Controlling the dynamic characteristics of machining systems through consciously designed joint interfaces

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ABSTRACT

The precision of machining systems is ever increasing in order to keep up with components’ accuracy requirements. At the same time product variants are increasing and order quantities are decreasing, which introduces high demands on the capability of machining systems. The machining system is an interaction between the machine tool structure, the process and the control system and is defined in terms of capability by the positional, static, dynamic and thermal accuracy. So far, the control of the machining system, in terms of static and dynamic stability is process based which is often translated into sub-optimum process parameters and therefore low productivity.

This thesis proposes a new approach for control of the machining system which is based on the capability to control the structural properties of the machine tool and as a result, controlling the outcome of the machining process. The control of the structural properties is realized by carefully designed Joint Interface Modules (JIMS). These modules allow for control of the stiffness and damping of the structure, as a result of tuning the contact conditions on the interface of the JIM; this is performed by control of the pre-load on the interface, by treatment of the interface with damping enhancing materials, or both.

The thesis consists of a presentation of the motivation behind this work, the theoretical basis on which the proposed concept is based and a part describing the experimental investigations carried out. Two prototype JIMs, one for a milling process and one for a turning process were used in the experimental investigations that constitute the case studies for examining the validity of the proposed concept and demonstrating its applicability in a real production environment.

Keywords: Production, machine tool, machining system control, joint interfaces, JIM, stiffness, damping, vibrations, machining process, milling, turning, deflections, accuracy, dynamic response
“Απόψε απ' το σπίτι μου θα βγώ
Κι όλα θα τα τσακίσω
Κλεισμένος έμεινα πολύ καιρό
Κοντεύω να σαπίσω, ναι!”

Κομοδίνα 3 (λογοκριμένο)
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# TABLE OF CONTENTS

1 INTRODUCTION ..................................................................................................................................... 1

1.1 BACKGROUND AND MOTIVATION .......................................................................................... 1

1.2 RESEARCH QUESTIONS .............................................................................................................. 4

1.3 THESIS STRUCTURE .................................................................................................................... 5

2 STIFFNESS AND DAMPING IN MACHINE TOOLS ........................................................................ 7

2.1 STIFFNESS, DAMPING AND THE ROLE OF JOINTS IN MACHINE TOOLS ......................... 7

2.2 CONCLUSIONS ............................................................................................................... ............. 13

3 STATIC AND DYNAMIC INTERACTIONS IN MACHINING SYSTEMS .................................. 15

3.1 STATIC INTERACTIONS IN MACHINING SYSTEMS ................................................................ 15

3.2 DYNAMIC INTERACTIONS IN MACHINING SYSTEMS ............................................................ 18

3.2.1 FREE VIBRATIONS .................................................................................................................. 19

3.2.2 FORCED VIBRATIONS ............................................................................................................ 19

3.2.3 CHATTER ..................................................................................................................................... 19

3.3 PROCESS DAMPING ................................................................................................................... 20

3.4 CONCLUSIONS ............................................................................................................... ............. 22

4 CONTROL STRATEGIES OF MACHINING SYSTEMS ................................................................ 23

4.1 CHATTER DETECTION, IDENTIFICATION AND CONTROL ............................................... 23

4.2 STABILITY LOBE DIAGRAMS .................................................................................................. 24

4.3 PASSIVE TECHNIQUES ............................................................................................................... 27

4.4 ACTIVE TECHNIQUES ............................................................................................................... 28

4.5 CONCLUSIONS ............................................................................................................... ............. 29

5 THE JOINT INTERFACE MODULE CONCEPT ............................................................................. 31

5.1 INTRODUCTION .............................................................................................................. ............ 31

5.2 REQUIREMENTS FOR JOINT INTERFACE MODULES ......................................................... 32

6 MILLING PROCESS CONTROL THROUGH TUNING OF THE JOINTS’ CHARACTERISTICS ....................................................................................................................................... 37

6.1 THE JOINT INTERFACE MODULE (JIM) WORK-HOLDING DEVICE FOR MILLING ...... 37

6.1.1 INTERFACE TREATMENT FOR DAMPING ENHANCEMENT............................................ 38

6.2 MODELING AND SIMULATION OF THE STRUCTURAL CHARACTERISTICS OF THE SYSTEM 39

6.2.1 SIMULATION OF THE UNTREATED CONFIGURATION ................................................... 40

6.2.2 SIMULATION OF THE VEM CONFIGURATION .................................................................. 42

6.3 EXPERIMENTAL INVESTIGATIONS: THE EFFECTS OF PRE-LOADING AND DAMPING ENHANCEMENT ON THE STRUCTURAL CHARACTERISTICS ON THE JIM ............................... 44

6.3.1 EFFECT OF PRE-LOADING ON STRUCTURAL CHARACTERISTICS - UNTREATED CONFIGURATION ................................................................. 44

6.3.2 EFFECT OF PRE-LOADING ON THE STRUCTURAL CHARACTERISTICS – VEM CONFIGURATION ............................................................................ 48

6.3.3 EFFECT OF PRE-LOADING ON THE STRUCTURAL CHARACTERISTICS – COATED CONFIGURATION ....................................................................................... 50
| 6.3.4 | THE EFFECT OF ENHANCING DAMPING ................................................................. 53 |
| 6.4  | RESPONSE OF THE SYSTEM TO MACHINING EXCITATIONS ................................. 55 |
| 6.4.1 | EXPERIMENTAL SETUP ....................................................................................... 56 |
| 6.4.2 | STRUCTURAL CHARACTERISTICS OF THE TOOL USED ..................................... 56 |
| 6.4.3 | RESPONSE OF THE SYSTEM IN THE UNTREATED INTERFACE CONFIGURATION... 58 |
| 6.4.4 | RESPONSE OF THE SYSTEM IN THE VEM CONFIGURATION ................................ 62 |
| 6.4.5 | RESPONSE OF THE SYSTEM IN THE COATED CONFIGURATION .......................... 65 |
| 6.4.6 | CONTROL PARAMETER EFFECTS ON STABLE CUTTING CONDITIONS .............. 68 |
| 6.5  | CHAPTER CONCLUSIONS .................................................................................... 70 |

| 7   | TURNING PROCESS CONTROL THROUGH TUNING OF THE JOINTS’ CHARACTERISTICS .................................................. 75 |
| 7.1  | EXPERIMENTAL SETUP ..................................................................................... 76 |
| 7.2  | MACHINING EXPERIMENTS RESULTS ............................................................... 77 |
| 7.3  | CHAPTER CONCLUSIONS ................................................................................ 82 |

| 8   | APPLICATIONS AND CONTROL STRATEGIES ...................................................... 85 |

| 9   | CONCLUSIONS, DISCUSSION AND FUTURE WORK ............................................. 87 |
| 9.1  | CONCLUSIONS AND DISCUSSION ..................................................................... 87 |
| 9.2  | FUTURE WORK ............................................................................................... 89 |
1 INTRODUCTION

1.1 BACKGROUND AND MOTIVATION

Throughout the years, manufacturing has been a field of continuous change and improvement, always driven by the demand for products that can either fulfill a functionality gap, or provide a competitive advantage to the product users. Quality requirements for the manufactured components have only been increasing, driven by demands for products of high accuracy. To make matters more complicated, accuracy demands are now independent of the scale of the component leading to features of high precision in very large components (an example given in [1]). Figure 1 shows how over the years the precision of machining systems is increasing to keep up with components’ accuracy requirements.

![Figure 1. Progress in machining accuracy [2]](image_url)

The machining system is an interaction between the machine tool structure, the process and the control system [3] and is defined in terms of capability by the positional, static, dynamic and thermal accuracy (Figures 2 and 3). Positional/kinematical, static and thermal deviations of the machining system are ultimately reflected on the machined part in the form of geometric and dimensional errors [4], while compromised dynamic accuracy will result in poor surface roughness and integrity [5]. Therefore, high capability of the machining system in terms of positional/kinematical, thermal, static and dynamic accuracy is required in order to produce parts according to design specifications.
More recently new trends have appeared, especially regarding the reduction of the products’ life cycles and a trend towards mass customization. This means
that the life cycle of products is now significantly shorter than the life cycle of manufacturing equipment, consequently manufacturing systems have to cope with new product variants more often [10], [11]. At the same time mass customization increases even more the amount of product variants that a manufacturing system has to be able to produce. It is obvious from the above that an economically viable manufacturing system will have to be able to deliver outputs that are to an extent unpredictable in the beginning of the production equipment’s lifecycle (i.e. acquisition of the machine tools) but nevertheless with high accuracy demands. Needless to say, this is largely reflected to demands on the capability of the production equipment (and for the scope of this thesis, the machining system) to handle manufacturing variations. These “external” sources of variability are added to “ordinary” variations machining systems face, which can influence the outcome of the process due to various sources such as work materials, tool/work clamping, production environment, machine tool structure, deviations from the designed process plan etc.

All these factors that can challenge the quality of the final product have to be addressed with a minimum burden on productivity. It is often the case that poor quality has to be addressed by changes in the process parameters. Given that lead times are always a target for reduction and WIP levels have to always be kept at a minimum, such disturbances of the process plan will deteriorate the material flow within the production system and ultimately limit its profitability.

From all the above, it is evident that modern machining systems should be able to respond to high variability in the processes they have to carry out, without compromising neither the accuracy of the product nor the productivity of the process. This will require control of the machining system in order to maintain its stability. It is known that as the force path is closing through the tool/work piece interface, the stability of the system can be achieved either by controlling the process parameters or the machine tool structure. Therefore, in order to expand the stable ranges of the machining system without compromising productivity, control strategies should move from the traditional paradigm of control through the process and focus on ways of controlling the structure of the machine tool.

The motivation behind this work is to develop a novel concept to control static and dynamic stability of the machining system by exploiting the
configuration of structural joint interfaces without any alteration of the process parameters.

Static stability refers to resistance to elastic deformations, determined by the compliance of the machine tool. Dynamic stability refers to resistance to vibrations and very often focuses on self-excited vibrations, or vibrations due to resonance. The structural behaviour of the machine tool is dictated by three parameters: mass, stiffness and damping. The classical design paradigm for machine tools (in terms of both static and dynamic stability) is rigidity maximization in order to reduce the compliance and therefore reduce deflections. Rigidity is usually enhanced by structural modifications which lead to higher mass, while maximum load is often applied on joint interfaces. Higher mass is rarely favorable, as it increases gravitational forces and decreases natural frequencies, while inertial forces of heavy moving components also increase, which is detrimental to the precision of the machine tool. High stiffness on the joint interfaces also has detrimental effects on damping as it will be explained in chapter 1.

Therefore, if one wishes to move away from the process-based control strategy, the design paradigm for machine tools has to be changed in order to accommodate structural joints that will allow for controllable static and dynamic capability.

This is an innovative solution that has not been proven in practical applications; by conscious design of the characteristics of the structural joints (stiffness and damping) and by enabling the functionality to control these characteristics can it be possible to control the machining system?

1.2 RESEARCH QUESTIONS

The previous discussion leads to the following research questions that will be addressed in this work:

1. Is it possible to incorporate consciously designed structural joints with controllable internal parameters, which when altered will allow for control of static and dynamic stiffness of the machining system?

2. Is it possible by adapting the dynamic stiffness by means of tuning the joint interface characteristics, to control the response of the machining system to excitation from the process?
1.3 THESIS STRUCTURE

The thesis begins with a discussion on stiffness and damping in machine tools in chapter 2, followed by a discussion in chapter 3 on the static and dynamic interactions between the machine tool and the process. Chapter 4 focuses on control strategies of machining processes with a focus on dynamic stability. Chapter 5 describes the design concept of the Joint Interface Modules (JIMs), followed by chapters 6 and 7 which present the findings of the investigations regarding the interface control parameters’ effects on the structure’s modal characteristics and their subsequent effects on the system’s response. Chapter 8 includes a brief discussion on applications of the JIMs and different kinds of control strategies that can be applied for their operation. Finally chapter 9 offers some concluding remarks and propositions for future work.
2 STIFFNESS AND DAMPING IN MACHINE TOOLS

Machine tools are elastic structures and their mechanical properties describe how the system will deflect under static and dynamic loads respectively. Such deflections, regardless of their static or dynamic origin will eventually affect the dimension, form and surface of the machined parts.

2.1 STIFFNESS, DAMPING AND THE ROLE OF JOINTS IN MACHINE TOOLS

One of the most important design requirements for machine tools is rigidity. The all increasing precision demands on machine tools are reflected on the capability of the machine tool to withstand excitation forces that otherwise will have detrimental effects on the accuracy of the machined components. Therefore, static stiffness is together with kinematic accuracy one of the most important criteria for design. At the same time the machine must incorporate sufficient damping in order to have stable dynamic performance. This becomes even more evident in the case of light weight machine tool structures, which are necessary for high speed machining. Such structures can exhibit a deteriorated dynamic performance since they exhibit lower attenuation characteristics.

There are three basic vibration energy dissipation mechanisms within a machine tool, which are basically the sources of damping:

- Material Damping, where energy is dissipated from the components’ materials
- Friction Damping, where energy is dissipated via friction and micro slip in the joints between contacting components, which is often contributing the most in the machine’s overall damping
- Viscous Damping from oil films in joints, bearings, guide ways etc.

