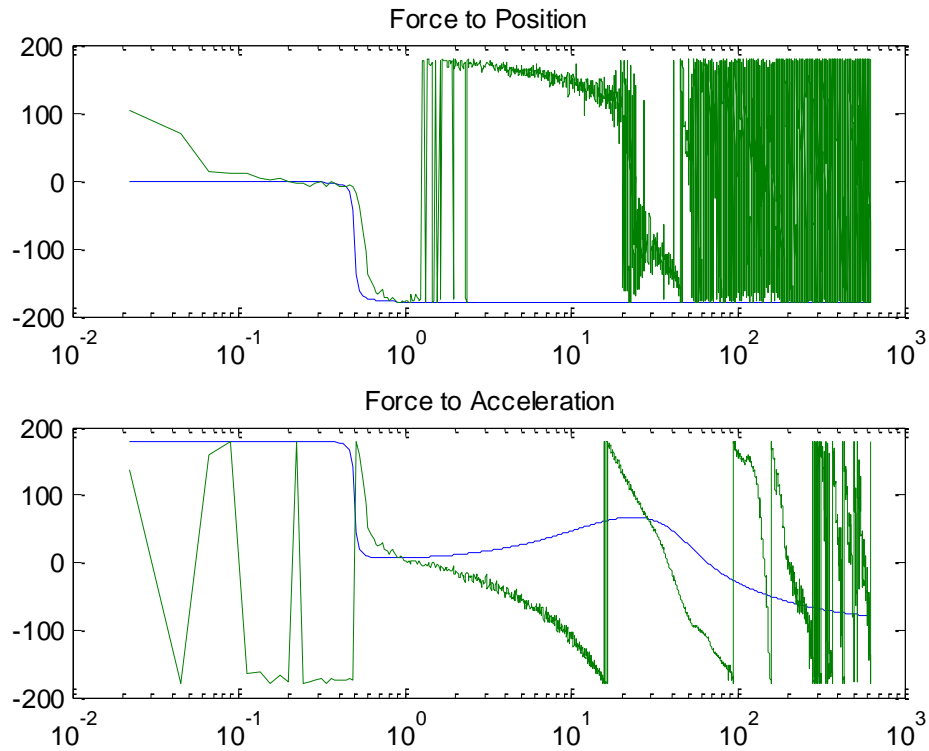


Figure 4-14 - Modelled vs identified plant frequency response phase

4.3 Closed loop

Finally the controllers have been set on the closed loop. The results are the following:

The frequency response of the manually shaped controller is shown in Figure 4-15. The results show that the manually tuned controller does not improve on the open loop response. Except at high frequencies, but the region of interest remains the same.

In the case of the H-infinity methods they both ended up being unstable. This is because the methods are very dependent on the plant's model and the used model is not all that accurate.

