Improving Machining System Performance through Designed-in Damping
Modelling, Analysis and Design Solutions

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“You can't connect the dots looking forward; you can only connect them looking backwards. So you have to trust that the dots will somehow connect in your future. You have to trust in something – your gut, destiny, life, karma, whatever. This approach has never let me down, and it has made all the difference in my life.”

Steve Jobs (1955-2011)
Stanford commencement address, June 2005
Abstract

With advances in material technology, allowing, for instance, engines to withstand higher combustion pressure and consequently improving performance, comes challenges to productivity. These materials are, in fact, more difficult to machine with regards to tool wear and especially machine tool stability. Machining vibrations have historically been one of the major limitations to productivity and product quality and the cost of machining vibration for cylinder head manufacturing has been estimated at 0.35 euro per part.

The literature review shows that most of the research on cutting stability has been concentrating on the use of the stability limits diagram (SLD), addressing the limitations of this approach. On the other hand, research dedicated to development of machine tool components designed for chatter avoidance has been concentrating solely on one component at the time.

This thesis proposes therefore to extend the stability limits of the machining system by enhancing the structure’s damping capability via a unified concept based on the distribution of damping within the machining system exploiting the joints composing the machine tool structure. The design solution proposed is based on the enhancement of damping of joint through the exploitation of viscoelastic polymers’ damping properties consciously designed as High Damping Interfaces (HDI).

The tool-turret joint and the turret-lathe joint have been analysed. The computational models for dimensioning the HDI’s within these joints are presented in the thesis and validated by the experiments. The models offer the possibility of consciously design damping in the machining system structure and balance it with regards to the needed stiffness.

These models and the experimental results demonstrate that the approach of enhancing joint damping is viable and effective. The unified concept of the full chain of redesigned components enables the generation of the lowest surface roughness over the whole range of tested cutting parameters. The improved machining system is not affected by instability at any of the tested cutting parameters and offers an outstanding surface quality.

The major scientific contribution of this thesis is therefore represented by the proposed unified concept for designing damping in a machining system alongside the models for computation and optimisation of the HDIs.

From the industrial application point of view, the presented approach allows the end user to select the most suitable parameters in terms of productivity as the enhanced machine tool system becomes less sensitive to stability issues provoked by difficult-to-machine materials or fluctuations of the work material properties that may occur in ordinary production processes.

**Keywords:** Machining performance, Cutting stability, Passive damping, High Damping Interface, Boring bar, Turret.
Preface and acknowledgments

Writing this acknowledgement gives me a chance to thanks the many people who have contributed to this work in one way or another. To just compile a list of names does not feel like enough recognition for their efforts, especially for those who are probably expecting something more personal, as it should be. One way of doing this is basically to start from the beginning. This is an obvious place to start, isn't it? The path that brought me to this point began a long time ago, in my last year of compulsory studies. I had to choose what kind of “scuola superiore” (Italian for high school) to attend; my English teacher, Prof. Baccio Caramelli, was convinced that I had a predilection for foreign languages and tried to persuade my parents to send me to the “Liceo Linguistico” to develop this skill. Practicalities made this choice impossible, since the nearest 'linguistic high school' was a two hour trip by bus from my home-town (the beautiful Quarrata in Tuscany). Also, this school was a private college with very expensive fees and there was no possibility of accessing a scholarship. For these reasons my mother persuaded me to discard that choice and convinced me that the public “Liceo Scientifico”, in nearby Pistoia, was a much better choice – it offered a much broader range of subjects (from Latin to Chemistry, as well as Philosophy and Mathematics) which would give me more choice in the future when selecting university courses. Of course my mother was right; mothers are always right. However Prof. Caramelli was also right and, in fact, I managed to learn three more languages on my own beside my mother tongue, and I hope he will be proud of me up there in Heaven. Thank you Baccio!

At that point, after the “Liceo”, I had attended five years of classes in Latin, math, English, biology, chemistry, Italian, and so on, and I had to choose what kind of University degree to pursue. My mother, again, took up an old notebook from second grade where I clearly stated: “da grande sarò ingegnere”, i.e. “when I’m big I’ll be an engineer”. Who was I to deny the dreams of a seven-years-old me? I pursued the mechanical engineering programme at Florence University, and it felt good, I avoided Latin and all those classical subjects that made me suffer so much during high school. At this point of this acknowledgement I feel that I should definitely thank my mother, Sandra, who, for the second time convinced me to make the right choice. My father, Paolo, my uncles Maurizio and Fernando and my grandfather Renzo also deserve to be thanked as they have been a great inspiration to me.

After a few years of studying and working (as an interior designer in the family business), I was falling into an everyday routine and I felt that I needed a change of air. I heard about the ERASMUS programme and started the application process, planning to spend a year in Copenhagen at DTU, since they had courses in English. The problem was that Florence University had only four positions available there over a hundred applicants. At that point I was so determined that I tried to learn Danish, since this would have been enough to give me first position in the ranking for selection of students for admission. So I ran to the international book-store in Florence and bought a cassette course (yes, at that time CDs were still a rarity). After
having learned how to count to 20 and to bend the verbs “to be” and “to have”, it was time to get into “unit two, page six”. I pressed start on the cassette player and a voice started reading what was supposed to be written on page six. I could not see that, the sound coming out from the headset was an incomprehensible mumbling spoken by a lady who seemed to be speaking with a potato in her throat and a bucket over her head. No, Danish was not my call; it was a sign of destiny and it led to me apply for a place at KTH. It was the first time there was collaboration between the two universities and no other student wanted to be the “guinea pig”, so I got to come to Stockholm skipping the competition. As I said, it must have been destiny, because after just five days in Sweden I got to meet the woman that later became my wife, Anna Karin. After ten months I should have gone back to Italy, but I could not do it, how could I leave the woman I love? No, I decided to stay, but this was not a simple endeavour; the bureaucratic process almost defeated me. If it wasn’t for Ezio Farini, a fellow student in Florence, and Dr. Laura Pierucci (a relative of mine and researcher at Florence University) I would have never got the right papers in time from Florence. On the Swedish side I had the luck of having Rebecca Ljungqvist as coordinator for the exchange students. She was a great help to me in negotiating the KTH bureaucracy so I could stay. I will be eternally grateful to Rebecca.

Being a student far away from home is quite hard. There is the local language to learn. You want to survive with as little money as possible so as not to weigh excessively on your family. Nevertheless my family has and would never left me unsupported. If I need financial or emotional support they have always been there. They have always been present when I have needed them; for anything. So this achievement owes much to their unyielding support.

Eventually, I did my specialization at the Department of Production Engineering and I started my master thesis work with Prof. Mihai Nicolescu as supervisor. He took me under his wing and guided me through the work. He believed in me and gave me the chance of continuing work at the Department as PhD student. During my time here he has always been supportive; he never left me alone in difficult times, and always stood up for me in every situation. Thank you Mihai, you have been like a father when I have been far from home during these years. I know I made you sweat in recent times, towards the end of my PhD studies, but I hope you won’t be disappointed with the result.

At this point I should definitely mention my co-supervisor, Dr. Amir Rashid. I first met Amir during my master thesis work, when he was concluding his PhD, even then he supported me with his significant experience in the field of machining dynamics. During the last six years I have had the honour and pleasure of working together with fantastic people at the Department. Two of them deserve a special mention, Dr. Andreas Archenti and Dr. Anders Berglund. We started this adventure at the same time in the same research group, and have come to be more than colleagues - to define Andreas and Anders as my two “best friends” is an understatement. We have been travelling east and west around the World, while trying to disseminate our research results; we have had great fun together. Thank you both! You have been a rock during these years. I also need to mention Dr. Danfang Chen, Faraze
Asif and Mathias Werner and thank them for the beautiful and insightful discussions about our research work and our social life outside the Department.