A machine tool, being an assembly rather than a monolithic structure, has its natural characteristics defined by the characteristics of its components and largely by the properties of the joints.

The effect of joints’ stiffness and damping in machine tools have been extensively studied by Ito [12] and Rivin [13]. It is a known fact that with every joint introduced in the machine, its stiffness is decreasing and its damping is
increasing. Inamura and Sata illustrated this effect by comparing a solid beam and a beam with a bolted joint [14].

Ito very elegantly describes, for various types of machine tools, which joints are the “weakest links” and how much of the stiffness decrease is attributed to them [12]. Yoshimura tried to examine how pre-load and lubrication affect stiffness and damping in a joint and define them as stiffness and damping per unit area of contact in order to use the values for estimating the characteristics of machine tool joints[15].

![Figure 5. Classification of joints in machine tools [12]](image)

Various studies have revealed how stiffness and damping are affected by pre-load, oil in the joint interface and surface quality [16], [17], [18]. Also both Ito and Rivin describe extensively how for every joint, the stiffness and damping are depending on its geometrical configuration, the pre-load between the interfacing components and the roughness of the surface. In general, with increased pre-
load, the static stiffness will increase and damping will decrease. This means that dynamic stiffness\(^1\) can reach a maximum value depending on pre-load.

Having said all that, it is evident that the rigidity requirement is reflected as maximum stiffness in the joint interfaces. However, as mentioned earlier, this leads to lower damping which can deteriorate the dynamic behaviour of the machine tool. In the case of stable conditions where forced vibrations are comprising the response of the system a reduction in damping will not deteriorate significantly the response of the system. On the other hand in the case of an unstable process even a slight increase in damping could provide huge benefits. Therefore a tradeoff between static stiffness and damping has to be achieved for improving the dynamic response of the system during a machining process. This was pointed out even in the 1970’s by Sadek and Tobias when they clearly stated that it is dynamic stiffness the maximizing target [19], while Rivin exhibited that an optimal balance between stiffness and damping can be achieved in the tool-tool holder interface to increase the stability limit [20] by maximizing the product of stiffness and damping logarithmic decrement.

\(^1\) Dynamic stiffness is defined as \(K_{\text{dyn}} = K_a + iK_b\) where \(K_a\) is the in-phase stiffness and \(K_b\) the quadrature static-related stiffness. Put simply, it is the frequency dependent vibration response of the system over the excitation as a result of rigidity and damping.
Typical machine tool components have been studied regarding their damping behaviour and their dynamic stiffness. Ito provides a comprehensive summary of the dynamic behaviour of sliding joints, bolted joints, chucks and tool clamping mechanisms [12]. Rivin provides an extensive summary of the dynamic performance of available tool holders [21]. Fu studied the joint interface effects in the tool – tool holder interface [22] and demonstrated how dynamic stiffness can be optimized by adjusting pre-stress conditions in order to reach an optimal response. A study on the tool holder – spindle interface showed how increasing drawbar force leads to higher static stiffness. Additionally, the authors exhibited that an HSK interface is significantly stiffer than a taper one [23]. Cao and Altintas, in their study of spindle bearings, exhibited the increase in static stiffness with pre-load, cross examined with rotational speed effects and demonstrated how the static stiffness of the spindle is decreasing as its rotational speed is increasing [24], [25]. Brecher et al. studied the damping behaviour of linear guides and ball screw drives [26]. Interestingly, they discovered that depending on the direction of measurement, the viscous or structural damping model is appropriate to describe damping. Additionally they identified that damping ratio in the drive direction is up to 2.5 times the damping ratio in the cross direction. Furthermore they verified that with increasing pre-load in the guides, damping is decreasing as shown in Figure 8.
Hung et al. investigated the effects of preload on spindle bearings and linear guides. They identified that to the examined range of preload (12-65N), increased preload on the guides and bearings increased the dynamic stiffness of the spindle [27], [28]. However they acknowledge that bearings in more realistic, larger spindles would offer higher damping. This could mean that an increase in pre-load would have different effect on dynamic stiffness since the magnitude of the decrease in damping could be higher as shown earlier in [29]. This has been also been observed by Spiewak and Nickel, who showed that an increase in preload in the bearings increased the compliance of the spindle at higher natural frequencies together with their static stiffness [30]. At this point it has to be mentioned that pre-loading rolling bearings used in spindles, could have detrimental effects on their service life [31]. Furthermore, in their investigations presented in [32], they verified that an increase in pre-load of linear guides increased dynamic stiffness and the critical stability limit within the range measured.

Figure 8. Damping ratios of linear guides’ fundamental frequencies. [26] An increase in preload leads to a decrease in damping ratios, together with a shift of natural frequencies to higher levels.

Figure 9. Effects of linear guides preload on dynamic stiffness [32]. An observation at the local minima of the curves shows that with increased pre-load dynamic stiffness is increasing.
Lin and Tu, in order to examine the dynamic behaviour of a spindle, focused on the first two modes of a spindle. They examined how the preload in the bearings, the distance between them, the length of the spindle, the material of the shaft and the distance between the midline of the shaft and the cutting point affect these two modes, all cross examined with rotational speed effects [33]. A similar investigation was described in [34], where a spindle was optimized with regards to the factors contributing to its dynamic characteristics in order to reach a desirable stable cutting zone.

Liang et al. also studied the characteristics of linear guides and ball screws [35]. They identified that an increase in pre-load in the guideways has a different effect depending on the frequency band and the direction of measurement, either axial or normal to the travel direction. In some conditions it caused an increase in dynamic stiffness and in other cases a decrease, showing the complex effect of pre-load on dynamic stiffness. In the case of ball screws, they identified much weaker effects of pre-load on dynamic stiffness compared to linear guides.

Shinigawa and Shamoto also examined linear guides with respect to different friction forces and correlated that to Stability Lobe Diagrams (which will be described later). They verified that there is an optimum friction force where dynamic stiffness is maximized, and that exceeding that force causes a damping decrease which consequently decreases the stability limit [36].

The aim of such investigations is often to create models that can be used to estimate the natural characteristics of a machine tool and ultimately use them to predict its response. A great challenge in such efforts is the inherent non-linearity of the load-deflection curve which makes the estimation not only of the magnitude but also of the shape of the curve extremely hard.

Furthermore, a typical problem in dynamic investigations of a machine tool structure is that the measurements obtained, e.g. from EMA, are providing damping “globally” and cannot be easily used for understanding the contribution of every joint to the dynamic stiffness of the machine tool. On the other hand, examining the joint in a disassembled configuration leads to a characterization that does not take into consideration the effects from the rest of the structure. Cao and Altintas demonstrated that for accurate modeling, the models of at least spindle shafts, tool and holders, bearing preload, connection between the spindle and machine tool housing, speed and machining process have to be integrated [37].
Simulations by FEM can capture the frequencies of the dominant modes but fail to predict the amplitudes of compliance and modal frequencies originating from joints and other components as seen in Figure 10. More than often, FEA simulations do not take into consideration pre-loading effects discussed above, which is a significant source for discrepancies from experimental characterization. Additionally, even if the characteristics of the joints are known, substructuring or modal synthesis techniques so far suffer from accuracy in predicting the global response of the system. Yigit et al. in their efforts towards reconfigurable machine tools provided a summary of relevant methods applied for machine tools and proposed a method for dynamic stiffness evaluation of assembled structures from their individual components [38].

![Figure 10. Comparison of receptance FRFs from FEM and EMA testing of a 5-axis milling machine [39]. The FEM simulation overestimates stiffness and shows high discrepancies in both frequency content and amplitudes.](image)

### 2.2 CONCLUSIONS

The existing joints in machine tool, from the machine body joints to bearings, largely define the structural characteristics of the machine tool and therefore its static and dynamic capability. The contact conditions on the interface define these characteristics and conscious control of these conditions can allow for control of the structural characteristics. By careful design of the geometry of the interface and application of the optimum pre-load on it, it is possible to optimize the dynamic stiffness of the structure.

Although being aware about the significant contribution of joints’ stiffness and damping to the overall capability of the machining systems, the classical theory of the machining system lacks a unified concept for consciously designing structural interfaces with controllable characteristics. Apparently, there is a tendency today both among scholars and manufacturers to develop and
implement complex schemes for on-line monitoring and control of machining systems. The simple explanation is the existing knowledge gap between machine tool manufacturers and machine tool users regarding static and dynamic capability. Due to unlimited combinations of tools’ geometries and materials, work pieces’ shapes, dimensions, and materials, fixtures and toolholders it is nearly impossible to predict the behaviour of a machine tool and by this the accuracy of resulted parts. The consequence is that the machine tool users are forced to add advanced sensor systems for monitoring and controlling the machining system. In an industrial environment these solutions are costly and not reliable due to the adverse conditions they are subjected.
3 STATIC AND DYNAMIC INTERACTIONS IN MACHINING SYSTEMS

In order to understand the control strategies of static and dynamic capability of the machining system it is important to study the interaction between the two subsystems: machine tool structure and the cutting process. The two subsystems are assumed to in static and dynamic stability when considered as independent systems; it is their interaction that can drive the system into an unstable state and cause deflections that are responsible for errors in the geometrical features of the part.

The previous chapter focused on stiffness and damping originating from the machine tool structure. The other source of stiffness and damping in the machining system comes from the interaction between the structure and the process.

3.1 STATIC INTERACTIONS IN MACHINING SYSTEMS

The static loads from the process and the weight of the components are propagating through the structural loop and the kinematic chain. As the components of the machine are moving (table, spindle, turret, etc.) these loads are changing in the applied position, direction and magnitude, which in their turn apply different moments on the machine tool structure. At the same time the stiffness of the structure changes at different positions and directions, with an example shown in Figure 12. The spatial dependency of stiffness is not only due to macro – scale parameters like the geometry of the structure or the relative
position of components, but also due to geometric errors on joint interfaces (e.g. deviations on the clearance between contacting components of a guideway can cause deviations on the modulus of elasticity of the interface)[41]. Errors due to thermal cycles are also a source for deviations; however they are outside the scope of this thesis.

All these factors influence the outcome of the interaction of the loads and the structure; the change of loads and stiffness along the tool path will be reflected on deviations from the desired geometry on the work piece.

The forces developing during the cutting process cause the system to deflect not only due to the finite stiffness of the structure, but also due to the finite stiffness of the process itself. Therefore, if the process is modeled as a spring between the tool and the work piece, the deformation occurring will be
which implies that geometrical variations during the process will occur either due to changes in the cutting forces because of a varying depth of cut, or due to changes in the machine tool structure. The cutting force developing during the process can be written as:

\[ F = C b h^m \] 

(3.2)

where \( C \) is a material constant, \( m \) an exponent related to the geometry, \( b \) the width of cut and \( h \) the chip thickness. Assuming an entrance angle \( \sigma \), \( b \) and \( h \) can be written as:

\[ b = \frac{a}{\cos \sigma}, \quad h = f_r \cos \sigma \] 

(3.3)

Where \( a \) is the depth of cut and \( f_r \) the feed (in mm/rev). The cutting force can now be expressed as:

\[ F = C \cos^{m-1} \sigma f_r^m a \] 

(3.4)

The cutting stiffness can now be defined as:

\[ r_a = C \cos^{m-1} \sigma f_r^m = \frac{F}{a} \] 

(3.5)

which is the coefficient of proportionality between the cutting force and the depth of cut. It is obvious now that any change in the depth of cut during the process will affect the cutting stiffness and therefore this will be reflected on the dimensions of the work piece. This means that any dimensional error on the raw material or a semi finished work piece will result in variation in the depth of cut and eventually will be copied to the machined surface. The ratio of cutting over structure stiffness, defined as

\[ \mu = \frac{r_a}{K_a} \] 

(3.6)

is a reflection of the static interaction between the machine tool and the machining process. Then for every pass we can calculate the ratio \( i \) which describes the rate of improvement of the form errors on the work piece[41]:

\[ X = \frac{F}{K_a} \] 

(3.1)
However, modern manufacturing needs require shortest possible cycle times, which means that the process must be finished in the minimum possible amount of passes, which places limitations on how this way of increasing precision can be exploited.

3.2 DYNAMIC INTERACTIONS IN MACHINING SYSTEMS

A both inevitable and undesirable outcome of machining processes is vibrations. During the material removal process, the machining system is subjected to excitation from the forces that are developing and vibrations are the dynamic response of the structure to the excitation. Vibrations even in a stable machining process can have detrimental effects on surface finish and geometrical accuracy of the machined product, productivity of the machining operation, tool life, machine tool health, noise in the work environment etc.[43],[44],[45]. Ultimately all these factors have a detrimental effect on production costs and even health and safety of the operators. Such vibrations can be of three types; free, forced or chatter.