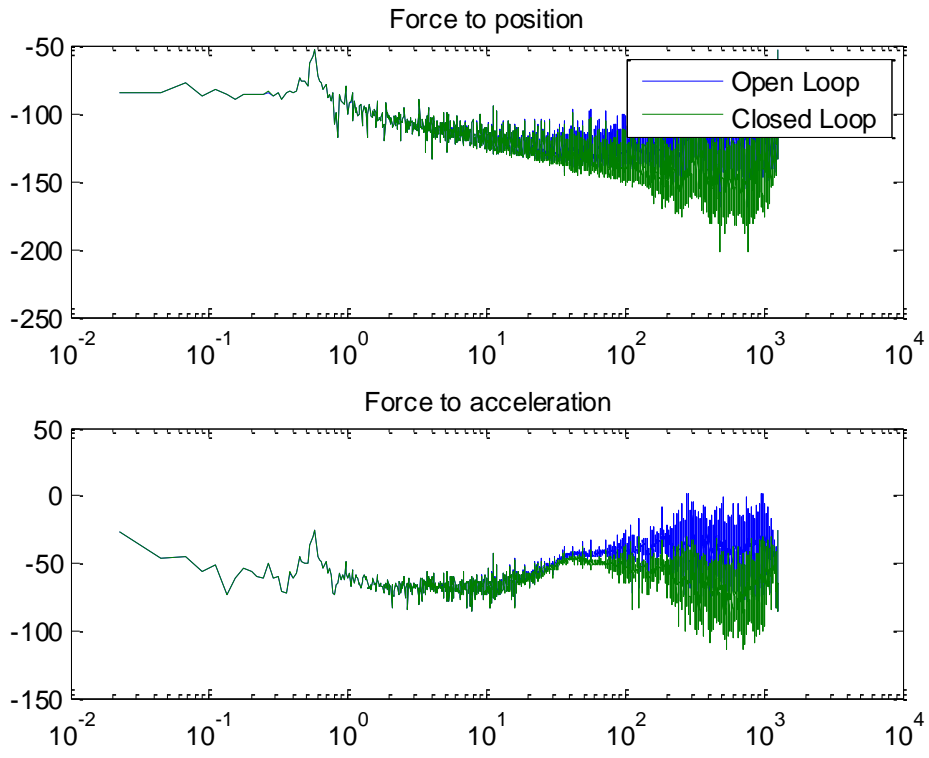


Figure 4-15 - Manually tuned controller closed loop.

5 Conclusiones y trabajo futuro

En la fase de diseño se crearon tres controladores mediante tres métodos distintos diseñados para aumentar el aislamiento de un modelo simplificado a una sola dirección.

El método clásico debido a su naturaleza iterativa es exigente en el tuneado aunque finalmente se obtuvo un buen rendimiento en el aislamiento de vibraciones.

El método de H-infinito básico se centraba en minimizar el movimiento de la masa a aislar. El controlador obtenido mediante este método no mejora el rendimiento del aislamiento. Esto se debe a la reducida cantidad de información sobre el modelo a aislar que se insertó en la planta aumentada. La intención a partir de aquí era la de mejorar este método para reducir la cantidad de estudio previo necesario para los métodos de H-infinito además de aumentar la relación entre el método y las respuestas en frecuencia de la planta a aislar.

El método de H-infinito mejorado crea un controlador bajo la premisa de minimizar la diferencia entre las respuestas en frecuencia del modelo y las respuestas en frecuencia deseadas. El controlador obtenido obtiene un rendimiento parecido al obtenido mediante el método clásico. Lo que se extrae de estos resultados es que el método creado se puede utilizar para generar controladores con la misma confianza que los métodos clásicos con la ventaja de que los métodos H-infinito son escalables a modelos más complejos sin añadir complejidad al tuneado.

Una vez obtenidos los controladores en la simulación se procedió a probarlos en tiempo real sobre un máquina física.

En el entorno real los controladores resultaron ser o inestables o no proporcionaban un aislamiento mejorado. La razón por la que esto ocurre es que la diferencia que existe entre el modelo simulado y la máquina física es mayor que el margen permitido para la incertidumbre.

Para poder comprobar si es posible crear controladores para las máquinas físicas es necesario crear un modelo más preciso. Además en el estudio llevado a cabo no se añadió ninguna información sobre incertidumbre. Para evitar las posibles inestabilidades es necesario estudiar la incertidumbre de la máquina física para poder añadirla al diseño del controlador.

Trabajos futuros

Hay distintas vertientes que se pueden ser la continuación de este proyecto todas basadas en el método de H-infinito basados en distintas características del método. Como son la posibilidad de añadir conocimiento sobre la incertidumbre del modelo o la facilidad con la que se puede escalar a modelos más complejos. También es aprovechable la capacidad de este tipo de modelos de otorgar siempre controladores óptimos para estudiar otros aspectos que afectan de forma paralela al aislamiento activo. Los temas que se podrían tratar son los siguientes.