I should not forget at this point to thank the rest of the research group, Dr. Ove Bayard, Dr. Mats Bejhem, Jerzy Mikler and Magnus Areskoug and also the new additions to the group, Tomas Österlind, Qilin Fu, Tigist Fetene, Mariam Nafisi and Costantinos Frangoudis. It has been a pleasure to work side by side with you all.

The Italian enclave at the department, Antonio Maffei and Prof. Mauro Onori, deserves a special mention as well. “Grazie”, having the chance to talk freely, in your own language, about research, career, politics and football (not necessarily in this order!) with people sharing the same cultural heritage really makes a difference when you live and work far from your country.

Working with industrial projects gives you plenty of opportunities to meet interesting people outside the sphere of the Department; a very special person who I had the pleasure of meeting was Jan Danielsen. I have never met anybody as ingenious, curious, active and motivating as Jan. Thank you Jan, you are an inspiration to all younger generations.

A special mention should go to Jan “Janne” Stamer, the wonderful technician at our lab. You can ask anything of him and he will create it for you; he is some kind of wizard. Janne has taught me everything I know about practical CNC machining, for this, and everything else he has done for me, I will be eternally grateful.

The last colleague I would like to mention is Dr. Thomas Lundholm. He has been, and still is, a great support during this period of my life. Among other things, he has helped me to get in touch with Swedish culture more than anybody else. Thomas among other initiatives has introduced me to the fabulous world of running, thanks to the tradition of the “fredagsrus” and made me finally appreciate one of the few Latin proverbs I still remember, “MENS SANA IN CORPORE SANO”. I will always be grateful to Thomas for this.

This work would have not been possible without the precious participation and support of Mircona AB, Spirex-tools, ETP Transmission AB, SSAB Oxelösund, Scania CV AB, LEAX and System 3R.

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- **DampComat** (EUREKA financed through VINNOVA)
- **Production 4μ** (EU IP project within FP6)
- **FFI Robust Machining** (financed through VINNOVA)
TIBI DEDICATUM, NEMO, FILI MI
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These are the publications on which this thesis is based:

**Paper I:**

Daghini L., Archenti A., Nicolescu C.M.
"Design, Implementation and Analysis of Composite Material Dampers for Turning Operations"
ICME 2009: International Conference on Mechanical Engineering, Tokyo, Japan, 2009

**Paper II:**

Daghini L., Archenti A., Nicolescu C.M.,
"Design and Dynamic Characterization of Composite Material Dampers for Parting-off tools"
Journal of Machine Engineering, Vol. 10 nr.2 2010, 57-70

**Paper III:**

Daghini L., Nicolescu C.M.
"Influence of the join system turret-boring bar on machining performance of the cutting process"
CIRP 2nd International Conference on Process Machine Interactions, Vancouver, Canada, 2010

**Paper IV:**

Archenti, A., Daghini, L., Nicolescu, C.M.
“Recursive estimation of machine tool structure dynamic properties”
CIRP 4th International Conference on High Performance Cutting, Gifu, Japan, 2010

**Paper V:**

Daghini L., Archenti A., Rashid A., Nicolescu C.M.
“Active alignment chuck for ultra precision machining”
Not appended publications

Daghini L., Nicolescu C.M.
"Influence of inserts coating and substrate on TOOLOX 44 machining",

Daghini L., Nicolescu C.M.
"Characteristics and Stability Analysis of Tooling Systems with Enhanced Damping"
CIRP 1st International Conference on Process Machine Interactions, Hannover, Germany, 2008

Daghini L., Nicolescu C.M.
"Design of Compact Vibration Damping Turret with Hydrostatic Clamping System for Hard to Machine Materials"

Daghini L.
“Theoretical and Experimental Study of Tooling Systems – Passive Control of Machining Vibration”

Kurdve, M., Daghini, L.
“Sustainable metalworking fluid systems: Best and common practice for metalworking fluid maintenance and system design in Swedish industry”
International Journal of Sustainable Manufacturing.
(Accepted for publication)
## Nomenclature and abbreviations

### Abbreviations (in alphabetical order)

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
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<tbody>
<tr>
<td>ARMA</td>
<td>Auto Regressive Moving Average</td>
</tr>
<tr>
<td>CERS</td>
<td>Contactless Excitation and Response System</td>
</tr>
<tr>
<td>CLD</td>
<td>Constrained Layer Damper</td>
</tr>
<tr>
<td>DBB</td>
<td>Damped Boring Bar</td>
</tr>
<tr>
<td>DVA</td>
<td>Dynamic Vibration Absorber</td>
</tr>
<tr>
<td>EMA</td>
<td>Experimental Modal Analysis</td>
</tr>
<tr>
<td>FLD</td>
<td>Free Layer Damper</td>
</tr>
<tr>
<td>FRF</td>
<td>Frequency Response function</td>
</tr>
<tr>
<td>HDI</td>
<td>High Damping Interface</td>
</tr>
<tr>
<td>ODP</td>
<td>Operational Dynamic Parameters</td>
</tr>
<tr>
<td>ODR</td>
<td>Operational Damping Ratio</td>
</tr>
<tr>
<td>OF</td>
<td>Operational Frequency</td>
</tr>
<tr>
<td>RKU</td>
<td>Ross Kerwin Urban</td>
</tr>
<tr>
<td>SLD</td>
<td>Stability Limit Diagram</td>
</tr>
<tr>
<td>TVD</td>
<td>Tuned Viscous Damper</td>
</tr>
<tr>
<td>VE</td>
<td>Viscoelastic</td>
</tr>
</tbody>
</table>

### Nomenclature

#### Chapter 3

- $B$: Flexural rigidity
- $H_i$: Thickness of the $i^{th}$ layer
- $H_{i1}$: Distance between the $i^{th}$ layer and the neutral plane
- $K_i$: Extensional stiffness of the $i^{th}$ layer
- $\varphi$: Flexural angle of the primary plate
- $\psi$: Shear strain angle of the middle layer
- $D$: Distance between the neutral plane of the primary plate and the neutral plane of the complete composite plate
- $g$: Shear parameter
- $G_2$: Shear modulus of the VE layer
- $p$: Wave number

#### Chapter 4.1

- $E$: Young modulus
- $I$: Moment of inertia
- $F$: Static load
- $L$: Overhang
- $\delta$: Deflection at the hanging end of a cantilever beam
- $d$: Beam section diameter
- $\delta_y$: Deflection of tool clamped in hydrostatic clamp
- $\hat{\delta}$: Deflection of tool clamped in conventional screw clamp
Chapter 4.2