![Figure 14. Types of vibrations in machining](image-url)
3.2.1 FREE VIBRATIONS

Free vibrations occur when the system is displaced from its equilibrium position through some impulse and returns to its original position. Such type of response is rarely a problem for a structure like a machine tool that has high rigidity and relatively low excitation forces. In this case the equation of motion of a system with damping will be:

\[ m\ddot{x} + c\dot{x} + kx = 0 \quad (3.8) \]

3.2.2 FORCED VIBRATIONS

Forced vibrations occur either due to excitation from alternating cutting forces and internal or external sources. An intermittent process like milling is a typical source of forced vibrations. Other process related sources could be disparities in the material properties of the work piece, BUE or segmented chip formation. Internal sources could be imbalanced rotating components, worn moving parts or inertial forces. External sources could be excitations from the surroundings, which transfer through the foundations of the machine. In this case the equation of motion would be:

\[ m\ddot{x} + c\dot{x} + kx = F(t) \quad (3.9) \]

Forced vibrations, close to an eigen-frequency of the system can lead to resonance which can cause severe surface damage of the work piece and can prove catastrophic for the machine tool condition if the process becomes unstable.

3.2.3 CHATTER

The term chatter is used to describe vibrations during an unstable cutting process. It has been studies extensively since the 1960’s with Tobias [46], [47] and Tlusty [48],[49] being among the first researchers of the phenomenon. Chatter can be either due to self-excited vibrations or excitation at a resonance frequency of the machining system (e.g. in milling when the tooth passing frequency coincides with a natural frequency of the system).
Self-exited vibrations are guided by an interaction between the machine tool’s natural characteristics (including structure, tooling and work-piece) and the process parameters [50]. Such phenomena arise from 4 different mechanisms although all of them cause a variation in the cutting forces [44],[51]. These mechanisms are: velocity variations, frictional, regenerative and mode coupling. Regenerative chatter and chatter due to mode-coupling are the most common sources of chatter. Regenerative chatter occurs when cuts in a machining process are overlapping and a modulation in the uncut chip thickness works as a source of vibration amplification, while the dynamic stiffness of the system is not sufficient to keep the system in the stable regime. The regenerative effect described above can be easily understood if the machining system is illustrated in the form of a control loop diagram between the structure and the process as shown in Figure 15, where the variation of the relative displacement between tool and work piece is causing a variation in the cutting parameters and consequently a variation in the cutting force.

![Figure 15. Block diagram representation of a machining process with noise input](image)

### 3.3 PROCESS DAMPING

When it comes to machining process another source of damping is arising during the process, the so called process damping. Tyler and Schmitz recently provided a summary of efforts to model and exploit process damping [52]. Process damping’s influence on the overall damping is increasing at low cutting speeds where the flank of the tool is interfering with the machined surface, leading to increased energy dissipation. Based on the process damping force model proposed by Tyler and Schmitz, the overall damping in a milling process can be described by the following equations for up and down milling respectively:
\[ c_{x,\text{total}} = c_x - C \frac{b}{V} \cos^2(90 - \varphi_{\text{ave}}), \quad c_{y,\text{total}} = c_y - C \frac{b}{V} \cos^2(180 - \varphi_{\text{ave}}) \]  
(3.10)

\[ c_{x,\text{total}} = c_x - C \frac{b}{V} \cos^2(\varphi_{\text{ave}} - 90), \quad c_{y,\text{total}} = c_y - C \frac{b}{V} \cos^2(180 - \varphi_{\text{ave}}) \]  
(3.11)

Where \( c_x \) is the viscous damping coefficient, \( C \) the process damping coefficient, \( b \) the depth of cut, \( V \) the cutting speed and \( \varphi_{\text{ave}} \) the angle of average surface normal direction.

Apart from its velocity dependency process damping is strongly dependent on the tool geometry. Tool wear has been demonstrated to have a strong increasing effect on process damping [53], [52] and it is often the case in practice that new inserts are prone to exhibit chatter in an otherwise stable operation. In general, the bigger the contact at the flank, the higher the process damping is. Budak and Tunc have also clearly demonstrated that process damping is increasing with the hone radius, the radial immersion and the vibration frequency, while an increase in clearance angle and in the number of teeth in milling tools cause a decrease in process damping [54], [55]. Additionally, they describe that process damping coefficient is behaving like a softening spring as vibration amplitudes are increasing. Given that process damping is manifesting evidently in lower cutting speeds, all the above mentioned effects are also stronger in low cutting speeds. The focus of this thesis is towards the joint effects on machine tools’ response so this area will not be studied further. During the experiments that will be described further on, the relevant data were obtained in machining with used edges.

Figure 16. Effects of tool wear on process damping and process stability [53]. After using a new tool, the system becomes less sensitive to chatter development in rotational speeds below 4000 RPM.
3.4 CONCLUSIONS

The interaction between the machine tool and the machining process define the capability (static and dynamic) of the machining system. The force acting on the structural loop and the stiffness of the structure are constantly changing along the tool path, leading to geometrical deviations on the machined part. The dynamic stability of the process is also defined by the machine – process interaction, where the damping offered by the structure and the process can keep the system away from excessive vibrations, regardless of whether they are forced or self-excited.
4 CONTROL STRATEGIES OF MACHINING SYSTEMS

The control of machining system stability can be achieved by changing dynamic parameters of the machine tool, the process or both. The traditional way to control machining system dynamic behaviour is by controlling the process parameters, i.e., depth of cut, rotational speed, speed rate as well cutting tools’ micro-geometry and material. In this manner, static stiffness in the direction of cutting force and overall damping of machining system are improved.

In the context of this thesis an important part is the control of the dynamic response of the machining system. Vibration control techniques so far can be classified in two main categories [56] which are shown in Figure 17.

![Figure 17. Classification of vibration control approaches.](image)

Process oriented approaches, which aim at selecting optimal parameters which keep the process stable. Design targeted approaches aim at altering the characteristics or the behaviour of the system in order to improve the dynamic performance of the machine. A comprehensive synopsis of the advantages and disadvantages of every technique have been described by Daghini [57].

4.1 CHATTER DETECTION, IDENTIFICATION AND CONTROL

Chatter detection strategies depend on sensors for acquiring signals necessary to evaluate if the system is drifting out of stability. Such strategies are also a requirement for active techniques for chatter suppression. In the simplest form such a detection technique triggers a change in the cutting speed towards already
known stable parameters, or simply to avoid the coincidence of the tooth passing frequency and an eigen-frequency of the system. Kuljanić et al. provide a comprehensive summary of chatter identification systems [58]. Models for estimating operational damping on line have also been studied for turning [59], and milling [60], [42].

The classical approaches are based on non-parametric methods, e.g. Fourier transform algorithms (such as FFT). The magnitude of the signal for a certain frequency (frequencies) is monitored. The main drawback is the very low amount of energy for the chatter to be excited. This makes difficult to distinguished between forced and chatter vibration, to detect the instability border and to implement robust control strategies. Another drawback is the limited prediction capability of this approach that results in chatter marks on the work before chatter to be detected.

The major issues regarding the classical control are:

- Before controlling one has to discriminate between chatter and forced vibration because there are different approaches. For instance, chatter stability can be controlled by tuning the rotational speed which does not have effect on forced vibration produced for instance by some fault in the machining system.
- Difficult to implement in industrial environment due to complication with sensors and cables and auxiliary equipment.
- Difficult to be integrated in the NC system of machine tools.
- Difficult to change any of the three cutting parameters as depth of cut is established at process planning stage, feed rate is not obviously correlated to chatter and rotational speed is selected in relation to cutting speed which controls the thermal stability and productivity.

### 4.2 STABILITY LOBE DIAGRAMS

The most common approach in studying and improving the behaviour of a machine tool with respect to chatter is the Stability Lobe Diagram, with an example of it shown in Figure 18. This diagram distinguishes the area of combinations of depth of cut and spindle speed between stable (below the stability border) and unstable (over the stability border). An overview of the methods used for obtaining the SLD is given by Quintana et al [61]. An important quantity in this kind of analysis is the so called “critical stability limit”
which defines the boundary of depth of cut, under which stability is unconditional. This limit can be calculated for milling from the following equation:

\[ b_{lim} = \frac{-1}{2K_s \text{Re}[G(j\omega_t)]N_t^*} \] (4.1)

Where \( K_s \) is the specific cutting force, \( b_{lim} \) the critical depth of cut and \( N_t^* \) is the average number of teeth in cut, defined as:

\[ N_t^* = \frac{\varphi_e - \varphi_s}{360}N_t \] (4.2)

Where \( \varphi_e \) and \( \varphi_s \) the exit and entrance angles respectively. \( \text{Re}[G(j\omega_t)] \) corresponds to the minimum of the real part of the receptance FRF. For turning, the stability limit can be calculated from equation 3.5 by having \( N_t^* = 1 \)

A requirement for carrying out the SLD is that the natural characteristics of the system to be extracted and the Frequency Response Functions (FRFs) for the system to be known usually via Experimental Modal Analysis (EMA) or through FEM simulations.

Apart from the cutting speed and the depth of cut, other characteristics of the process have a significant effect on its stability with the position of the tool within the working area having a significant contribution. Based on the studies by Wanner, a process with decreasing chip thickness (down milling) should provide a process less sensitive to instability compared to a process with increasing chip thickness (up milling) [62].

In the case of ball end mills, which are important in 5-axes milling, an increased lead angle of the tool has been reported to provide a more stable process [63].

The SLDs allow for discovering the stable process parameters that provide the highest Material Removal Rate. The main drawback is that they are dependent on accurate structural modeling of the machine tool and this so far cannot be performed taking into account the process, which closes the loop between the work piece and the tool. It has been demonstrated that as the cutting force increases the dynamic compliance of the tool is decreasing [64], which can have strong effects in the reliability of the SLDs.
Additionally, the stiffness of spindles, which are often the source for chatter, is dropping at higher rotational speeds [65], [66], [67]. Ozturk et al. provided a summary for stiffness variations in machine tool spindles [68]. Furthermore the natural characteristics of the system change while the spindle column or table move along the feed axes as shown in Figure 19. For example the compliance of a ball screw-nut system can change up to 20% depending on the position of the nut [69]. Such effects on SLDs were also demonstrated in [70].

Another example of natural characteristics changing during the process is the case of thin walled components where the natural characteristics of the work-piece change since the volume of the material being removed is comparable to the volume of the material remaining [71]. These facts render FRFs acquired in the static, unloaded state of the machine unreliable for prediction of stability lobes, especially in high rotational speeds. Additionally, the choice of cutting data is subjected to many other constraints which elevates the issue into a complex optimization problem.

In the context of this thesis, SLDs were used as a tool to guide the experiments and it is not the scope of this work to study further or improve SLDs’ performance.
Figure 19. SLDs of a ram milling machine for different positions along feed directions. The stability limit and the stable speed ranges are varying significantly depending on the position of the spindle. [72]

4.3 PASSIVE TECHNIQUES

Most passive techniques fall within three main directions; introducing dynamic absorbers, mainly Tuned Mass Dampers (TMDs) [73], [74] and Tuned Viscoelastic Dampers (TVDs) [75], [76], increasing damping in the system [57], [77], [75], [78], or using cutters with variable pitch [79] in the case of milling.

Tuned mass dampers consist of a mass and spring imposed on the structure targeting to absorb the vibration energy while TVDs follow the same principle, but with viscoelastic polymer layers introducing a damper effect. TMDs and TVDs do not adversely affect the stiffness of the structure, however they can remove energy within a narrow frequency range, which is relevant only for a specific machining situation and they require tuning if the excited frequency changes. Other types of dynamic absorbers include replicated internal viscous dampers [80].

Increasing damping aims at expanding the stable region of machining as demonstrated in Figure 18. A popular way for increasing damping is to apply layers of viscoelastic material with high elastic modulus and loss factor on machine tool interfaces [57], [81]. The principle behind energy dissipation in
viscoelastic polymers is that shear strain on the material is transformed into heat as the long chain polymers of which they consist are deforming. The effectiveness of using such materials is depending on a careful selection of the placement position so that the amount of strain energy that shears the viscoelastic layer is maximized.

Such materials are used in three configurations: Free layer damper (FLD), Constrained layer damper (CLD) and the TVD concept described before. FLDs are consisting of a single layer of damping polymer applied on the structure [82]. CLDs are consisting of a layer of damping polymer between the structure and a constraining layer. Daghini [57] exhibits how the amount of polymer layers, the constraining layers and the pressure on the interface affect the loss factor of the sandwich structure when applied on a boring bar and a turret interface.

The main drawback with increasing damping in this way is that it often leads to a significant loss of stiffness and it is often the case that it has to be introduced with a redesign of the targeted structure in order to provide results in machining. Additionally, such materials have a limit on the pressure they can withstand, they are heavily affected by temperature, exhibit deterioration of their performance over time due to creep and have limited effectiveness when very large masses are involved.

Other damping materials include viscous fluids, magnetic, passive piezoelectrics or even sand used to fill the cavities of a machine tool structure.