- Mejorar el método de H-infinito de modelado de respuestas en frecuencia añadiendo conocimiento sobre la incertidumbre visto que esta información es necesaria para la estabilidad de los sistemas controlados.

- Expandir el modelo para diseñar controladores para modelos con seis grados de libertad que se aproximen más a la máquina física
- Es posible estudiar otras limitaciones al aislamiento activo como serían las características de los acelerómetros (niveles de ruido, sensibilidad.)

5 Conclusions and future work

The designing phase showed how it is possible to design a controller using optimal control design methods that provide good isolation results. The obtained results for the optimal controllers did not improve on the loop shaping method. However this is due to the simple approach taken on the H-infinity methods where the information used to characterize the plant was very little with only some characterization on the sensor noise and some performance requirements.

In the test setup the designed controllers were set on a real machine. The designed controllers turned out to be unstable or did not provide increased isolation. The reason for this is that the margins given for divergences between the real plant and the simulation were too small. That means that the controllers were not designed for the real machine and that makes their behaviour unpredictable.

For further research if the controllers in the design phase want to be optimized more information about the inputs and uncertainties of the machine should be studied. The study of the uncertainties of the plant by identifying the plant is very important to avoid obtaining controllers that end up being unstable.

The best way to design a controller through H-infinity methods is to start by making an extensive study on the frequency characteristics of the plant that it is going to be applied to. Of especial importance are obtaining a precise model of the plant as the H-infinity method is so dependent on it and the uncertainty bounds which provide the so much needed robust stability.

Once a good controller has been created for the 1DoF models it can be very interesting to extend the methods to 6DoF models. Using the H-infinity method to create a controller for 6DOF should not be much harder than doing it for a 1DOF. And a single controller created for 6DoF should also provide better results than those existing. This is because the couplings between the coordinates of the machine would also be optimized within the H-infinity algorithms.

6 Works Cited

- A.Joanknecht, H. (n.d.). *Performance optimization of active vibration isolators using modern control techniques*.
- C.Doyle, J., A.Francis, B., & R.Tannenbaum, A. (n.d.). *Feedback control Theory*. Macmillan.
- Damen, A., & Weiland, S. (n.d.). *Robust Control*.
- Ding, C., Damen, A., & Bosch, P. v. (2012). *Robust Vibration Isolation by Frequency-Shaped Sliding Surface Control with Floor Velocity/ Acceleration Measurement*.
- Ding, C., Damen, A., Bosch, P. v., & Janssen, J. (n.d.). *Vibration Isolation of an Electromagnetic Actuator with Passive Gravity Compensation*.
- F.Franklin, G., Powell, J. D., & Emami-Naeini, A. (n.d.). *Feedback Control of Fynamic Systems*. Addison Wesley.
- Meirovithc, L. (2003). *Fundamentals of Vibrations*. McGrawHill.
- Nakashima, K., Tsujino, T., & Fuji, T. (1996). Multivariable Control of a Magnetic Levitation System Using Closed Loop Identificantion and H-infinity Control Theory. *Conference on Decision and Control*. Kobe, Japan.
- Nelson, P. G. (n.d.). (TMC) Retrieved June 2013, from http://www.techmfg.com/techbkgd/techbkgd_5.html#figure17
- Pantazi, A., Sebastian, A., Pozidis, H., & Eleftheriou, E. (2005). Two-sensor-based H-infiniti control for nanopositioning in probe storage. *European Control Conference*. Seville, Spain.
- Peña, R. S., & Sznaier, M. (n.d.). *Robust Systems: Theory and Applications*. Wiley.
- Skogestad, S., & Postelwaith, I. (n.d.). *Multivariable Feedback Control*. Wiley.
- Wal, M. v., & Heertjes, M. (n.d.). *The effect of sensor selection on the achievable performance of vibration isolators*.
- Watanabe, K., & al., e. (1996). Combination of H-infinity en PI Control for an Electromechanically Levitated Vibration Isolation System. *Conference on Decision and Control*. Kobe, Japan.
- Yasuda, M., Osaka, T., & Ikeda, M. (1996). Feedforward Control of a Vibration Isolation System for Disturbance Suppresion. *Conference on Decision and Control*. Kobe, Japan.
- Zhou, K., & Doyle, J. C. (1999). *Essentials of Robust Control*. Prentice Hall.