\( r, \theta \) Polar coordinates
\( a \) Outer radius of the annular plate
\( b \) Inner radius of the annular plate
\( \rho \) Mass density per unit volume of the annular plate
\( w \) Transvers displacement
\( h \) Thickness of the annular plate
\( Q \) Shear force
\( M_{rr}, M_{\theta \theta} \) Moments per unit length of the orthotropic plate
\( E \) Young modulus
\( \nu \) Poisson ratio
\( \sigma_r \) Stress in \( r \)-direction
\( \sigma_\theta \) Stress in \( \theta \)-direction
\( \varepsilon_r \) In-plane strain
\( \varepsilon_\theta \) In-plane strain
\( \Delta T \) Temperature rise from a undeformed state
\( \alpha \) Coefficient of thermal expansion
\( D \) Bending stiffness
\( M_T \) Thermal moment
\( V_{\text{max}} \) Maximum strain energy
\( K_{\text{max}} \) maximum kinetic energy
\( D_{r\theta} \) Shear rigidity
\( G \) Shear modulus
\( \Delta \) Rheologic operator for the Maxwell model
\( \hat{\sigma} \) Cauchy stress tensor
\( p \) Volumetric stress
\( \hat{s} \) Deviatoric part of the stress tensor
\( \hat{\varepsilon} \) Deviatoric part of the strain
\( I(t - \tau) \) Relaxation shear modulus function
\( \tau_m \) Relaxation time constants of the spring-dashpot pairs in the same branch
\( G_m \) Stiffness of the spring in branch \( m \).
\( G' \) Storage modulus
\( G'' \) Loss modulus
\( \eta \) Loss factor
Chapter 1: Introduction

1.1 Background

Globalization and sustainability are the two major driving forces of today’s market. The globalization process has opened the possibility of generating products and services all over the world, but, at the same time, it has caused a much harder competition between companies. Within the manufacturing branch the trend has been to open or move productive plants to countries with lower labour cost in order to expand the market and lower the production cost at the same time. This plan of action is, however, not always for the best. The goods have to be transported back and forth around the world and the closed or downsized production plants have caused social instability. In this manner, the globalization process has been conflicting with at least two out of three aspects of sustainability [1, 2].

For this reason the Swedish manufacturing industries together with the Swedish Government have been looking for solutions that would allow for high competitiveness in the global market in a sustainable way. This effort has been translated into a multimillion Swedish kronor investment in various research projects. Among these is the FFI-Robust Machining, whose scope is to support industry with practical, fast and reliable methods and tools to evaluate and control the capability for robust machining with respect to product properties and with competitive manufacturing cost. A machining system can be defined as robust when the processes carried out within it are not affected by external or internal disturbances, such as material variations or time factors. Robustness in this case could be specifically described by the ability of the machining system to produce parts with quality (for instance surface roughness) within given tolerances even if, e.g., a tougher material is suddenly introduced.

One way of achieving this is to implement solutions with machine tool components that can enable higher removal rates with unchanged or even improved machining performance, reducing energy and material waste at the same time.

1.2 Thesis scope and aim

Machining system vibration has historically been one of the major limitations to productivity and product quality. A quantification of this problem is shown in a recent study made on cylinder head production within the Renault group. The yearly production is three million parts and the cost for machining vibration has been estimated at 0.35 euro per cylinder head [3].
The literature review presented in chapter 2 shows that most of the research on cutting stability has been concentrating on the use of the stability limits diagram (SLD), and addressing the limitations that this approach has been demonstrating. From a scientific point of view, the SLD approach is definitely interesting because of the challenges it still poses to the researchers. Its industrial application is though very limited due to these challenges still left for researchers to address. On the other hand, research dedicated to the development of machine tool components designed for chatter avoidance has been concentrating solely on one component at the time.

This thesis would like to cover the lack of research work on the effects of a chain of machine tool components appositively designed to improve the machine tool’s ability to withstand cutting instability. The aim of this thesis is therefore to propose a unified concept that takes into consideration a larger part of the machine tool elastic structure and to prove its effectiveness, in order to respond to the industrial need for such a solution.

The scope of this thesis is limited to the analysis and development of components for turning operations.

1.3 Research questions

The fundamental research questions addressed in this thesis concern the following two issues:

1. Theoretical treatment and experimental methodology for exploiting the dynamic properties of existing joints in the machine tool structure to control the overall damping capability of the machining system, possibilities and limitations.

2. Theoretical modelling and design approach for improving machining system performance by using single-HDI configuration and further by using multi-HDI configuration (i.e. distributing damping).

1.4 Summary of the appended papers and personal contribution

Paper I introduces the concept followed in the design of the boring bar, implementing the high damping interface (HDI) making use of VE polymer composites. Moreover the tool performance is compared to a geometrically equivalent conventional tool. The comparison is carried out in two stages: first the experimental modal analysis (EMA) comparing the tools and the clamping technique (screw and hydrostatic clamp) is illustrated and then machining tests at different cutting parameters are presented. The machining test results have been analysed utilizing autoregressive moving average (ARMA) models to capture the operational dynamic parameters. The personal contribution to this paper is the concept development, the design work, the experimental part and its analysis.
**Introduction**

*Paper II* introduces the same concept for a different typology of tool. The tool is employed for an industrial case study, grooving of camshaft, where very conservative cutting parameters are currently employed due to excessive instability. Similarly to the previous paper ARMA modelling has been employed in the evaluation of the machining test results. As with paper I, the personal contribution to this paper is the design work, the experimental part and its analysis.

*Paper III* deals with the Influence of the join system turret-boring bar on machining performance of the cutting process by introducing the computational model employed to evaluate the effect of multilayer VE polymer composite treatment and the design concept of a turret with enhanced damping capability. EMA and comparative machining tests have been used to evaluate the turret and the effect of combining the damped tool and the damped turret. The personal contribution to this paper extends to the whole content.

*Paper IV* deals with the estimation of structural dynamic properties of the machine tool. Currently, the conventional methodology to extract such properties does not take in consideration that these are dependent on the operational speed of the spindle. This paper introduces a contact-less test methodology (CERS) and a recursive algorithm for the extraction of structural dynamic properties during operation. These two methods are compared to conventional EMA. The personal contribution to this paper is limited to the experimental work and the analysis of the obtained data.

*Paper V* introduces the implementation of HDI in the active alignment designed for ultra-precision machining. The chuck functionality is briefly described as well as its thought application within the newly designed production process for optical components. The evaluation of the dynamic properties of the chuck is carried out by EMA and machining tests (fly cutting). The EMA illustrates the functionality of the HDI and the machining tests confirm this in operational conditions. The personal contribution to this paper covers the implementation of the HDI, the experimental work and the analysis of the obtained data.
1.5 Thesis structure

This thesis deals with the research work around the development of machine tool components with enhanced damping capability and it is based on the appended published papers. This thesis also integrates the papers with an extensive review of state of the art within the specific research area and more thorough presentation of the modelling approach.

The literature review on the subject of control of machining vibration is presented in Chapter 2. Chapter 3 introduces the subject of passive control of vibration using viscoelastic polymer metal composites, and discusses the analytical model employed to compute the effect on damping and stiffness of several design parameters in the design of a sandwich plate where multiple layers of VE polymer metal composite material are applied. Chapter 4 describes the principles followed in the design of the different components and thoroughly describes the computational model employed for the design of the damped boring bar (DBB). Chapter 5 will shortly illustrate the methodology used for the performance evaluation of the components and presents the result. The final discussion and the conclusions are presented in chapter 6.
Machine tools are notoriously subject to three types of vibration: free, forced, and self-excited [4]. Free vibrations take place when the stable system is displaced from its equilibrium by an impulse-like excitation; the system vibrates and eventually returns to the starting position according to its structural properties.