4.4 ACTIVE TECHNIQUES

Active techniques often target vibration cancellation by means of a system of sensors and actuators that either compensate for the dynamic forces [83], [84], [85] or deflections[86] which are developing during the process or adapt the cutting speed to keep the process stable[87]. The main drawback with these approaches is that even if the chatter detection process is very efficient, the control loop will change the behaviour of the system after the onset of instability, at which point it is very hard for the process to return to a stable state. Additionally such active approaches require expensive and sensitive equipment that are often unsuitable for manufacturing environments and often require significant modifications of machine tool components.
4.5 CONCLUSIONS

The aim of the chapter was to present the state of the art in the control of the dynamic behaviour of the machining system. Stability lobe diagrams are a common method to discover the stable machining regions as an input to the process plan. Chatter identification and control aims at identifying online whether the process is stable or not and trigger an action that will bring the process back to stability. Passive techniques mainly aim at disrupting the excitation of certain eigen-frequencies of the system. Active techniques depend on detection and identification techniques in order to change the process parameters or to compensate for forces or deflections. All these techniques can be useful but are either harmful for productivity as they often lead to suboptimal cases, or very complicated and expensive or narrow in their applicable range.
5 THE JOINT INTERFACE MODULE CONCEPT

5.1 INTRODUCTION

The previous chapters provide the motivation for developing a new concept for controlling the machining system. First of all, is the necessity to move towards machining systems with controllable, rather than fixed capability (in terms of static and dynamic performance). This necessity has been recognized in the past in the level of manufacturing systems with paradigms like flexible manufacturing systems [88], reconfigurable manufacturing systems and changeable manufacturing systems [10], which aimed to cope with mass customization, increasing varieties and decreasing volumes.

Such manufacturing systems paradigms, naturally create requirements for flexibility and robustness in the machine tools, so that they will be able to carry out processes with high levels of variability. The enablers of such capabilities that will allow for non-process based control of machining systems are machine tools that allow for control of their structural properties as explained in the introduction.

It has to be mentioned at this point that an issue when it comes to the dynamic performance of the machine is that chatter problems are detected only during the operational phase of the machine and depend a lot on the geometry of the work-piece. Therefore it is common that in real manufacturing situations chatter is addressed when it is actually occurring in a case-specific, problem-solving manner. In such cases the common solution of process-based control will drive the process away from the already defined optimal cutting parameters. Figure 20 shows an example of the signal from the milling process of an impeller; the complex shape of the impeller means that during the process, the radial depth of cut is changing and when the cutting tool is engaged in its full diameter, the stability limit is exceeded and the process becomes unstable. Any attempt to intervene in the process by reducing the allowable depth of cut would create a suboptimal tool path and eventually increase the cycle time.
If a design change in the machining system is put in place at such a late stage (e.g. designing a less compliant fixture), the cost would be rather high compared to its benefit [89] and can also require a re-evaluation of the process plan. Additionally, such redesigning often means introducing stiffer components, which in turn often lead to mass increase of moving components, clearly unfavorable as feed, jerk and precision requirements increase [90]. Such attempts will also be rather restricted in their effectiveness since without a unitary design approach, the partial stiffening of the structure will only be in reality shifting the problem from one part of the structure to another.

The following sections attempt to describe general directions for the requirements of such structure-based control, followed by the experimental investigations around the proposed concept.

5.2 REQUIREMENTS FOR JOINT INTERFACE MODULES

The first and most important principle in the proposed concept is the use of carefully and consciously designed joint interfaces that form a Joint Interface Module with controllable (which implies measurable and repeatable) stiffness and damping. If these JIMs are placed as an intermediate component in an already existing assembly, then the damping and the stiffness of the deliberately designed joint must overmatch the stiffness and damping of the adjacent interfaces. This means that all other connections between JIM and structural elements have very high stiffness and very low damping. All contact surfaces have to be carefully designed and machined for controllability purpose.

These interfaces can be either stationary or movable and should be applied as close as possible to the process (e.g. toolholding/workholding), or in components
that have a significant effect on the accuracy of the product (e.g. guideways). Figure 21 shows an example of how JIMs can be integrated in existing interfaces of a lathe.

![Diagram of JIM integration in a lathe](image)

**Figure 21.** A concept for integrating JIMs in the machine tool structure of a lathe.

It was explained in chapter 2 how the structural characteristics of a machine tool can change by changing the properties of the joints. The primary way JIMs aim to take advantage of this is by exploiting the effect pre-load has on the stiffness and damping of the joint. This requires that the JIMs need to provide the functionality of altering pre-load on the interface through some sort of actuation (e.g. piezoelectric or hydraulic).

An alternative way that JIMs can be used is without active control on the pre-load but by creating a high damping interface with the introduction of damping materials on the interface (e.g. viscoelastic polymer layers). Of course this does not exclude the combination of a pre-load mechanism with high damping surface treatment as it will be show further in the experimental part of this thesis. These passive JIMs rely only on the careful design of the interface geometry and the selection of the appropriate material and the right quantity. In line with the previous discussion on joints’ stiffness and damping, the passive JIMs can also be used with manual adjustment of pre-load on the interface in order to achieve the desirable dynamic stiffness for the application.

Whether active of passive, the successful introduction of such components requires a deviation from the traditional paradigm of designing for rigidity;
stiffness maximization and kinematic accuracy should not be the only design targets and the machine tool joints should not be treated as a source of compliance. In the traditional design of machine tools, joints are the main source of the uncertainty. It is very surprising that despite the effect of joints on overall structural stiffness and damping, no attempts have been made to develop joint modules with known and controllable characteristics. Joints have the main contribution to structural stiffness and damping. Joints are in machine tools the main source of uncertainty due to complex contact conditions and complex non-linear tribological parametric relationships. The condition of a joint varies in time, with the velocity of movement and also with respect to the relative position between tool and work piece. Machine tool design should be performed in a way that exploits the behaviour of the joints and embrace them as functional components that can control the outcome of the machining process.

Another important characteristic of the JIMs is that they have their limitations in their operating range based on their design. A machine tool that aims for a wide range of static and dynamic flexibility should exploit the functionality offered by the JIM concept. Therefore for every machine tool, a set of JIMs should be available, with each one covering the capability gaps of the others. It becomes obvious that since these modules have to be interchangeable, they have to be designed in a way that the assembly on the machine body is fast. Needless to say that positioning repeatability also needs to be high in order to avoid compromises on the precision of the machine tool. These facts elevate the need for highly modular design of a JIM machine tool; all the interfaces between the JIM and the machine tool (structural, hydraulic, pneumatic, electric) have to be taken into consideration, the covers of the machine tool have to accommodate positioning and assembly, even the work area has to be part of the design of the machine in order to accommodate a what is often called “plug and produce” machine tool component.

Last but not least, a necessary part which will close the control loop in a machining system with JIMs is the monitoring of the process which will provide the feedback for the control strategy of the JIMs.

As the JIMs’ purpose is to enhance the static and dynamic capability of the system monitoring has to target two directions:

- towards monitoring geometric deviations of the work pieces
- towards monitoring the stability of the machining process
Figure 22 shows an extended control loop for a JIM machining system. Based on measurements of the dynamic response of the system, of geometrical features’ dimensions and of surface integrity the control system identifies whether there are deviations from the desired geometry and calculates operational damping in order to identify if the process is becoming unstable. If necessary, the control system discovers the target values of stiffness and damping for the system in order to keep the desired outcome of the process and sends the target values for the actuating mechanism in order to achieve the desired levels of pre-load. However a simpler and more efficient control strategy can be deployed which does not require such a complex monitoring system, but just a sensor for monitoring the dynamic response; at an initial stage, the JIMs are set to high stiffness in order to ensure the dimensional stability of the work piece. At a second stage, damping is added if necessary based on the signal from the sensor. By measuring the geometrical and dimensional errors on the work piece the JIM’s stiffness and damping can be corrected for optimal operation.

It is not the scope of this thesis to focus more on the control aspects or algorithms for the JIMs but it was necessary to present how monitoring and control is interacting with the JIMs.
6 MILLING PROCESS CONTROL THROUGH TUNING OF THE JOINTS’ CHARACTERISTICS

6.1 THE JOINT INTERFACE MODULE (JIM) WORK-HOLDING DEVICE FOR MILLING

The prototype used for the investigations in this work consists of two basic components (1) - upper and (2) - lower which create a joint interface with a specific shape. The two components are cross connected with two U-shaped components (3) and (4) which are also providing the seating for three piezoelectric actuators (5). Each actuator is pre-stressed by a T-shaped component (6), on which a bolt (7) is exerting the compression force. This blocked-blocked configuration is the source for the controllable pre-loading on the interface; as voltage is supplied to the actuators they expand, pushing away the two U-shaped components. As these components are cross connected with the interface components (1 and 3, 2 and 4), these in turn are pushed towards each other, increasing contact pressure on the joint interface. When minimum torque is applied on the bolts of the actuation configuration and no voltage is supplied to the actuators, pre-loading of the joint is at its minimum, approximately 3 kN. When maximum torque is applied on the bolts and maximum voltage is supplied to the actuators then the pre-loading of the joint is at its maximum. At this configuration, pre-load is approximately 13 kN.

Figure 23. Representation of the JIM work holding device.
For the investigations carried out in this work, the work piece is bolted on steel plate which is in turn bolted on the JIM. DC Voltage to the actuators is provided by an amplifier with gain value of 20. The maximum voltage that can be supplied to the actuators is 140 V.

6.1.1 INTERFACE TREATMENT FOR DAMPING ENHANCEMENT

The damping configurations on the interface consist of:
- No damping enhancement (Metal to metal contact), from now on referred to as untreated configuration
- Application of a sandwich of viscoelastic material layers, from now on referred to as VEM configuration
- Coating of component 1 (Metal to coating contact), from now on referred to as Coated Configuration

a) Viscoelastic Damping Materials

For the VEM configuration investigated in this thesis 2 kinds of VEM were used. The first is a constrained layer type VEM (3M 2552) where the VEM is adhered on an aluminum sheet. The second type is a free layer VEM. In both cases the polymer is adhesive although often epoxy is used to strengthen the sandwich structure. Two constrained layers were applied on the top component of the joint interface and a final layer of free-layer VEM was applied. Additionally, layers of both types of VEM were applied on three features on the periphery of the JIM, which are used to restrict motion on the XY plane.

b) Carbon Based (CNx) Nano Composite Damping Material

The material used in the coating and its properties are described in [91], [92]. The principle behind its enhanced damping is via friction between the pillars of its microcolumnary structure. It is not known to increase friction damping caused by microslip between interfaces. The 700 µm thick CNx film layer was deposited onto the ‘top component’ (substrate for this study) of the work-holding device.
6.2 MODELING AND SIMULATION OF THE STRUCTURAL CHARACTERISTICS OF THE SYSTEM

Initially, the JIM was modeled as an assembly of masses, while the interfaces were modeled as thin elastic layers. Necessary inputs for the model are the stiffness and the damping for each of these thin elastic layers. Depending on the damping treatment configuration, an appropriate model for the interface stiffness and damping is used. Figure 24 shows a representation of the JIM as a system of masses, springs and dampers. The interface with controllable pre-load and damping treatment is shown in red.

![Diagram](image)

Figure 24. The JIM represented as a system of masses, springs and dampers. Interfaces are modeled as a spring-damper system. The third actuator is omitted from the sketch for better visualization.

The results of the simulations are showing that the subsystem of the actuators and the connecting components are having a significant contribution to the overall compliance of the system especially in frequencies over 2000 Hz. This is also demonstrated in the appendix where the actuator mechanism is replaced by a middle screw which was used as a pre-load delivery mechanism. In the context of this investigation, the eigenfrequencies of this subsystem are not relevant and only the results regarding mode shapes relevant to the controllable interface are discussed.
6.2.1 SIMULATION OF THE UNTREATED CONFIGURATION

Apart from modeling the geometry and the material properties, the characteristics of the joint interfaces have also to be modeled. For that reason every interface was defined as a thin elastic layer where the stiffness per unit area and the loss factor have to be defined. In the untreated configuration the following models are used to calculate stiffness and damping on the interface, which are then inputs to the model. Since for all interfaces the normal load on the interface is known, the stiffness per area of contact can be defined using the formula:

\[ K_{\text{unit area}} = \frac{1}{Cm} p^{1-m} \]  

(6.1)

and the damping ratio can be defined as:

\[ \zeta = \beta p^{-\gamma} \]  

(6.2)

Where \( p \) is the pressure on the interface and \( C, m, \beta \) and \( \gamma \) are constants, as described by Ito [12]. For different pre-load configurations, and therefore pressure on the interface, different stiffness and damping ratio are input to the simulation.

In the untreated configuration the modes calculated are shown in Table 1 and their respective shapes are shown in Figure 25. With the exception of mode 4, damping is reducing for the rest of the modes.

| Table 1. Natural Characteristics of the JIM, untreated configuration, simulation results |
|---|---|---|---|
| Low Pre-Load | High Pre-Load |
| **Frequency** | **Damping Ratio** | **Frequency** | **Damping Ratio** |
| 354 | 0,46% | 406 | 0,41% |
| 1000 | 0,29% | 1195 | 0,15% |
| 1128 | 0,49% | 1241 | 0,38% |
| 1954 | 0,15% | 1980 | 0,17% |
| 2524 | 0,19% | 2527 | 0,18% |
Figure 25. Mode shapes of the system, simulated results

Figure 26 shows the receptance FRFs in the X direction. The increase in stiffness as an outcome of the increase in pre-load, is reflected on a shift of the eigenfrequencies to the right. In terms of dynamic stiffness, it is increasing with pre-load only for modes 2 and 3, while for modes 1, 4 and 5 it is decreasing, illustrating how decreased damping can deteriorate dynamic stiffness.