7 PRESUPUESTO

1) Ejecución Material

- Licencia Matlab..... 2.000 €
- Sistema dSpace (RTI)..... 10000 €
- Ordenador 1500 €
- Componentes (sensores, cables etc...)..... 400 €
- Total de ejecución material 13900 €

2) Gastos generales

- 16 % sobre Ejecución Material..... 2224 €

3) Beneficio Industrial

- 6 % sobre Ejecución Material..... 834 €

4) Honorarios Proyecto

- 1500 horas a 15 € / hora..... 22500 €

5) Material fungible

- Gastos de impresión..... 60 €
- Encuadernación 200 €

6) Subtotal del presupuesto

- Subtotal Presupuesto..... 39718 €

7) I.V.A. aplicable

- 21% Subtotal Presupuesto..... 8340,78 €

8) Total presupuesto

- Total Presupuesto 48058,78 €

Madrid, Septiembre de 2013

El Ingeniero Jefe de Proyecto

Fdo.: Pablo Wildschut

Ingeniero Superior de Telecomunicación

8 PLIEGO DE CONDICIONES

Este documento contiene las condiciones legales que guiarán la realización, en este proyecto, de un método de diseño de controladores. En lo que sigue, se supondrá que el proyecto ha sido encargado por una empresa cliente a una empresa consultora con la finalidad de realizar dicho sistema. Dicha empresa ha debido desarrollar una línea de investigación con objeto de elaborar el proyecto. Esta línea de investigación, junto con el posterior desarrollo de los programas está amparada por las condiciones particulares del siguiente pliego.

Supuesto que la utilización industrial de los métodos recogidos en el presente proyecto ha sido decidida por parte de la empresa cliente o de otras, la obra a realizar se regulará por las siguientes:

8.1 Condiciones generales

1. La modalidad de contratación será el concurso. La adjudicación se hará, por tanto, a la proposición más favorable sin atender exclusivamente al valor económico, dependiendo de las mayores garantías ofrecidas. La empresa que somete el proyecto a concurso se reserva el derecho a declararlo desierto.

2. El montaje y mecanización completa de los equipos que intervengan será realizado totalmente por la empresa licitadora.

3. En la oferta, se hará constar el precio total por el que se compromete a realizar la obra y el tanto por ciento de baja que supone este precio en relación con un importe límite si este se hubiera fijado.

4. La obra se realizará bajo la dirección técnica de un Ingeniero Superior de Telecomunicación, auxiliado por el número de Ingenieros Técnicos y Programadores que se estime preciso para el desarrollo de la misma.

5. Aparte del Ingeniero Director, el contratista tendrá derecho a contratar al resto del personal, pudiendo ceder esta prerrogativa a favor del Ingeniero Director, quien no estará obligado a aceptarla.

6. El contratista tiene derecho a sacar copias a su costa de los planos, pliego de condiciones y presupuestos. El Ingeniero autor del proyecto autorizará con su firma las copias solicitadas por el contratista después de confrontarlas.

7. Se abonará al contratista la obra que realmente ejecute con sujeción al proyecto que sirvió de base para la contratación, a las modificaciones autorizadas por la superioridad o a las órdenes que con arreglo a sus facultades le hayan comunicado por escrito al Ingeniero Director de obras siempre que dicha obra se haya ajustado a los preceptos de los pliegos de condiciones, con arreglo a los cuales, se harán las modificaciones y la valoración de las diversas unidades sin que el importe total pueda exceder de los presupuestos aprobados. Por consiguiente, el número de unidades que se consignan en el proyecto o en el presupuesto, no podrá servirle de fundamento para entablar reclamaciones de ninguna clase, salvo en los casos de rescisión.