Forced vibrations are all those occurring due to dynamic forces applied to a stable system. Generally there are four types of sources that might generate such forces:

1. Alternating cutting forces, such as those induced by inhomogeneities in the workpiece material, break-off of built-up edge or changes in the chip cross section.
2. Interrupted cutting processes, such as milling.
3. Internal sources, such as unbalances in the rotating units.
4. External disturbances transmitted through the machine tool foundation.

Self-excited vibration or chatter is a complex phenomenon and is commonly the least desirable type of vibration as the machine tool structure enters an unstable state. Chatter depends on the design of the machine tool as a whole, on the workpiece material and geometry and on machining regimes; its occurrence is due to insufficient damping in the machine tool structure [4].

The phenomenon of machining vibration, and especially chatter, has been thoroughly studied by many researchers throughout the past and the current century, for instance Lindström [5, 6], Tlusty [7], Tobias [8], Nicolescu [9] and Altintaş [10].

The research around machining vibration control has mostly concentrated on chatter and possible ways to avoid it. This review is divided in four clusters [11]:

1. Research on the computation of the stability limit diagram (SLD) to select chatter-free cutting parameters.
2. Research on In-process strategies for chatter recognition and avoidance.
3. Research on changing the machining system structural behaviour by active means.
4. Research on changing the machining system structural behaviour by passive means.

The following sections will give an overview of these four research areas.

---

Giorgio Vasari
2.1 SLD for out-of-process selection of cutting parameters

This group comprises all those methods that allow stable machining processes by a priori selecting cutting parameter combinations in the stable zone of the SLD exploiting the lobbing effect (see Figure 1). The computation of the SLD implies, however, both in- and out-of-process investigation.

![Stability limit diagram (SLD)](image)

*Figure 1; Stability limit diagram (SLD). Example of exploitation of lobbing effect.*

The out-of-process approach to use the SLD aims to avoid machining vibration by selecting the most appropriate cutting parameters. In order to do this the SLD has to be computed in advance. This approach is based on the work that Tlusty [7] and Tobias [8] have carried out since the early second half of the last century. In order to identify the SLD, the system behaviour has to be modelled either by characterizing or simulating the response of the machine tool elastic structure (machine tool, tool holder, spindle, workholding and all the other structural components of the machine tool) [11]. The transfer function of a multi-degree-of-freedom system can be identified by structural dynamic tests, such as experimental modal analysis (EMA), by exciting the structure with an impact hammer equipped with a force transducer and measuring the response with displacement, velocity or acceleration sensors [12].

The use of accelerometers for carrying out EMA on the machine tool structure is rather common and well accepted within the scientific community. However, Özşahin et al [13] have found that the sensor’s mass can be an important source of error when identifying the SLD (see Figure 2). The authors also present a method to compensate for the accelerometers mass using a laser vibrometer to appreciate the influence of the accelerometers mass.
Rasper et al [14] analysed several possible sources of uncertainty in stability prediction through SLD and came to the conclusion that the major source of uncertainty occurs at the structural analysis stage. The authors concluded that using different excitation equipment (impact hammer or shaker) could result in important differences in the SLD. They also pointed out that, since the structural analysis has to be carried out in an idle machine tool state, neither the effect of the tool position nor the effect of the rotating spindle are taken into account in the SLD. Budak et al [15, 16] have been dealing with the effect of lead and tilt angle on the milling process. The authors found that lead and tilt angles change the chatter behaviour and stability limits and may provide, for the specific case studied in the article, four times increase in absolute stability limit [16]. In [15] the authors took into account tilt and lead angles in the stability prediction, but observed that the measured chatter frequencies were generally lower than the predicted ones and attributed this behaviour to the fact that the structural analysis had been carried out in static condition and the modal frequencies may shift during cutting. It was only after adjusting the modal data that the simulated stability diagrams agreed better with the experimental results.

The influence of the spindle rotational speed on the SLD has also been addressed by several other researchers. Movahhedy et al [17] found that gyroscopic effects lower the critical depth of cut in high speed milling, see Figure 3, and therefore the stability predictions based on stationary FRFs are not conservative when machining is executed at high speed (above 10000 rpm).
Gagnol et al [19] proposed a method for identifying the SLD in a manner that takes into account the effect of spindle speed on the dynamic behaviour of the spindle. However, the authors themselves concluded that the obtained SLD would need further adjustment to better fit experimental results. Cao et al [18] also studied high speed milling stability and obtained the speed-dependent FRFs at the tool tip and used those to identify the SLD. The authors found a significant shift of the lobes towards lower spindle speed (see Figure 4).

Abele et al [20] and Rantatalo [21] found that there is a loss of stiffness in the spindle bearings due to the centrifugal forces acting on each ball of the bearings; see Figure 5(a).
This behaviour obviously affects the calculation of the stability limits [20] causing a mismatch between calculated and experimental limits (see Figure 5(b)), since the structural analysis is carried out at idle state. Abele [20] introduced a modelling methodology for taking this in account, thus obtaining a better match between the experimental and the calculated stability limits. Archenti [22] proposed a solution for carrying out structural analysis on a rotating spindle, in order to overcome this limitation.

A common way to improve chatter avoidance is to use milling cutters with unequal tooth pitch. The idea is to disrupt the regenerative effect. Nevertheless, these tools are not free from chatter and their behaviour is highly non-linear. Sellmeier and Denkena [23] have made an attempt to calculate the stability limits for such a case. They proposed two methodologies for taking into account the non-linear behaviour of such type of tool and they demonstrated that in this case the stability limits appear more as “islands” than lobes (see Figure 6).
Altintaş and Weck [24] have recently reviewed fundamental modelling of chatter in metal cutting and grinding processes, and came to the conclusion that the limitation and therefore the challenge for such an approach is that the “chatter stability is still not solved when the process is highly nonlinear due to time varying and nonlinear cutting coefficients including the process damping at low speeds, and when the structural dynamics of the parts and machines vary along the tool path”. This statement gives a further reason why this approach has not yet been widely implemented in industry. In fact, one may observe that structural dynamics do change along the tool path due to the very nature of the cutting process, as the conditions of the tool-workpiece interface continuously vary either because the workpiece changes shape during machining (although in some cases this might be negligible) or the tool changes its position (such as in 5-axis machining) but also because the workpiece material is not homogeneous in its microstructure and the cutting force drastically changes along the tool path [25]. In most industrial cases all these occurrences take place simultaneously, making this approach inadequate since the SLD is only valid for one specific configuration of tool, tool-holder, spindle and workpiece [11]. Another major issue with this approach is given by the lack of an absolute criterion for distinguishing between forced and self-excited vibration [22]. Nevertheless, this methodology is an invaluable tool for predicting the effect of a change in the machine tool structural design.

Figure 7 summarises the advantages and disadvantages of this approach.
Control of machining vibration – state of the art

Advantages

• Allows for the choice of right cutting parameters in advance

Disadvantages

• Strongly sensitive to structural analysis accuracy
• Only applies to one given tool-workpiece configuration
• Non-linear behaviour difficult to take into account
• Lack of absolute criterion for stability
• End-users need deep knowledge on structural and machining dynamics
• Changing cutting parameters might compromise tool life and therefore productivity

Figure 7: Summary of advantages and disadvantages of using the stability lobe diagram (SLD) to select chatter-free cutting parameters.