Figure 26. Receptance FRF, Untreated configuration, X direction, simulated results. Red curve: low pre-load, Blue curve: high Pre-Load, simulation results
6.2.2 SIMULATION OF THE VEM CONFIGURATION

In the VEM configuration the metal to metal interfaces were modeled as previously. However the VEM has frequency dependant stiffness and damping. Frequency dependent loss factor can be obtained from manufacturer’s data sheet. Stiffness per unit area is modeled according to the following equation, where frequency dependant young’s modulus $E(f)$ and Poisson ratio ($\nu$) are also obtained by the manufacturer, while $H$ represents the material’s thickness.

$$K_{\text{unit area}} = \frac{E(f)(1-\nu)}{(1+\nu)(1-2\nu)H}$$ (6.3)

![Figure 27. Receptance FRF, VEM configuration, X direction, simulated results. Red curve: low pre-load, Blue curve: High Pre-Load, simulation results](image)

In the case of the VEM material, pre-loading does not seem to cause a shift in the natural frequencies which indicates that probably much of the pre-loading is consumed in compressing the viscoelastic material layers. Interestingly, the simulations showed that only for the last mode dynamic stiffness is increasing with pre-load, which indicates that friction damping decrease from the other interfaces of the system can influence its overall behaviour even though the VEM was applied on the main interface of the system.
Figure 28. presents a comparison of the FRF’s between the untreated and the VEM configuration. From the simulation results, the application of the VEM seems to have a strong effect especially in the first natural frequency where the displacement amplitude is almost 10 times less. Dynamic stiffness is also increasing for the rest of the modes. An examination of the frequency content shows that the application of the VEMs would cause a decrease in static stiffness of the system as the FRF curve is shifting towards lower frequencies. From the shape of the curves it can be seen that for especially the first mode, the application of the VEM would provide significant damping in the system.

![Figure 28. The effect of applying VEM on the interface, high pre-load configuration, X direction. Magenta: Untreated configuration, Green: VEM configuration, simulation results](image-url)
In order to understand the effect of pre-loading on the natural characteristics of the JIM, modal tests were performed supported by simulations of the structural behaviour of the system. Using an impact hammer as the source of excitation and 3 accelerometers on the work piece to measure the response, Frequency Response Functions were obtained. From the measured acceleration, the following quantities were obtained: natural frequencies, static and dynamic stiffness and damping ratios for the natural frequencies. Firstly, results are presented in order to explain how pre-loading is affecting the structural characteristics in the different damping configurations. Then, the results are examined regarding the effect of different damping configurations on the response. All receptance FRFs in this chapter are synthesized unless stated otherwise.

### 6.3.1 EFFECT OF PRE-LOADING ON STRUCTURAL CHARACTERISTICS - UNTREATED CONFIGURATION

The first set of experiments was carried out in order to identify how the JIM behaves in the untreated configuration when the pre-loading on the interface changes. Table 2 shows how static stiffness is affected by changes in pre-loading. As expected, when pre-loading is increasing, the system becomes more rigid and static stiffness is increasing. This can be observed in Figure 29, where the receptance curve is moving upwards at zero frequency and a shift of natural frequencies to the right is observed. However the extent of the shift differs depending on the mode which illustrates the non-linear nature of the system.

<table>
<thead>
<tr>
<th>X direction</th>
<th>Y direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low Pre-Load</td>
<td>High Pre-Load</td>
</tr>
<tr>
<td>55865</td>
<td>129843</td>
</tr>
</tbody>
</table>

It has to be mentioned here that without mode shape analysis of the results it would be impossible to distinguish which modes are the “same” between the two pre-stress configurations; the non-linear frequency shifts make it impossible to
compare natural frequencies just by the order they appear on the FRF. Additionally, it is not uncommon that the change in pre-loading can allow modes that are masked under a higher amplitude mode to become distinct in the second configuration and modes that are identified in one configuration either to become masked or even not to be excited. For that reason, in the following comparative analyses, the natural frequencies presented are the ones that are present in both comparing cases and identified as corresponding to the “same” modes.

Although the JIM is symmetrical around its Z axis, some special features of the design create a slight difference on the static stiffness in the two directions. Additionally, the JIM is mounted on the table of the machine, therefore the response acquired on the work piece is actually receiving contributions from the table which is also not symmetrical along the X and Y axis.

![Figure 29. Receptance FRFs from the work piece, untreated configuration](image)

Table 3 presents the natural characteristics of the system for the untreated configuration in the X direction.
An increase in pre-load on the interface has different effects depending on the mode examined. This further illustrates how inherent non-linearities of machine tool components inhibit the mapping of their natural characteristics under loads, as discussed in the beginning of this thesis. With regards to dynamic stiffness, an increase of pre-load causes an increase for some modes, while for others the system becomes more compliant.

![Dynamic Stiffness, Untreated configuration](image-url)

**Figure 30. The effect of pre-loading on dynamic stiffness, X direction, Untreated configuration**
This can be easily understood if we think the JIM as a system of masses connected with springs. An increase in pre-load on an interface is equivalent to increasing the stiffness of that spring and decreasing damping. Depending on how large the change in these characteristics is, the dynamic stiffness might increase or decrease. Additionally, when the spring’s stiffness is increasing, the movement around the associated connection point becomes harder to excite. At the same time, some other spring (i.e. interface) now becomes a weaker elastic link in comparison to the previous spring and excitation energy is easier to cause motion around a connection point corresponding to that interface, therefore that mode now shows lower dynamic stiffness. Therefore by changing the stiffness on one interface, other modes of the structure can be affected, even though the structural characteristics of the components causing that mode are not altered. This is simply because of the load-displacement characteristic of the springs being non-linear.

**Figure 31. The effect of pre-loading on damping (percentage of critical damping $\xi$), x direction, untreated configuration**

With regards to damping, the tests showed that for most of the modes damping decreased with an increase of JIM’s pre-loading, which verifies that in this configuration, the primary damping mechanism is friction damping. As mentioned earlier, interfaces contribute to damping by a friction mechanism between contacting components. As pre-load increases this motion becomes more constrained, leading to a less efficient energy dissipation mechanism and therefore decreased damping. This further explains how for some modes, even though the system becomes more rigid with the increase in pre-load their
dynamic stiffness is decreasing e.g. for the modes 2 and 3, as a consequence of lower damping.

6.3.2 EFFECT OF PRE-LOADING ON THE STRUCTURAL CHARACTERISTICS – VEM CONFIGURATION

The results of the examinations in the VEM configuration are presented below. In terms of rigidity, an increase in pre-load leads to an increase in static stiffness as shown in Table 4

<table>
<thead>
<tr>
<th>X direction</th>
<th>Y direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low Pre-Load</td>
<td>High Pre-Load</td>
</tr>
<tr>
<td>29070</td>
<td>49504</td>
</tr>
</tbody>
</table>

Whereas in the untreated configuration the increase is 132% and 62% for the X and Y directions respectively, in the VEM configuration it is 70% and 24%, respectively. This indicates that in the presence of the VEM layers, much of the available force is consumed in compressing the material rather than actually making the structure more rigid.

Figure 32. Receptance FRFs from the work piece, VEM configuration
Table 5. The effect of pre-loading on natural characteristics, VEM configuration, X direction, left: Low Pre-Load Configuration, right: High Pre-Load Configuration

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (HZ)</th>
<th>Damping (ζ)</th>
<th>Dynamic Stiffness (N/mm)</th>
<th>Frequency (HZ)</th>
<th>Damping (ζ)</th>
<th>Dynamic Stiffness (N/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>279</td>
<td>2,37%</td>
<td>3597</td>
<td>322</td>
<td>1,99%</td>
<td>19608</td>
</tr>
<tr>
<td>2</td>
<td>312</td>
<td>1,88%</td>
<td>3937</td>
<td>488</td>
<td>2,75%</td>
<td>6623</td>
</tr>
<tr>
<td>3</td>
<td>353</td>
<td>1,73%</td>
<td>2874</td>
<td>503</td>
<td>2,68%</td>
<td>5525</td>
</tr>
<tr>
<td>4</td>
<td>438</td>
<td>3,52%</td>
<td>20747</td>
<td>696</td>
<td>3,41%</td>
<td>10000</td>
</tr>
<tr>
<td>5</td>
<td>486</td>
<td>2,07%</td>
<td>16611</td>
<td>816</td>
<td>0,76%</td>
<td>36900</td>
</tr>
<tr>
<td>6</td>
<td>593</td>
<td>1,56%</td>
<td>49505</td>
<td>951</td>
<td>1,15%</td>
<td>124533</td>
</tr>
<tr>
<td>7</td>
<td>848</td>
<td>2,00%</td>
<td>208768</td>
<td>1127</td>
<td>2,69%</td>
<td>416667</td>
</tr>
<tr>
<td>8</td>
<td>1100</td>
<td>0,84%</td>
<td>124533</td>
<td>1237</td>
<td>0,63%</td>
<td>80645</td>
</tr>
<tr>
<td>9</td>
<td>1401</td>
<td>0,57%</td>
<td>214592</td>
<td>1593</td>
<td>0,28%</td>
<td>130548</td>
</tr>
<tr>
<td>10</td>
<td>1561</td>
<td>0,24%</td>
<td>520833</td>
<td>1711</td>
<td>1,23%</td>
<td>689655</td>
</tr>
<tr>
<td>11</td>
<td>1920</td>
<td>0,29%</td>
<td>24155</td>
<td>1989</td>
<td>1,34%</td>
<td>72993</td>
</tr>
<tr>
<td>12</td>
<td>2557</td>
<td>0,61%</td>
<td>230947</td>
<td>2574</td>
<td>0,40%</td>
<td>469484</td>
</tr>
</tbody>
</table>

With regards to dynamic stiffness, the behaviour is different to the untreated configuration. An increase in pre-load will increase dynamic stiffness for most of the identified modes.

Figure 33. The effect of pre-loading on Dynamic Stiffness, VEM configuration, X direction
In the case of damping, it can be seen that for most modes damping is increasing with pre-loading for 5 out of the 12 modes. This, together with the fact that dynamic stiffness increases with pre-load indicates that for some modes the main damping mechanism is not friction damping as previously but structural damping from the viscoelastic material. Furthermore it seems that for these modes pre-load is favourable for the damping capacity of the system. It has to be mentioned at this point, that the VEM layers were applied only on one interface, therefore damping remains of frictional nature in all other interfaces. Therefore a decrease in damping in modes originating from these interfaces is expected.

![Figure 34. The effect of pre-loading on damping, VEM configuration, X direction](image)

**6.3.3 EFFECT OF PRE-LOADING ON THE STRUCTURAL CHARACTERISTICS – COATED CONFIGURATION**

The results from applying the coating on the interface are shown in the following tables and figures.

**Table 6. Static Stiffness (N/mm) for different pre-loading and direction**

<table>
<thead>
<tr>
<th></th>
<th>X direction</th>
<th>Y direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low Pre-Load</td>
<td>59880</td>
<td>91743</td>
</tr>
<tr>
<td>High Pre-Load</td>
<td>103950</td>
<td>100502</td>
</tr>
</tbody>
</table>
In terms of static stiffness, similarly to the other configurations, it increases with pre-loading, while the FRF is exhibiting again a shift to the right. Table 7 exhibits that in the Y direction, the increase of stiffness is marginal while in the X direction is around 74%.

Table 7. Effect of pre-load on natural characteristics, Coated Configuration, X direction, left: Low Pre-Load, right: High Pre-Load

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
<th>Damping (ζ)</th>
<th>Dynamic Stiffness (N/mm)</th>
<th>Frequency (Hz)</th>
<th>Damping (ζ)</th>
<th>Dynamic Stiffness (N/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>479</td>
<td>3,89%</td>
<td>5319</td>
<td>613</td>
<td>2,11%</td>
<td>4854</td>
</tr>
<tr>
<td>2</td>
<td>545</td>
<td>3,01%</td>
<td>20661</td>
<td>697</td>
<td>1,79%</td>
<td>24450</td>
</tr>
<tr>
<td>3</td>
<td>867</td>
<td>0,87%</td>
<td>52910</td>
<td>877</td>
<td>0,76%</td>
<td>75188</td>
</tr>
<tr>
<td>4</td>
<td>1015</td>
<td>1,40%</td>
<td>80645</td>
<td>1303</td>
<td>0,47%</td>
<td>151976</td>
</tr>
<tr>
<td>5</td>
<td>1173</td>
<td>0,62%</td>
<td>37736</td>
<td>1370</td>
<td>0,57%</td>
<td>21277</td>
</tr>
<tr>
<td>6</td>
<td>1703</td>
<td>0,52%</td>
<td>59880</td>
<td>1749</td>
<td>0,86%</td>
<td>109290</td>
</tr>
<tr>
<td>7</td>
<td>1742</td>
<td>0,44%</td>
<td>74074</td>
<td>1791</td>
<td>0,69%</td>
<td>60976</td>
</tr>
<tr>
<td>8</td>
<td>1875</td>
<td>0,28%</td>
<td>196850</td>
<td>1998</td>
<td>0,26%</td>
<td>150830</td>
</tr>
<tr>
<td>9</td>
<td>2184</td>
<td>2,19%</td>
<td>202840</td>
<td>2240</td>
<td>0,21%</td>
<td>446429</td>
</tr>
<tr>
<td>10</td>
<td>2423</td>
<td>0,63%</td>
<td>147275</td>
<td>2509</td>
<td>0,81%</td>
<td>170648</td>
</tr>
<tr>
<td>11</td>
<td>3122</td>
<td>1,07%</td>
<td>884956</td>
<td>3162</td>
<td>1,02%</td>
<td>952381</td>
</tr>
</tbody>
</table>

In terms of dynamic stiffness, an increase in pre-loading leads to an increase in dynamic stiffness for most of the natural frequencies. In terms of damping, the behaviour of the system is similar to the behaviour in the untreated configuration; most of the natural frequencies exhibit lower damping with an increase or pre-load and this is verifying the assumption that in both configurations the damping mechanism is that of friction between the two interfacing components, with the coating adding to that its inherent damping capability.
Figure 35. Receptance FRFs, coated configuration

Figure 36. Effect of pre-load on natural characteristics, coated configuration, X direction
6.3.4  THE EFFECT OF ENHANCING DAMPING

In the previous section, the effects of altering the pre-load on the JIM’s natural characteristics were analyzed for every configuration separately. In the present section, the effects of these configurations are discussed with respect to the applying enhanced damping treatment.