8. Tanto en las certificaciones de obras como en la liquidación final, se abonarán los trabajos realizados por el contratista a los precios de ejecución material que figuran en el presupuesto para cada unidad de la obra.

9. Si excepcionalmente se hubiera ejecutado algún trabajo que no se ajustase a las condiciones de la contrata pero que sin embargo es admisible a juicio del Ingeniero Director de obras, se dará conocimiento a la Dirección, proponiendo a la vez la rebaja de precios que el Ingeniero estime justa y si la Dirección resolviera aceptar la obra, quedará el contratista obligado a conformarse con la rebaja acordada.

10. Cuando se juzgue necesario emplear materiales o ejecutar obras que no figuren en el presupuesto de la contrata, se evaluará su importe a los precios asignados a otras obras o materiales análogos si los hubiere y cuando no, se discutirán entre el Ingeniero Director y el contratista, sometidos a la aprobación de la Dirección. Los nuevos precios convenidos por uno u otro procedimiento, se sujetarán siempre al establecido en el punto anterior.

11. Cuando el contratista, con autorización del Ingeniero Director de obras, emplee materiales de calidad más elevada o de mayores dimensiones de lo estipulado en el proyecto, o sustituya una clase de fabricación por otra que tenga asignado mayor precio o ejecute con mayores dimensiones cualquier otra parte de las obras, o en general, introduzca en ellas cualquier modificación que sea beneficiosa a juicio del Ingeniero Director de obras, no tendrá derecho sin embargo, sino a lo que le correspondería si hubiera realizado la obra con estricta sujeción a lo proyectado y contratado.

12. Las cantidades calculadas para obras accesorias, aunque figuren por partidaalzada en el presupuesto final (general), no serán abonadas sino a los precios de la contrata, según las condiciones de la misma y los proyectos particulares que para ellas se formen, o en su defecto, por lo que resulte de su medición final.

13. El contratista queda obligado a abonar al Ingeniero autor del proyecto y director de obras así como a los Ingenieros Técnicos, el importe de sus respectivos honorarios facultativos por formación del proyecto, dirección técnica y administración en su caso, con arreglo a las tarifas y honorarios vigentes.

14. Concluida la ejecución de la obra, será reconocida por el Ingeniero Director que a tal efecto designe la empresa.

15. La garantía definitiva será del 4% del presupuesto y la provisional del 2%.

16. La forma de pago será por certificaciones mensuales de la obra ejecutada, de acuerdo con los precios del presupuesto, deducida la baja si la hubiera.

17. La fecha de comienzo de las obras será a partir de los 15 días naturales del replanteo oficial de las mismas y la definitiva, al año de haber ejecutado la provisional, procediéndose si no existe reclamación alguna, a la reclamación de la fianza.

18. Si el contratista al efectuar el replanteo, observase algún error en el proyecto, deberá comunicarlo en el plazo de quince días al Ingeniero Director de obras, pues transcurrido ese plazo será responsable de la exactitud del proyecto.

19. El contratista está obligado a designar una persona responsable que se entenderá con el Ingeniero Director de obras, o con el delegado que éste designe, para todo relacionado con ella. Al ser el Ingeniero Director de obras el que interpreta el proyecto, el contratista deberá consultarle cualquier duda que surja en su realización.

20. Durante la realización de la obra, se girarán visitas de inspección por personal facultativo de la empresa cliente, para hacer las comprobaciones que se crean oportunas. Es obligación del contratista, la conservación de la obra ya ejecutada hasta la recepción de la misma, por lo que el deterioro parcial o total de ella, aunque sea por agentes atmosféricos u otras causas, deberá ser reparado o reconstruido por su cuenta.