2.2 In-process strategies for chatter recognition and avoidance

In-process strategies aim to avoid machining instability by adaptation of cutting parameters during the machining operation. In this case chatter is recognized by analysing signals acquired during machining by sensors like dynamometers [26] or accelerometers [27]. A first strategy for automatic chatter recognition and online modification of the cutting speed to a stable area was proposed by Weck et al [28]. This idea has been further developed by Tlusty et al. [29, 30], where the signal emitted by the cutting process is sensed and used to recognize chatter, and the cutting speed is modified thereafter. Tangjitsitcharoen [31, 32] suggested an in-process strategy for the identification of cutting states based on the power spectral density (PSD) of the dynamic cutting force measured during cutting. Kuljanic et al [33] proposed a multi-sensor chatter detection system that makes use of neural network based classification system. The authors found that different types of sensors were needed depending on the machining operation in order to be suitable for a wider range of applications. Yao et al [34] proposed a methodology based on wavelet and support vector machines, allowing chatter identification before it is fully developed. Nicolescu et al [9, 35, 36] suggested an approach for chatter identification in turning based on auto regressive moving average (ARMA) models, Nicolescu’s ideas have been further developed by Archenti [22, 37, 38, 39, 40], and applied to milling operations. This methodology differs from the previously mentioned ones as the authors established that the operational damping ratio (ODR) might be employed as absolute criterion for distinguishing between forced and self-excited vibration enabling instability identification. In this manner the
approach can also be employed for the evaluation of a structural modification of the machine tool elastic structure (as later in this thesis).

Figure 8 summarises the advantages and disadvantages of the in-process way of action.

![Advantages and Disadvantages of In-process Strategies](image)

**Advantages**
- Parameters are optimized/adapted during process
- No need of structural analysis
- Can be integrated in the machine tool control system

**Disadvantages**
- Chatter recognition might take place when chatter has already occurred
- Requires high computational power
- Research mostly covers the identification part, not many solutions for vibration avoidance

2.3 Changing the machining system structural behaviour by active means

This group includes those methods that avoid cutting instability by altering the system behaviour and modifying the stability limit by active means. The working principle of the active strategies is to monitor the dynamic state of the machine tool system, diagnose an incidence and actively implement an action that would change the system to a more adequate situation. The compensation for the arising dynamic forces is suggested by Harms et al. [41] who designed a tool equipped with piezoelectric actuators and force sensors with interchangeable tool head (see Figure 9). Browning et al [42] also suggested a solution in this direction for boring bars. Rashid et al [43] proposed a similar approach for milling operations, where force sensors and piezoelectric actuators were embedded in the workholding system, decreasing the amplitude of the dynamic force by 70% (see Figure 10).
Albizuri et al [45] explored the possibility of embedding sensing and acting equipment in the screw nut of a centerless grinding machine, attaining a significant reduction of vibration and consequently improving the accuracy of the part produced. In the same fashion, Dohner et al [44] proposed an active vibration cancellation for milling applications by implementing the sensing and actuating equipment in the spindle (see Figure 11). Olgac and Hosek [46] presented a novel methodology for chatter elimination by applying a delayed resonator, i.e. a tuneable frequency vibration absorber formed using a feedback control law on a passive spring-mass damper. However, this technique has only been treated in a theoretical way and has not been tested in practice.
Ganguli et al [47] proposed active damping as a strategy to enhance the stability limits of the system and presented a study of its effect showing the ability of such an approach to enhance the stability limits especially in the low stability regions of the SLD.

The major advantage of the active approach is its adaptability. The major drawbacks are that recognition of instability can only happen when instability has already occurred and that it relies upon the presence of sophisticated sensors, actuators and signal processing units; hence this approach can become very expensive. However, this sort of equipment becomes more and more inexpensive with the technological advances, thus the increasing popularity of the active approach. Figure 12 summarises the advantages and disadvantages of this approach.
Figure 12; Summary of advantages and disadvantages of changing the system behaviour by active means.

Advantages
- No need for structural analysis
- Can be integrated in the machine tool control system

Disadvantages
- Chatter recognition might take place when chatter has already occurred
- Requires high computational power
- Requires sensing equipment and actuators for every component
2.4 Changing the machining system structural behaviour by passive means

Passive strategies are based on the enhancement of the machine tool structure design in order to improve its performance against vibration. The objective of this approach is basically to extend the stable region of machining; Figure 13 illustrates, for instance, the effect of increasing the damping ratio from 0.02 to 0.2 on the SLD for a lathe having a natural frequency of 60 Hz [4].

Figure 13: Effect of the damping ratio on the SLD for a typical single degree of freedom structure with a natural frequency of 60Hz [4].

There are two major approaches for achieving this:

1. By implementation of dynamic vibration absorbers (DVA). The basic principle of this technique is to add a mass residing on a spring and a viscous damper at the point of maximum displacement. This additional single degree of freedom (SDOF) system must have approximately the same natural frequency as the component in order to obtain large relative displacements, and if the viscous damper is properly designed it will dissipate the mechanical energy [48].

2. By introduction of damping. Damping can be introduced either by implementation of layered treatment or by enhancing frictional damping. For these methods to be effective they should be designed to reside where the strain energy is maximal [48].
Rivin et al. [49] suggested a design based on the first approach where the inertia weight was integrated in the tool, hanging on rubber rings. The absorber was tuned by changing the stiffness of the additional system. Another example of application of DVA principle is proposed by Lee et al. [50], the DVA was, in this work, tuned by changing the inertia mass. Rashid [51] has been studying the use of tuned viscous dampers (TVD; a special class of DVA) applied to workholding systems for milling operations. This resulted in decreased average vibration amplitude during machining (see Figure 14).

Figure 14; Time history of vibration amplitudes as measured in X coordinate during milling of the steel workpiece: response of the system without (upper) and with (lower) TVDs. With TVDs, the steady-state maximum reduces from 160 to 110 m/s² [51].

Wang and Lee [52] have been studying a certain milling process and identified the weakest component as being the spindle. In their publication the authors redesigned the spindle introducing a DVA device appositely tuned for the application.

The second approach has been adopted by Rashid, who has been working on implementing integrated damping interfaces for workholding systems for milling operations [53] exploiting the damping properties of viscoelastic polymers. Marui et al [54] proposed to introduce a friction plate within the overhanging part of the tool shank in order to improve the damping capacity of the system by friction during vibration acting between the inner wall of the hole and the inserted plate surface. Ziegert et al [55] proposed to place a multi-fingered insert into a slender end mill. The idea was (similarly to Marui) to increase damping capability exploiting the friction these fingers develop during high speed machining.

Whereas DVA type approach still requires tuning depending on tool overhang, other approaches such as integrated damping interfaces, are free from such preparation work.
Figure 15 summarises the advantages and disadvantages of the passive approach.

Advantages
- Enhances resistance to chatter for given process
- No need for structural analysis
- No need for sensing and/or actuating equipment
- End-user can adopt solution with no need for complicated training

Disadvantages
- Depending on the design may need tuning
- In case of tuning, need for further training of end-user
- Cannot enhance damping as desired without compromising stiffness
- Only applied to one component/process

Figure 15; Summary of advantages and disadvantages of changing the system behaviour by passive means.

2.5 Chapter conclusions

The literature review presented here shows that most of the research on cutting stability has been focusing on the use of the stability limits diagram (SLD), addressing the limitations that this approach has been showing for specific applications. This approach still lacks of an absolute stability criterion, in addition to this, SLD’s are sensitive on the accuracy of the structural analysis; therefore, in order to attain accurate and reliable results, the analysis ought to be carried out by personnel with experience in the field. Further, once the SLD is computed, it only applies for the specific configuration of workpiece (material and geometry) and tool. Finally, neither non-linear behaviours nor structural dynamics variations of the parts and machines along the tool path can be taken into account by this approach, these being the most common occurrences in manufacturing of advanced products.