Figure 38 shows the effect of the damping treatment on static stiffness. The application of VEM has significantly detrimental effect on static stiffness, where even in the high pre-load configuration the system is 50-60% less rigid that the untreated configuration. The application of the coating, at lower pre-load leads to an increase of the JIM’s rigidity, although in the high pre-load configuration the application of the coating leads to a decrease of stiffness, although not to extent exhibited by the VEM. It is known that rigidity is increasing with pre-load because surface anomalies are deformed, leading to an increased contact area. The fact that at low pre-load the coated configuration is more rigid than the untreated one can be explained by the fact that the coating material has lower stiffness than aluminum (Young’s modulus around 40 GPa) allowing for higher deformation of surface roughness in low pre-load. At higher pre-load when surface asperities have been compressed, the coating material’s reduced stiffness (as compared to the metal-to-metal contact) leads to a reduction in the structure’s static stiffness.
Figures 39 and 40 present a comparison of dynamic stiffness between the damping configurations. In the case of VEM an observation of the local minima of the dynamic stiffness curve shows that they are in most cases higher than their respective minima for the untreated configuration. In the case of the coated configuration, the picture is mixed with some local minima showing a higher dynamic stiffness. In both VEM and Coated configuration at higher frequencies the system seems to have higher dynamic stiffness and some modes are either shifted to the left of the spectrum disappear in the presence of damping enhancement.

Figure 39. Dynamic stiffness comparison of untreated and VEM configurations, X direction, measured data
6.4 RESPONSE OF THE SYSTEM TO MACHINING EXCITATIONS

The following sections present the findings of the investigations carried out in order to identify the effect of altering the control parameters and therefore natural characteristics, as described in the previous chapter on the response of the system to machining excitations.

Figure 40. Dynamic Stiffness comparison of untreated and coated configurations, X direction

Figure 41. The machining setup.
6.4.1 **EXPERIMENTAL SETUP**

The machining tests were carried out in a Hermle C50 5-axis milling machine. Acceleration signals were recorded by two tri-axial accelerometers mounted on the spindle bearing and the work piece respectively. For the following analyses the work piece acceleration signals are presented, since they were closer to the process and the examined interface, therefore allowed for better monitoring of the response and the effects of the control parameters on it. Furthermore the spindle accelerometers contained signal that was relevant to the machine’s components (bearings, motor, control frequency, etc.) which were of not adding value for this work.

A 16 mm solid carbide end mill with 4 teeth was used, with a 3.2 length to diameter ratio. The cutting speed was 150 m/min, at 2984 RPM resulting in a tooth engagement frequency of 199 Hz. Feed was 0.05 mm/tooth. The process was down milling with the feed direction along both YZ and XZ planes. Axial depth of cut was 10 mm and the radial depth of cut was varied starting from 1 mm and increasing until the process became unstable. The aim was to identify the differences in the response of the system to machining as a result of the changes in the control parameters described previously, and examine if the stability limit can be affected.

It has to be mentioned at this point that the results presented further on are regarding the cases where the feed was along the YZ plane, because only in that feed direction the cutting process became unstable (within the examined depth of cut ranges). As mentioned earlier, the mode excited during cutting was the 774 Hz natural frequency and an observation of Table 8 shows that along the Y direction, the mode exhibits lower damping and higher compliance. This difference in the natural characteristics can explain why the system is more sensitive with regards to instability in this feed direction and highlights how important the structure’s influence is in the dynamic response of the system, even for a mode that is common in both directions.

6.4.2 **STRUCTURAL CHARACTERISTICS OF THE TOOL USED**

Six points were tested on the tool to identify its natural characteristics. Figure 42 exhibits the receptance FRFs for the measurement point at the tip of the cutter
for both directions, while table 3 presents the data for the first five natural frequencies of the tool.

Table 8. Static, stiffness, eigen frequencies of the tool and damping as percentage of critical damping (ζ)

<table>
<thead>
<tr>
<th>Mode</th>
<th>X direction</th>
<th>Y direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mode 1</td>
<td>289 Hz, 4.75%</td>
<td>Mode 1: 289 Hz, 5.63%</td>
</tr>
<tr>
<td>Mode 2</td>
<td>519 Hz, 4.00%</td>
<td>Mode 2: 513 Hz, 3.52%</td>
</tr>
<tr>
<td>Mode 3</td>
<td>774 Hz, 3.12%</td>
<td>Mode 3: 692 Hz, 3.03%</td>
</tr>
<tr>
<td>Mode 4</td>
<td>962 Hz, 3.98%</td>
<td>Mode 4: 774 Hz, 2.30%</td>
</tr>
</tbody>
</table>

As it will be obvious from the analysis of the machining experiments further on, the eigenfrequency of 774Hz is of particular interest since the chatter frequency identified is close to this eigenfrequency. This mode is common for both X and Y direction and is originating not from the cutting tool itself but the whole tool holder-cutter structure as it can be seen from Figure 43.

Figure 44 presents the SLD for the tool used, calculated for the 774 Hz natural frequency. The diagram was calculated based on the FRF shown earlier, for specific cutting force $K_c$=1400 and 10 mm axial depth of cut. According to the SLD, the stability limit is at about 1 mm depth of cut for this cutting operation. The cutting parameters chosen initially were around 3500 RPM corresponding to 175 m/min cutting speed, where according to the SLD, the system should be excited with depth of cut higher than 1 mm. In reality this was not the case, as the process was stable in that area. The process parameters where the process became unstable were identified around 2900-3200 RPM with
2984 RPM being the case where the phenomenon was more intense as it will be described further on.

![Figure 44. SLD for the tool used](image)

6.4.3 RESPONSE OF THE SYSTEM IN THE UNTREATED INTERFACE CONFIGURATION

The results of the machining experiments regarding the response of the system are presented below. In the untreated configuration, in both pre-load configurations the stability limit was identified at 1.5 mm radial depth of cut. Figure 45 presents the response of the system in the frequency domain.

The chatter frequency in the low pre-load configuration was identified at 823 Hz, while no change in the chatter frequency was observed in the high pre-load configuration. Although the stability limit was not affected by the change in pre-load, in the low pre-stress configuration the chatter phenomenon was more profound and the amplitude at the chatter frequency was significantly higher in the low pre-load configuration, as seen in Figure 45.
Furthermore, in the low pre-load configuration, at 1 mm depth of cut the chatter frequency is present, although no sound indicative of chatter was detected. This means that instability is on its onset and this indicates that the critical stability limit is slightly lower compared to the high pre-load configuration, where no such frequency is present. In this configuration, higher rigidity as a result of higher pre-load seems to be beneficial for the response of the system with regards to chatter even though damping is reduced with pre-load. Additionally, an examination of the vibration amplitudes in the time domain presented in Figure 46 shows that increased pre-load improves the response of the system.

The results from the modal tests showed that the system in the high pre-load configuration exhibited lower damping and slightly lower dynamic stiffness in that frequency range. Based on those findings, it would be expected that the system’s response would deteriorate with increased pre-load. However, as observed from the machining tests this is not the case.

These facts indicate that the loss of rigidity in the low pre-load configuration is to such an extent that cannot be compensated by the damping capacity of the system. This naturally leads to the conclusion that there is a threshold of rigidity below which the system is “too compliant” for damping to have an effect.
Figure 46. Acceleration signal, time domain, Untreated Configuration, X direction

Figure 47 exhibits that after using a small portion of the available pre-load, the effect of pre-loading on stiffness is not as profound (the shift of frequencies between the red and black curve is noticeably higher than the shift between the black and the blue and purple curves) which supports the notion of a stiffness threshold which is achieved with a small portion of the available pressure on the interface.

Figure 47. Receptance FRFs for different levels of pre-load, Untreated configuration, X direction.

Furthermore, the comparison of the response for 78% of the available pre-load and the maximum pre-load presented in Figure 48 shows that with higher pre-load (and therefore lower damping) the system exhibited higher vibration...
amplitudes around the chatter frequency. This further supports the concept of the rigidity threshold that has to be satisfied in order to alter the response by altering friction damping.

An examination of the responses presented in Figure 49 reveals increased vibration amplitudes for frequencies 623 Hz and 724 Hz in the high pre-load configuration. These frequencies are very close to the modal frequencies for modes 1 and 2 of the JIM-work piece subsystem and for which damping was decreased with increased pre-load. This further indicates that in the untreated configuration, given that the system is rigid enough, control of the pressure on the interface can alter the response of the system through its effect on friction damping.
Regarding the system’s response to forced vibration, an examination of the vibration amplitudes in the tooth engagement frequency and its first harmonic shows that in the high pre-load configuration the system exhibits improved response as a result of its increased static stiffness.

<table>
<thead>
<tr>
<th>Tooth Engagement frequency (199 Hz)</th>
<th>Low Pre-Load</th>
<th>High Pre-Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st harmonic (398 Hz)</td>
<td>0.22g</td>
<td>0.05g</td>
</tr>
<tr>
<td></td>
<td>0.97g</td>
<td>0.20g</td>
</tr>
</tbody>
</table>

6.4.4 RESPONSE OF THE SYSTEM IN THE VEM CONFIGURATION

The results of the machining experiments regarding the response of the system are presented below. In the VEM configuration, the stability limit was identified at 2.5 mm radial depth of cut in the low pre-load configuration and at 3 mm in the high pre-load configuration.
The positive effect of pre-load in the response of the system, can be attributed to the increased dynamic stiffness and damping of the system in that frequency range, as shown in the modal tests. This is in line with the fact that the VEM interface requires a level of pre-load in order to work in shear. In the low pre-load configuration, the chatter frequency is 857 Hz, while in the high pre-load it is 836 Hz. This phenomenon will be discussed further on in the conclusions of this chapter.

Interestingly, the stability limit was doubled in comparison to the untreated configuration, although the results from the modal analysis did not show levels of damping and dynamic stiffness that could point to such an improvement. This could be explained by the fact that for a VEM layer configuration to exhibit high
damping, significant strain energy much be introduced into the system [92] while for polymeric materials, dynamic stiffness is dependant to vibration amplitudes [13]. In comparison to modal tests, the strain developed on the interface during a cutting process is significantly higher and so are vibration amplitudes. Therefore the energy dissipation mechanism could be more effective in such a case.

With regards to forced vibrations, the system did not show any significant change in its response to excitations when pre-load was increasing.

Figure 51. Response of the system at chatter, 3mm radial depth of cut, red/top: low pre-load, blue/bottom: high pre-load, black: work piece receptance FRF, VEM configuration, X direction

With regards to forced vibrations, the system did not show any significant change in its response to excitations when pre-load was increasing.
6.4.5 RESPONSE OF THE SYSTEM IN THE COATED CONFIGURATION

The coated configuration was subjected to the same procedure of increasing radial depth of cut until chatter is reached in both pre-load configurations. In the case of the low pre-stress, chatter manifested at 2.5 mm radial depth of cut at a frequency of 829 Hz, although the intensity of the phenomenon was limited as the distinctive chatter sound was not intense. In the high pre-loading configuration, chatter manifested with similar intensity at 2 mm radial depth of cut at the same frequency. The application of the coating, therefore increased the stability limit by 1 mm in the low pre-load configuration and 0.5 mm in the high pre-load configuration. Additionally the intensity of the phenomenon was more pronounced in the untreated configuration compared to the coated one, comparing the vibration amplitudes.
At the high pre-load configuration, a test was made for 2.5 mm depth of cut and the intensity of the phenomenon, i.e. the sound emitted from the process was significantly higher, as seen also by the amplitude of vibration for the chatter frequency. As described in chapter 7, the increase in pre-load causes the damping behaviour of the JIM to deteriorate in the coated configuration. This can explain why the stability limit decreases in the high pre-load configuration.