21. El contratista, deberá realizar la obra en el plazo mencionado a partir de la fecha del contrato, incurriendo en multa, por retraso de la ejecución siempre que éste no sea debido a causas de fuerza mayor. A la terminación de la obra, se hará una recepción provisional previo reconocimiento y examen por la dirección técnica, el depositario de efectos, el interventor y el jefe de servicio o un representante, estampando su conformidad el contratista.

22. Hecha la recepción provisional, se certificará al contratista el resto de la obra, reservándose la administración el importe de los gastos de conservación de la misma hasta su recepción definitiva y la fianza durante el tiempo señalado como plazo de garantía. La recepción definitiva se hará en las mismas condiciones que la provisional, extendiéndose el acta correspondiente. El Director Técnico propondrá a la Junta Económica la devolución de la fianza al contratista de acuerdo con las condiciones económicas legales establecidas.

23. Las tarifas para la determinación de honorarios, reguladas por orden de la Presidencia del Gobierno el 19 de Octubre de 1961, se aplicarán sobre el denominado en la actualidad "Presupuesto de Ejecución de Contrata" y anteriormente llamado "Presupuesto de Ejecución Material" que hoy designa otro concepto.

8.2 Condiciones particulares

La empresa consultora, que ha desarrollado el presente proyecto, lo entregará a la empresa cliente bajo las condiciones generales ya formuladas, debiendo añadirse las siguientes condiciones particulares:

1. La propiedad intelectual de los procesos descritos y analizados en el presente trabajo, pertenece por entero a la empresa consultora representada por el Ingeniero Director del Proyecto.

2. La empresa consultora se reserva el derecho a la utilización total o parcial de los resultados de la investigación realizada para desarrollar el siguiente proyecto, bien para su publicación o bien para su uso en trabajos o proyectos posteriores, para la misma empresa cliente o para otra.

3. Cualquier tipo de reproducción aparte de las reseñadas en las condiciones generales, bien sea para uso particular de la empresa cliente, o para cualquier otra aplicación, contará con autorización expresa y por escrito del Ingeniero Director del Proyecto, que actuará en representación de la empresa consultora.

4. En la autorización se ha de hacer constar la aplicación a que se destinan sus reproducciones así como su cantidad.

5. En todas las reproducciones se indicará su procedencia, explicitando el nombre del proyecto, nombre del Ingeniero Director y de la empresa consultora.

6. Si el proyecto pasa la etapa de desarrollo, cualquier modificación que se realice sobre él, deberá ser notificada al Ingeniero Director del Proyecto y a criterio de éste, la empresa consultora decidirá aceptar o no la modificación propuesta.

7. Si la modificación se acepta, la empresa consultora se hará responsable al mismo nivel que el proyecto inicial del que resulta el añadirla.

8. Si la modificación no es aceptada, por el contrario, la empresa consultora declinará toda responsabilidad que se derive de la aplicación o influencia de la misma.

9. Si la empresa cliente decide desarrollar industrialmente uno o varios productos en los que resulte parcial o totalmente aplicable el estudio de este proyecto, deberá comunicarlo a la empresa consultora.

10. La empresa consultora no se responsabiliza de los efectos laterales que se puedan producir en el momento en que se utilice la herramienta objeto del presente proyecto para la realización de otras aplicaciones.

11. La empresa consultora tendrá prioridad respecto a otras en la elaboración de los proyectos auxiliares que fuese necesario desarrollar para dicha aplicación industrial, siempre que no haga explícita renuncia a este hecho. En este caso, deberá autorizar expresamente los proyectos presentados por otros.

12. El Ingeniero Director del presente proyecto, será el responsable de la dirección de la aplicación industrial siempre que la empresa consultora lo estime oportuno. En caso contrario, la persona designada deberá contar con la autorización del mismo, quien delegará en él las responsabilidades que ostente.