On the other hand, most effort in improving machining performance by changing the machining system structural behaviour (either by active or passive means) has been solely concentrating on one specific component at the time.
Chapter 3: Vibration damping through multilayer VE-polymer-metal composite treatment

Common to all the appended papers is the application of viscoelastic (VE) polymer damping in the form of high damping interfaces (HDI).

The aim of this chapter is to present a computational model for selecting the right design parameters in order to optimise towards high loss factor maintaining high rigidity.

The concept of passive control of vibration through VE polymers is introduced. In addition to this, the analytical model used in Paper III for computing the effect of multiple layers of VE polymer, of base material type and thickness is described and used to illustrate how different design parameters can affect the damping treatment.

3.1 Passive control of vibration through VE polymers.

The response of a structure to a time-varying input depends on the stiffness, damping, and mass of the structure. Therefore the reduction of the response might be realized by adopting one or several of the following solutions: reducing input, increasing stiffness and mass or increasing damping. In machining the input energy is difficult to control since it depends on the operational conditions. A reduction of the input energy will affect, in most cases, the productivity of that particular machining operation. High stiffness and damping are each necessary, but not individually sufficient requirements for a precision machine [56].

In later years it could be observed that the trend in machine tool design is going towards lightweight structures. This means that vibrations are transmitted with higher intensity. However, low mass can help to increase the controllable bandwidth, but, on the other hand, high mass does attenuate high-frequency vibration [57]. Increasing stiffness would cause a mode to shift upwards in frequency, however, given the random excitation of machine tool structures due to the dynamic cutting force, this solution would not secure a vibration-free machining.

The major restrictions on the implementation of the damping treatment are (i) weight and (ii) the treatment has to be applied without disassembly of the components.

The benefits of passive damping for vibration suppression are well established in various fields in mechanical and civil engineering respectively. Although not always consciously designed in. Machine tool structures, for instance, benefit from passive
damping that comes from friction in fixed and movable joints [4, 58]. As these structures are overdimensioned in terms of strength [22], the main part of elastic deformation occurs in joints. This deformation mechanism can be employed in design for improving the dynamic performance of the machine tool structures through the damping treatment of the joints. A significant increase in damping may be achieved by placing damping material in joints. The advantage of this type of damping is that it requires very little added weight. As far as the accuracy of the machine is concerned, it would be best to find a trade-off between high stiffness and high damping [56].

Engineered passive damping for structures is usually based on one of four damping technologies: viscoelastic polymers, viscous fluids, magnetics, or passive piezoelectrics. Each of these damping mechanisms must be understood in order to select the most appropriate type of damping treatment. All passive damping treatments work by absorbing significant amounts of strain energy from the vibration modes of interest and dissipating this energy through some type of energy dissipation mechanism [59, 60].

Viscoelastic polymers provide high energy dissipation. Viscoelastically damped structures have been successfully applied in many engineering fields, particularly in the aerospace industry [61] generally employing the VE polymer in three different ways: as free-layer dampers (FLD), as constrained-layer dampers (CLD), in tuned viscoelastic dampers (TVD) [62].

FLDs are defined as those dampers composed by a single layer of damping material positioned on top of the base material by gluing or other bonding techniques [62] (Figure 16(a)). The system loss factor of the FLD increases with the thickness, storage modulus and loss factor of the damping layer as computed by Kerwin et al [63, 64].

CLDs consist of a sandwich structure where the damping material is constrained between two layers of elastic material (Figure 16(b)) [62]. Kerwin et al have also studied this configuration and come also to the conclusion that this is far more effective than the FLDs [63]. CLDs have also been studied in patched applications in order to optimize the usage of damping material and enhance damping only for particular vibration modes [65]. TVDs consist of a mass residing on a damping layer bonded to the base material (Figure 16(c)) [62].

Figure 16; (a) Free layer damper, FLD. (b) Constrained-layer damper, CLD. (c) Tuned viscoelastic damper, TVD.
This damper is a sort of dynamic vibration absorber (DVA) were the VE polymer works as both spring and damper. In order to achieve the desired result the TVDs must be accurately designed, since they only act on one natural frequency.

Design of structures using VE polymers requires proper methods to predict the overall damping values expected from various structural configurations. The loss factor and the response of the damped structure to the dynamic load are critical parameters that describe the effectiveness of the damping treatment. The loss factor is a measure of the inherent damping in a material when it is dynamically loaded. It is typically defined as the ratio of energy dissipated in unit volume per radian of oscillation to the maximum strain energy per unit volume. Loss factor damping is sometimes referred to as material or structural damping.

Viscoelastic polymers are polymeric materials whose long-chain molecules cause them to convert mechanical energy into heat when they are deformed, as schematically illustrated in Figure 17.

![Figure 17; Typical polymeric structure](image)

The most important advantage of VE polymers is their high loss factor and low storage modulus. As mentioned above, the loss factor is a measure of the energy dissipation capacity of the material while the storage modulus is a measure of the stiffness of the material. The storage modulus (shear modulus) is important in determining how much energy gets into the viscoelastic, and the loss factor determines how much energy is dissipated.

Both the shear modulus and loss factor of VE polymers are temperature and frequency dependent, though temperature has a greater effect on damping performance (see Figure 18).
Passive damping treatments for complex structures might be designed using, for instance, finite element techniques, taking into consideration both frequency and temperature dependencies of passive damping mechanisms. Damping design is not just the selection of a high loss mechanism (material, device) for the temperature range of interest; it is also an integrated structural and materials design process. To achieve damping, two conditions must be met [59]:

1. significant strain energy must be directed into the high loss mechanism for all modes of interest and
2. the energy in the mechanism must be dissipated.

The first condition requires most of the design effort and is dependent on structural properties, location, mode shapes, stiffness, wave lengths, thickness of material, etc. The second condition is met by selecting the mechanism with the proper loss factor that matches the designed stiffness.

Machine tools are continuously changing their structural configuration to adapt to the process condition. The consequence is the variation of dynamic characteristics in a relatively wide range. Adding to this problem the large variation of excitation energy during the process, it is easy to understand the importance of finding efficient solutions for vibration control.

VE polymers are suited for their employment in passive damping of machining systems. Unfortunately the lack of rigidity and excessive long term creep prohibits the fabrication of a structure entirely from such high damping polymers and one has to resort to a composite construction of metal with damping layers bonded to it.

The following section will introduce the computational model employed to design a multilayer application of VE polymer in sandwich plate configuration.

To estimate how the flexural rigidity and the damping ratio are influenced by the choice of plate material and the number of layers of VE polymer composite applied the analytical solution obtained from RKU equation [63] can be employed.