Although the decreased preload is favourable in terms of chatter, an observation of the tooth engagement frequency and its 1st harmonic show that with increased pre-load the amplitudes decrease (Table 9). This can be attributed to the increased static stiffness of the high pre-load configuration which decreases the compliance of the system with regards to forced vibrations.
Table 9. Vibration amplitude, tooth engagement frequency and its 1st harmonic, X direction

<table>
<thead>
<tr>
<th></th>
<th>Low Pre-Load</th>
<th>High Pre-Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tooth Engagement frequency (199 Hz)</td>
<td>0,08g</td>
<td>0,04g</td>
</tr>
<tr>
<td>1&lt;sup&gt;st&lt;/sup&gt; harmonic (398 Hz)</td>
<td>0,72g</td>
<td>0,21g</td>
</tr>
</tbody>
</table>

Figure 54 presents the signal acquired in the time domain for the two pre-load configurations at the maximum depth of cut reached. It is obvious that the response of the system deteriorates in the presence of chatter when pre-load is increasing, since the acceleration amplitudes are increasing with pre-load, despite the increase in rigidity.

Figure 54. System response for both pre-load configurations (coated), 2.5 mm depth of cut, X direction

Figure 55 presents a comparison of the two pre-load configurations at 2.5 mm depth of cut. From that figure it can also be seen that another frequency band around 493 Hz is excited in the low pre-load configuration, which is close to the first mode of the JIM. In the high pre-load configuration this natural frequency shifts to 625 Hz and it can be seen from the figure that when pre-load is increasing, this frequency band becomes excited, showing how a change in the control parameters can shift the excitation frequency. This is another way that the system’s response can be altered, i.e. by altering the pre-load the frequency can be altered and drive the system away from resonance, which could lead to instability.
Furthermore an examination of higher frequency bands (ca. 1750 Hz and 2530 Hz which are close to the first and second harmonic of the chatter frequency) that coincide with natural frequencies of the JIM reveals that the decrease in damping can affect the response of the system along a wide range of the frequency spectrum.

These facts indicate that in the coated configuration it possible to alter the response of the system by altering the pressure on the interface and consequently the natural characteristics of the system.

With regards to the effect of the coating on the response of the system compared to the untreated configuration, it was observed that the stability limit was increased by 1 mm in the low pre-load configuration and 0.5 mm in the high pre-load configuration. Additionally the intensity of the phenomenon was more pronounced in the untreated configuration compared to the coated one.

6.4.6 CONTROL PARAMETER EFFECTS ON STABLE CUTTING CONDITIONS

The following section discusses the effects of the different control parameters on the response of the system in stable cutting conditions. The cutting speed is the kept at the same value as before but the radial depth of cut was 1mm, which
according to the results discussed before is below the stability limit for all configurations. In stable conditions, damping is expected to be of minor importance, while stiffness is expected to be the determining factor.

![Figure 56. Response of the system in stable machining conditions, X direction](image)

**Table 10. Vibration levels for different configurations, data from Figure 56**

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Low Pre-Load</th>
<th>High Pre-Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Untreated</td>
<td>3.02g</td>
<td>1.26g</td>
</tr>
<tr>
<td>Coated</td>
<td>1.98g</td>
<td>1.88g</td>
</tr>
<tr>
<td>VEM</td>
<td>1.44g</td>
<td>1.21g</td>
</tr>
</tbody>
</table>

As seen in Figure 56 and Table 10, in the low pre-load configuration, the application of VEM layers leads to a significant improvement in vibration amplitudes even though it deteriorates the static stiffness of the structure. The coated configuration also shows improved response compared to the untreated configuration. This indicates that in low rigidity conditions, a treatment that restricts motion on the interface affects positively the response of the system.

In the high pre-load configuration the effects of the damping enhancement are becoming less evident since the JIM becomes more rigid. The application of the coating deteriorates the response of the system possibly due to its decreased stiffness compared to the untreated configuration. The application of the VEM
coating however, even though it is causing a reduction of stiffness, still decreases vibration amplitudes and acts like an efficient vibration damping mechanism for ordinary machining conditions. Therefore even in stable cutting conditions where the response is comprising of forced vibrations, enhancing damping can prove beneficial, especially in a low rigidity structure.

It is evident that even in stable cutting conditions, enhancing damping can be beneficial for the dynamic loads that a machine tool can be subjected to, which as discussed in the beginning of this thesis can be detrimental to the performance of the system in the long run.

6.5 CHAPTER CONCLUSIONS

The aim of the work described in this chapter was to examine whether it is possible with that contact conditions setup to affect the natural characteristics of the structure and map the dependencies of its natural characteristics on the control parameters of the interface, i.e. pre-load and damping enhancement. Then it was examined how the process can be controlled as a result of these changes in the structural characteristics.

- Conclusions from EMA and Simulations

In all configurations an increase in pre-load increased the rigidity of the structure as expected and a shift of the response spectrum to higher frequencies (although not in a linear fashion). More specifically, in the untreated configuration, the increase in pre-load led to:

- A reduction in damping for most of the modes
- Increased dynamic stiffness for most of the modes

In the VEM configuration the increase in pre-load led to:

- An increase in dynamic stiffness
- An increase in damping for some modes

In the coated configuration

- An increase in dynamic stiffness
- A decrease in damping for most of the modes

With respect to the effects of enhancing damping, the application of the VEM layers led to an increase in dynamic stiffness despite the significant loss of rigidity and an increase in damping for most of the modes. This indicates that
apart from friction damping, structural damping is introduced into the system in levels that can compensate for the loss of static stiffness.

The application of the coating also increased dynamic stiffness although in the high pre-load configuration, its effects are marginal. With regards to damping the coated configuration exhibits higher damping only for some modes compared to the untreated configuration. In terms of rigidity, in the low pre-load configuration the application of the coating increased static stiffness, while in the high pre-load configuration the untreated JIM was more rigid.

What is evident from the previous chapter is the highly non-linear nature of the system which is challenging the notion that higher pre-load on a joint interface will increase dynamic stiffness. Static stiffness increases and possibly dynamic stiffness in regions between natural frequencies but, as shown, not necessarily close to natural frequencies, which are of interest when it comes to machining stability. Especially, in the case of friction damping it was shown that it is often the case that the benefits of high rigidity are nullified by the significant decrease in damping. Only after characterization of all the modes (or at least the most dominant, or interesting for each application) can the behaviour of the system be described.

Figures 57 and 58 present the comparisons between the measured and simulated FRFs of the JIM in the untreated and VEM configuration. A comparison of the FRFs generated from the simulations and the ones obtained from the modal tests shows that the simulations are in general managing to capture some of the dominant modes, but suffer in predicting low amplitude ones.

![Figure 57. Comparison of simulated and measured receptance FRFs in the untreated configuration.](image-url)
More specifically, in the case of the untreated contact configuration, the simulations are managing to capture the dominant modes although with some discrepancies in the frequencies, giving a rough estimate of the behaviour of the system and the trends of the effects of pre-load on the natural characteristics of the system. Additionally, the simulations seem to overestimate the amplitudes of the natural frequencies, possibly due to a reduced effect from interface-induced friction damping, as a result of the model reduction necessary for keeping the computational time reasonable. In the case of the untreated configuration, the changes in dynamic stiffness as pre-load changes are following the trends observed from the EMA examinations.

![Comparison of simulated and measured receptance FRFs in the VEM configuration](image)

**Figure 58. Comparison of simulated and measured receptance FRFs in the VEM configuration**

In the case of the VEM configuration, many of the low amplitude modes are not identified, while in the lower frequencies, the damping effect is overestimated which could be the reason why many of these modes are not emerging in the FRF. In the range around the 1st natural frequency (as identified by the EMA) for example, a mode is identified but with very high damping ratio. Nevertheless, some of the dominant peaks were observed in the FRFs.

**Conclusions from machining experiments**

Examining the results with regards to damping enhancements, it was shown that the use of VEM increased the stability limit up to 1.5 mm while the use of the coating up to 1 mm. Both configurations improved the response to forced excitation when pre-load was low, however at increased pre-load the effects of
enhancing damping were marginal, with only the VEM exhibiting a beneficial effect.

Examining the results with regards to the effects of pre-loading, in the case of the untreated an increase of pre-load led to an improvement of the response of the system. This behaviour indicated that the system was too compliant and therefore any increase of pre-load would improve the response as it would reduce the relative displacement between the tool and the work piece. The examinations hinted towards a concept of a rigidity threshold which has to be satisfied (within a given damping capacity of the system) in order to alter the response of the system by altering the level of friction damping in the system. Furthermore the frequency content of the response changed as a reduction in damping, caused by an increase in pre-load allowed for excitation of natural frequencies of the JIM.

In the case of the VEM configuration the stability limit increased by 0.5 mm at high pre-load compared to the low pre-load configuration. Additionally, the higher pre-load configuration exhibited lower vibration amplitudes around the chatter frequency. This behaviour could be attributed either to the increase in stiffness which made the system overcome the necessary threshold, either to the increase in damping that was observed in the modal tests. In any case dynamic stiffness was in general increasing with pre-load and this can explain the behaviour of the system subjected to machining.

In the coated configuration, an increase in pre-load led to a deterioration of the response due to the deterioration in damping, with the stability limit decreasing as pre-load increases. Additionally with higher pre-load, one of the excited natural frequencies exhibited a shift to the right of the spectrum which means that the system can be driven out of resonance with a change in the joint’s characteristics.

An examination of the chatter frequencies with respect to static stiffness across all configurations shows that as stiffness increases (due to pre-load increase) the chatter frequency decreases, with the VEM low stiffness configuration exhibiting a chatter frequency that is 34 Hz higher that the chatter frequency at the untreated, high preload configuration. From the stability lobe theory [93], [94] it is known that:

\[ T \omega_c = \epsilon + 2k\pi \]  

(6.3)
where $T$ is the tooth passing period, $\omega_c$ is the chatter frequency, $\epsilon$ the phase shift between inner and outer modulations of the chip thickness and $k$ an integer.

As $T$ remains constant, since cutting speed is constant, an increase in $\omega_c$ means that the phase shift between the inner and outer modulations is increasing. This shows that the importance of the interaction of all the system’s components in what ultimately becomes the response of the system. By changing the stiffness of the structure close to the process, the characteristics of the chip thickness modulation that cause the chatter phenomenon are changing. This means that by adjusting pre-load and therefore stiffness even on an interface that is not the source for instability, this modulation can be minimized in order to reduce the intensity of the regenerative effects.

In terms of the overall scope of the thesis it was possible to exhibit that by altering the structural characteristics of the JIM, either by controlling pre-load on the interface or by enhancing damping, it was possible to control process from a dynamic behaviour point of view.
7 TURNING PROCESS CONTROL THROUGH TUNING OF THE JOINTS’ CHARACTERISTICS

As described in the milling case, an increase of the normal load on the controllable interface of the JIM, will cause an increase in static stiffness and a decrease in damping. In this manner it was possible to control the response of the system which is ultimately reflected on the accuracy of the machined work piece. The following chapter describes the experiments carried out in order to examine if this concept could function in the case of a turning process. In this case, the Joint Interface Module was incorporated in the tailstock of a lathe as shown in Figure 59.

Figure 59. The Joint Interface Module incorporated in the tailstock of the lathe. 1: Stack of two piezoelectric actuators, 2: Upper part of the JIM, 3: Intermediate Plate, 4: Lower part of the JIM, 5: Actuator positioning components

The JIM consists of three components, creating two controllable interfaces; the upper component (2) in which the quill mechanism is residing, the intermediate plate with a determined design (3) and the lower component (4) which is mounted on the machine body. Two stacks of piezoelectric actuators were mounted on the sides of the tailstock (1) and as voltage is supplied, the piezoelectric elements are expanding, thus pulling together the positioning
components which are bolted on the upper and lower parts of the JIM. Consequently, the two components of the JIM are pulled together, increasing the load on the two interfaces created by components 2, 3 and 4. In a case similar to the milling work holding JIM, 6 bolts are guaranteeing a minimum pre-load on the interface which amounts to approximately 120 kN.

In this set of experiments, contrary to the case of the milling work holding JIM, only the effects of changing the load on the interface were examined. No enhanced damping material configurations were tested.

7.1 EXPERIMENTAL SETUP

The machining tests were carried out in an AFM TAE-35 lathe, shown in Figure 60. The machining experiments consisted of longitudinal turning processes, on a steel slender bar, clamped between the chuck and the tailstock of the lathe. Spindle speed was kept constant at n=2000 RPM and feed at f=0,2 mm/rev. The bars had a length of 350mm and a starting diameter of 32 mm and were cut according to the following scheme:

<table>
<thead>
<tr>
<th>Pass No.</th>
<th>Initial Diameter (mm)</th>
<th>Depth of Cut (mm)</th>
<th>Final Diameter (mm)</th>
<th>L/D Ratio</th>
<th>Cutting Speed m/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>32</td>
<td>2</td>
<td>28</td>
<td>12,5</td>
<td>176</td>
</tr>
<tr>
<td>2</td>
<td>28</td>
<td>1</td>
<td>26</td>
<td>13,5</td>
<td>163</td>
</tr>
<tr>
<td>3</td>
<td>26</td>
<td>1</td>
<td>24</td>
<td>14,6</td>
<td>151</td>
</tr>
</tbody>
</table>
The system was tested between two extremes of load on the interface, as a result of supplying 0V to the actuators for the minimum pre-load configuration and 150 V for the maximum pre-load configuration. In the maximum pre-load, the actuators can deliver 17 kN of force. This results to a range of pre-load on the interfaces between 120-137 kN.

7.2 MACHINING EXPERIMENTS RESULTS

During machining, at the high pre-load configuration, the first two passes (at diameters of 32mm and 28mm respectively) were stable, however the last pass, which would shape the bar to a final diameter of 24 mm, was unstable, with chatter starting to develop as the tool was moving towards the middle of the bar. For this reason the following discussion will only focus on the response of the system in the last pass. Figures 61 and 62 show the finished bar and the waterfall diagram generated from the microphone signal acquired during this cut.
In the beginning of the cut (region A), the process is stable, as close to the tailstock, the bending stiffness of the system is high while damping is low. As the tool approaches the middle of the bar and bending stiffness is decreasing, the process is becoming unstable (region B) and the chatter develops as the system lacks damping. Soon the process becomes definitely unstable as it reaches the middle of the bar where bending stiffness is minimized (region C). As the cutting tool travels towards the chuck in a higher bending stiffness area, chatter associated vibrations decrease (region D) and soon the process becomes stable again (region E).

In the case of the low pre-load configuration (when 0V was supplied to the actuators), at the same diameter, no chatter was manifesting during the process. Figure 63 shows a comparison of the bars after machining in both pre-load configurations. In a manner similar to the milling work holding JIM, the reduction of the pressure on the interface led to an increase in damping. This damping increase consequently changed the way the system responds to machining excitations, in the way that a previously unstable process became stable. In the high pre-load configuration (upper) where damping is lower, the manifestation of chatter causes the distinctive chatter marks on the surface of the bar, which disappear in the low pre-load configuration (lower) as damping increases.
Figures 64 and 65 present a comparison of the sound signals acquired during machining in the time and frequency domains. The red curve presents the signal in the high pre-load configuration, where chatter is developing when the tool is approaching the middle of the bar (peak in the signal) and as it moves away from it, the phenomenon slowly fades with the process moving back into stability. The blue curve presents the signal in the low pre-load configuration, where the process was stable throughout the whole length of cut as a result of higher damping. The chatter frequency of 560 Hz coincides with the natural frequency of the bar which is around 550 Hz.
Figure 64. Comparison of microphone signals acquired during machining with high pre-load (red) and low pre-load (blue) – time domain. The amplitude of the signal increases sharply as the process becomes unstable.

Figure 65. Comparison of microphone signals acquired during machining with high pre-load (red) and low pre-load (blue) – time domain. The distinctive chatter frequency lies around 560 Hz.

In the first series of experiments, the control was done stepwise from one pass to another. As the diameter was decreasing the JIM pre-load was tuned to higher damping values (and lower stiffness). In this experiment the high damping JIM configuration was initiated only at the onset of chatter; the system was set in the high pre-load configuration and as the process was moving out of stability (distinctive chatter sound staring to appear), the voltage to the actuators was dropped to the minimum in order to examine whether it was possible to bring the process back to stability. It is worth reminding at this point that the
bars used in this test were of a 24 mm final diameter and a 14.6 length-to-diameter ratio.

From the signals shown in Figure 66, it can be seen from the grey curve that the signal amplitude is starting to increase and by dropping the voltage, and therefore adding damping to the system, the response of the system returns to a stable process.

![Figure 66. Comparison of microphone signals acquired during machining in the high pre-load configuration (red) and controlled pre-load configuration when chatter appeared (grey).](image)

Figure 67 shows the machined bar in the high pre-load configuration (upper) and the bar where the interface pre-load was reduced and therefore damping increased (lower). The surface of the lower bar shows marks that resemble the onset of an unstable process, which disappear when the load on the interface is dropped and damping is increased. This capability could prove useful for the way JIMs can be operated; they are not restricted in usage within pre-defined configurations, but coupled with a monitoring and control system they can cope with unexpected deviations in the process response. If for example, a process that is usually stable in the high pre-load (i.e. low damping) configuration is starting to drift towards instability, a monitoring process of operational damping can trigger a reduction of voltage to the actuators, which will release the pressure on the interface and consequently add damping to the system.
7.3 CHAPTER CONCLUSIONS

The concept of controlling the response of the system to machining excitations was tested in the case of a turning process, where a JIM was incorporated in the tailstock of a lathe. Similarly to the case of the milling work-holding JIM, it was possible to change the response of the system by controlling the pre-load on the JIM interface. By reducing pre-load a process that was before unstable became stable as a result of additional damping introduced from the interface. It has to be pointed that this sort of control of the process was realized without changing the process parameters, thus not compromising the accuracy and productivity of the work piece.

As mentioned before, an important characteristic that the JIMs must incorporate is an extent of modularity, which allows them to be easily exchangeable in order to “fit” different process requirements. At this point it is worth mentioning that the installation of the prototype tailstock with the JIM took less than an hour, including the connection with the hydraulic piping, the connection of the actuators to the control system and the alignment of the tailstock and spindle axis. Taking into consideration the fact that the covers of the machine were not reconstructed in order to facilitate the assembly of the
tailstock on the body, this demonstrates its potential as a viable “plug and produce” component for machine tools.

In order to preserve the dimensional and geometrical stability of machined parts, the machining system should be kept at higher stiffness levels and switched to higher damping only when the risk for dynamic instability is imminent.
8 APPLICATIONS AND CONTROL STRATEGIES

The aim of this chapter is to briefly discuss areas of applications of the JIM concept and elaborate strategies for designing and controlling structural characteristics of the machine tool.

First of all, the JIMs must be carefully designed in terms of shape and dimensions. The contact surface on the interface must very carefully controlled; the surface in contact must be maximized and the load on the interface should be controllable in order to achieve high repeatability. Then, the location of the JIM has to be carefully selected. They must be on the path of energy (force) flow and close to the tool-work piece interface. Some JIMs can be used to control the stiffness/damping levels and others to correct the pre-load which can vary during the life cycle of the component. The later might apply in case of movable joints such as spindle bearings or ball screw driver. Thus some JIMs are used for controlling the machining system and others to maintain the precision of the machining system, for instance by tuning the pre-load during a JIM component’s life cycle.

In this thesis, the JIMs presented were located on the work holding side and were stationary joints. However it is possible to introduce JIMs for rotating tool holding applications both passive and active. Such JIMs can be very useful in milling applications with long overhangs, common in 5 axes machining of complex and/or thin walled geometries. Other applications include movable JIMs that can adjust the stiffness and damping of the structure along the tool path as presented in Figure 24. Another focus area for JIMs are tool holders and turrets internal turning processes, where chatter problems often arise due to the limited dynamic stiffness of the system.

Regarding the control strategies for JIMs, these can be distinguished in two levels. The first stage of the control strategy in a machining system with several JIMs is the selection of the correct JIMs depending on the process sequence, the size of the work piece and the capability that the machining system must have, in order to meet the quality requirements for the part’s features.

The second stage has to do with the control of the stiffness and damping of the JIM during the process and it refers to the target values that are set for the JIMs’ control parameters in order to acquire the desired structural characteristics. In general, the stiffness of the structure varies with the relative position of the tool, while the damping is defined mainly by the contact
conditions on the joints. In cases of large forces like in rough machining or turning big components that induce large inertia forces, stiffness is very important in order to avoid deflections of the work piece or changing position with respect to the clamping points. These positional variations will induce geometrical errors in the final machined part. On the other hand, a finishing internal turning operation will require a system that is very stable from a dynamic point of view and the structure will have to deliver higher levels of damping to avoid chatter. The second stage strategies can be executed either as triggered control actions from the process monitoring and control system, or as predefined strategies in cases that we know what the response of the system will be beforehand.

As an example we can take the case of turning a large slender bar. A JIM incorporated in the tailstock of the lathe would be selected at the first strategy level and it would be set to a high stiffness configuration at the second strategy level. At the final finishing pass, when the length/diameter has decreased and the system is prone to chatter, a second level strategy is executed and the system is set to the high damping configuration.

If an internal turning operation has to be performed on the same machine, a JIM in the tooling side can be picked in the first strategy level and be set to high damping configuration as to keep the process stable. These are obvious examples where a preset strategy can be applied. Nevertheless, the same effect can be achieved if the strategies are decided by the control system based on the control inputs.

In the case of movable JIMs, when the stiffness has to be modified in order to cope with static deflections of the system, another input for determining the strategy has to be introduced, that is the position of the tool relative to the work piece. Assuming that a work piece is machined, carrying a deviation in the desired geometry, for example has a barrel shape rather than a cylindrical one. This input can be read by the JIM control system together with the position of the tool from the machine control system. Then, as the depth of cut increases because of the barrel shape of the work piece, a strategy can be deployed, where stiffness and therefore the value of the control parameter for the actuation of the JIM depends on the position of the tool.

In a similar manner, known errors on components like guideways can be compensated by a change in stiffness, again depending on the position of the moving component.
CONCLUSIONS, DISCUSSION AND FUTURE WORK

9.1 CONCLUSIONS AND DISCUSSION

This thesis discusses the necessity of introducing structure-based process control, provided an overview of how this can be achieved by exploiting the nature of the machine tools as assemblies forming joints and presented prototypes (JIMs) that could demonstrate this novel concept.

The introduction of controllable JIMs brings forward above all a necessity for a novel design of machine tools that will allow for this sort of structure-based control; rather than rigidity being the maximization criterion, a design paradigm should be developed that is based on a stiffness-damping optimization (or dynamic stiffness) criterion. The introduction of JIMs with known and controllable characteristics in the machine tool structure will be the result of such a design paradigm and will enable the implementation of such deterministic control.

By such means, the machine tool manufacturers can control the static and dynamic behaviour of the machining system and not only of the machine tool. This allows end users to control the quality and productivity of the machined part and process without introducing auxiliary, ad-hoc systems in the usage stage of the machine tool’s life-cycle. The concept of designing the JIMs as modules requires that actuation; sensing and a degree of intelligence are incorporated as functional blocks at the design phase. This can be a way to overcome the obstacles and expenses of retrofitting sensors, calibrating them, etc. which is extremely important as a strong trend for future machine tools is the monitoring and control functionalities. At the end user, it is only necessary to select a proper JIM corresponding to the work piece and cutting condition selected, and to adjust JIMs to an initial pre-stress.

Focus on such a design paradigm will subsequently drive efforts towards a better understanding of the nature of machine tool joints and how they affect the final accuracy of the machined components. This is necessary not only in order to optimize the design of the JIMs for the various applications but also to draft suitable control strategies for the operation of the JIMs. Additionally, the need
for quick and accurate interchangeability of the JIMs will motivate the design of truly modular machine tools, which has been in the spotlight of manufacturing engineering for many decades.

**RQ 1:** Is it possible to incorporate consciously designed structural joints with controllable internal parameters, which when altered will allow for control of static and dynamic stiffness of the machining system?

From the experiments described in this investigation it was first of all possible to exhibit that by introducing the JIMs it was possible to control the characteristics of the structure (stiffness and damping) through controlling the pre-load on the interface and by introducing damping enhancement on the interface.

**RQ 2:** Is it possible by adapting the dynamic stiffness by means of tuning the joint interface characteristics, to control the response of the machining system to excitation from the process?

By altering these natural characteristics the response of the system could be steered to a certain direction to increase the stability limit or change the amplitude and/or the frequency spectrum of the response and therefore control the response of the system.

In the case of milling, a significant finding in this examination is that even though the chatter phenomenon was correlated with a natural frequency from the tool-tool holder-spindle structure, altering the natural characteristics of the work-holding device led to a change in the response of the system. This strengthens the claim that the whole machine tool structure is contributing to the outcome of the process in a closed loop and that the characteristics of the joints, especially the ones close to the process determine to a great extent this outcome. It is therefore of great importance when studying the dynamic behaviour of machine tools, to take into consideration not just the weakest link in the machining system (i.e. often the tool) but also incorporate the effects of many of the joints in the machine tool.

In the case of turning, a significant additional finding was that it was possible to control the response of the system even during the process when the system was going into the unstable region and avoid the full evolution of the chatter
phenomenon. Furthermore, the repeatability of the control was very high; the outcome remained the same and unaffected even by subsequent disassembling and assembling of the JIM and movement of the whole machine between different positions.

9.2 FUTURE WORK

This work showed how the response can be affected by changing the structural characteristics of the machining system rather than the process parameters. Given that very often dynamic problems in machining originate from tool-spindle structures, the examination of similar controllable interfaces in tool holders is evidently of great interest.

The optimization of the design of the interface together with exploring different ways to apply the pre-loading is of high significance since piezoelectric actuators are not the most suitable devices for machining environments with lots of disturbances. The application of VEM materials also seems to have potential as a means for realizing high damping interfaces and optimization of the layers configuration following the method described by Daghini [57] could prove beneficial for optimizing the interface properties.

However a very important and unexplored area which is necessary for the future application of such devices with controllable dynamic stiffness is to further examine the correlations between the control parameters, the natural characteristics and the response in machining and move from the trial and error approach to a modeling approach. Such an approach should include the functionalities of FEA and multi body simulations, but with focus on modeling the characteristics of the interfaces and simulating how their characteristics would affect the whole structure’s natural characteristics and ultimately what the machining process outcome will be. By having such an approach that will reliably provide information about the response of the machining system by linking the process parameters with the machine tool structure it will be possible to efficiently design JIMs in the design stage and draft control strategies to be executed during the operational phase.
REFERENCES