Equation (1) computes the flexural rigidity of a composite plate formed by a primary plate, a VE polymer layer and a constraining layer (see Figure 19).
\[
B = K_1 \frac{H_1^2}{12} + K_2 \frac{H_2^2}{12} + K_3 \frac{H_3^2}{12} - K_2 \frac{H_2^2}{12} \frac{\partial \psi}{\partial \phi} + \\
+ K_1 D^2 + K_2 (H_{21} - D)^2 + K_3 (H_{31} - D)^2 - \\
+ \left[ \frac{K_2}{2} (H_{21} - D) + K_3 (H_{31} - D) \right] H_2 \frac{\partial \psi}{\partial \phi}
\]

(1)

where \( H_i \) is the thickness of the \( i \)th layer, \( H_{i1} \) is the distance from the centre of the \( i \)th layer to the neutral plane of the primary plate (see Figure 19). \( K_i \) is the extensional stiffness of a unit length of the \( i \)th layer (\( K_i = E_i H_i \)). \( \phi \) is the flexural angle of the primary plate and \( \psi \) is the shear strain angle of the middle layer (see Figure 20).

\[
\frac{\partial \psi}{\partial \phi} \text{ represents } \left( \frac{\partial \psi}{\partial x} / \frac{\partial \phi}{\partial x} \right) \text{ and its general expression is:}
\]

\[
H_2 \frac{\partial \psi}{\partial \phi} = \frac{K_1 H_{31} + K_2 (H_{31} - H_{21})}{K_1 + \frac{K_2}{2} + (K_1 + K_2 + K_3)g}
\]

(2)

\( D \) is the distance between the neutral plane of the primary plate and the neutral plane of the complete composite plate (see Figure 20). The general expression for \( D \) is given by the following equation:

\[
D = \frac{K_2 \left( H_{21} - \frac{H_{31}}{2} \right) + (K_2 H_{21} + K_3 H_{31})g}{K_1 + \frac{K_2}{2} + (K_1 + K_2 + K_3)g}
\]

(3)

where \( g \) is defined as the “shear parameter” and its expression is

\[
g = \frac{G_2}{K_3 H_2 p^2}
\]

(4)

where \( G_2 \) is the shear modulus of the VE layer and \( p \) is the wave number. The loss factor of the composite plate can be extracted from (1) as the imaginary part of the flexural rigidity if all the Young moduli are expressed in their complex form.

The approach suggested by Jones [67] has been employed in order to evaluate the flexural rigidity in case of multiple VE composite layers (see Figure 21). The RKU equation is employed to compute the complex flexural rigidity for the first three layers. These three layers are considered as one layer characterized by the complex flexural rigidity computed in step 1. Step 1 and 2 are repeated until the structure can be treated as a three-layer plate and the RKU equation can be employed for one last time.
Table 1 summarizes the material data used for the computation of the complex flexural rigidity. The VE polymer properties are frequency and temperature dependent; the data relative to 25°C has been extracted from the datasheet provided by the supplier and fitted to a curve. The choice of temperature was made considering that the damping treatment is positioned far enough from the cutting zone to not be affected by the heat arising during machining. Equations (5) and (6) are the obtained expressions for Young modulus and loss factor as functions of frequency.

**Table 1: Material properties.**

<table>
<thead>
<tr>
<th>Material</th>
<th>Young modulus [Pa]</th>
<th>Loss factor</th>
<th>Wave propagation speed [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>$70 \cdot 10^{11}$</td>
<td>$5 \cdot 10^{-4}$</td>
<td>4700</td>
</tr>
<tr>
<td>Steel</td>
<td>$210 \cdot 10^{11}$</td>
<td>$5 \cdot 10^{-4}$</td>
<td>6100</td>
</tr>
<tr>
<td>VE polymer</td>
<td>Frequency dependent according to equations (5) and (6)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
In this analytical model the materials in Table 1 are characterized by the Young-modulus in its complex form, i.e.:

\[ E_{VE}(f) = 2.98 \cdot \left(0.2233 \cdot f^{0.4915} - 0.1727\right) \cdot 10^6 \] (5)

\[ \eta_{VE}(f) \] (6)

\[ \eta_{VE}(f) = 0.5117 \cdot \exp \left(-\left(\frac{f + 91.33}{915.2}\right)^2\right) + 0.1386 \cdot \exp \left(-\left(\frac{f - 1196}{322.7}\right)^2\right) + \\
+ 0.09709 \cdot \exp \left(-\left(\frac{f - 2113}{1057}\right)^2\right) + 0.4992 \cdot \exp \left(-\left(\frac{f - 1.48 \cdot 10^4}{4.665 \cdot 10^5}\right)^2\right) \] (6)

where \( \eta \) represents the loss factor.

The following section will illustrate how different design parameters affect the result of the damping treatment.

**Effect of design parameters on the complex flexural rigidity**

The computational model has been employed primarily for the design of the HDI on the turret-lathe joint. In order to run the model, the structure has been simplified as consisting of three substructures: primary plate, VE polymer composite and back plate (see Figure 22). The design parameters of interest for this study are, as shown in Figure 22, the plates’ material, the primary plate thickness and the number of applied VE polymer composite layers.
Figure 23; Computed loss factor at 1200Hz as a function of the number of VE polymer layers and primary plate thickness. a) Steel. b) Aluminium.

All these parameters influence the complex flexural rigidity of the structure, as it may occur by observing equation (1) and all of its elements; however it is difficult to instinctively understand how the design parameters affect the complex flexural rigidity due to the complexity of the analytical model where the layers of VE polymer composite also vary. In addition to this, as the VE polymer properties and the wave
number \( (p, \text{ in equation (4)}) \) are frequency dependent, the complex flexural rigidity also results in being a function of the excitation frequency. This makes the graphical exemplification of the model somewhat complicated. To facilitate the analysis of the computational model one may chose the frequency one is interested in and observe the influence of the design parameters for a specific primary and back plate material. This approach has been employed to generate the results presented in this section.

Figure 23 displays the loss factor as a function of the number of VE polymer composite layers and primary plate thickness for steel (a) and aluminium (b) computed for a frequency of 1200 Hz. At a first glance the aluminium primary plate appears to provide a larger loss factor than the steel one, as the absolute maximum (in the domain studied here) is greater than the one provided by a steel primary plate. However this maximum is only attainable by applying a large amount of VE polymer layers and for primary plates thicker than 40 mm. If steel is employed as primary plate material, the absolute maximum for loss factor is attained at a slightly lower thickness.

Figure 24; Computed loss factor (at 1200Hz) as a function of the number of VE polymer layers for a 50mm thick plate. Comparison between steel (blue) and aluminium (red).

The computed loss factor for the steel primary plate displays an interesting behaviour: the loss factor has a local maximum at low number of VE polymer composite layers (see Figure 23 and Figure 24). This behaviour has to be attributed to the back plate properties. As it may be observed in equations (1) to (4), the complex flexural rigidity is dependent on the complex extensional stiffness of the back plate \( (K_3) \). As illustrated in Figure 25, where the computed loss factor as a
function of the number of applied layers of VE polymer composite to a 50 mm thick steel primary plate varying the Young modulus of the back plate is represented, the back plate strongly affects the loss factor for lower amounts of applied VE polymer composite layers.

The reason for this behaviour has to be searched in the shear parameter \((g)\) as this governs both the distance between the neutral plane of the primary plate and the neutral plane of the complete composite plate (equation (3)) and the strain ratio (equation (2)) which in turn are important parts of the complex flexural rigidity (equation (1)). Equation (4) provides an explanation of the dependency of \(g\) on the extensional stiffness of the back plate \((K_3)\). As Figure 26 illustrates, the alteration of the back plate Young modulus only affects the shear parameter for the number of VE polymer layers below 40. After that the curves converge to a common trend as for the loss factor. This can be physically explicated by the fact that as the number of VE polymer composite layers increases, the contribution of the back plate mechanical properties on the shear parameter becomes negligible.

![Figure 25: Loss factor as a function of the number of applied layers of VE polymer composite for a 50 mm thick steel primary plate at 1200 Hz. Effect of the Young-modulus of the back plate on the loss factor.](image-url)
Figure 26: Amplitude of the shear parameter \((g)\) as a function of the number of applied VE polymer composite layers computed for 1200 Hz excitation frequency and 50 mm thick steel primary plate.

Effect of the Young-modulus of the back plate.

As pertaining to the comparison of the computed flexural rigidity, the employment of steel as material for the primary plate ensures an overall greater rigidity than aluminium (see Figure 27), the difference between the two materials becomes greater with increasing of primary plate thickness.

Studying Figure 27(a) it can be noticed that, as expected, even the flexural rigidity has a local maximum at a low number of VE polymer composite layers. Figure 28 demonstrates that the overall flexural rigidity is governed by the primary plate properties, while the effect of the back plate properties is appreciable only when the number of VE polymer composite layers is below 40 although not as extensively as for the loss factor.
Figure 27; Computed flexural rigidity at 1200Hz as a function of the number of VE polymer layers and primary plate thickness. a) Steel. b) Aluminium.
Figure 28; Flexural rigidity as a function of the number of applied layers of VE polymer composite for 50 mm thick steel (solid line) and aluminium (dotted line) primary plates. Effect of the Young-modulus of the back plate on the flexural rigidity.

A further observation on the behaviour of the flexural rigidity has to be made. The model presented so far does not take in consideration possible design limitations on the total thickness of the composite plate; hence the model shows that the flexural rigidity increases with the increasing number of applied VE polymer composite layers. Running the model once more for a steel plate considering that, for instance, the total thickness of the composite plate should be 80 mm, consequently reducing the primary plate thickness for every VE polymer composite layer applied generates the result illustrated in Figure 29(a). As expected the flexural rigidity decreases with the increasing number of applied layers of VE polymer composite. Whereas computing the loss factor considering the same constraints generates the result in Figure 29(b). Even this behaviour is somewhat expectable as the contribution of the VE polymer composite material properties increases with the increasing number of layers and the primary plate’s contribution decreases accordingly, as its thickness diminishes.
Figure 29; Flexural rigidity (a) and loss factor (b) computed for a steel plate putting a constraint on the total thickness (80mm) of the composite plate (blue) and for a steel primary plate of 80 mm of thickness without constraints on the total thickness (red).
3.3 Chapter conclusions

The modelling approach presented in this chapter is particularly valuable for understanding the behaviour of the VE polymer composite employed in the design of the different machine tool components treated.

For the purpose of this research work, the major conclusion that can be drawn from this model is that steel has to be preferred to aluminium as material for the primary plate since it ensures overall greater rigidity and higher loss factor especially when applying less than eight layers of VE polymer composite.

The model can be employed to evaluate the effect of multiple layers of VE polymer composite even in the case of design constraints on the total thickness of the composite plate.
Chapter 4: Design and Implementation of Passive control of Machining Vibration

This section describes the design principles applied to the design of the machine tool components. A short review on the effect stiffness and damping in mechanical structures is given and the concept of High Damping Interface (HDI) is introduced. The core of this chapter is the presentation of the computational model employed for the design of the damped boring bar.

4.1 Design principles

The ultimate objective when designing machine tool components capable of withstanding cutting instability in a passive manner is to enhance both stiffness and damping. On the other hand, these two properties are intrinsically linked to one another and the enhancement of one, usually compromises the other [56]. It is in fact well known that for the majority of machining operations it is the product of stiffness \( K \) and damping \( \delta \) that determines the vibratory conditions in the system [56]. Another aspect to take in consideration is that the overall damping ability of a complex structure, such as a machine tool, not only depends on the individual components’ damping capacity but also, and more considerably, on the damping associated with joints between the very components [4, 58]. Thus the necessity to make use of different components of the machine tool for enhancing both \( K \) and \( \delta \).

4.1.1 Maximizing stiffness using hydrostatic clamping

To maintain a high level of static stiffness, it was chosen to adapt hydrostatic clamping systems to the tools. The effect of this kind of clamping system on the tool deflection is well recognized [68] and straightforward to compute. If the tool is considered to be a cantilever beam, with infinite clamping stiffness, then its transversal vibration will be derived from the equation:

\[
\frac{\partial^2}{\partial x^2} \left( EI \frac{\partial^2 v(x,t)}{\partial x^2} \right) = -\rho A \frac{\partial^2 v(x,t)}{\partial t^2}
\]

(7)

The well-known solution for computing the deflection of a cantilever beam under static load is given by equation (8):

\[
\delta = \frac{F \cdot L^3}{3 \cdot E \cdot I}
\]

(8)
where $F$ is the load, $L$ the overhang, $E$ the Young module and $I$ the moment of inertia. When expressing the moment of inertia for a round section beam (as the common boring bars have) equation (8) becomes:

$$
\delta = \frac{64 \cdot F \cdot L^3}{3 \cdot E \cdot \pi \cdot d^4} = \frac{64 \cdot F}{3 \cdot E \cdot \pi \cdot d} \left( \frac{L}{d} \right)^3
$$

(9)

Where $d$ is the beam section diameter. Equation (9) puts in evidence the importance of the ratio between the overhang and the section diameter.

It is common knowledge that the effective overhang of a tool has to be considered from the outmost fixed point [68], which is the first screw on the conventional screw clamp, and the outmost face on the hydrostatic clamp (see Figure 31). Thus the measured overhang of a tool mounted in a screw clamp does not correspond to the effective overhang. The effect of the hydrostatic clamp can be easily quantified by employing equation (9) to compute the deflection of the tool clamped in it ($\delta_H$) and in the screw clamp ($\delta_C$). As the screw is usually positioned 12 mm from the end of the clamp, the effective overhang of the tool mounted in the screw clamp will be 12 mm longer than the one of the tool clamped in the hydrostatic clamp. The relative difference can be expressed by (10):

$$
\frac{\delta_C - \delta_H}{\delta_C}
$$

(10)

Figure 30; Relative difference of the deflection of the tool as function of the measured overhang.

If, for instance, a boring bar is mounted with a measured overhang of 120 mm, its deflection can be reduced by 25% by employing a hydrostatic clamp instead of a conventional screw clamp (see Figure 30).
This solution does indeed help to minimize vibration amplitude but it does not create any vibration dissipation; what is achieved with such clamping technique is an enhancement of the static stiffness. This means that it is more difficult to excite the system tool-clamp at its natural frequency but when this occurs the system will not oppose resistance. In fact, the stiffness obtained solely using such clamping system for vibration control purposes could actually result in an excessive reduction of damping ratio, defeating the purpose of reducing vibration [56]. Therefore, it is important to be aware of it and exploit it by accompanying the clamping system to a properly designed damping system.

![Comparison between conventional screw clamp (a) and hydrostatic clamp (b).](image)

**Figure 31:** Comparison between conventional screw clamp (a) and hydrostatic clamp (b).

### 4.1.2 Maximising damping using high damping interfaces (HDI)

Interface damping is a well-known phenomenon – having been studied first by Da Vinci and later by Coulomb. Friction arises whenever two, or more, surfaces are in contact and participating to a vibratory movement and this friction eventually translates into damping. High damping interfaces (HDI) are intentionally introduced interfaces where the damping ratio is enhanced by introduction of VE polymer metal composites between the two metallic surfaces composing the interface. As every vibratory mode is characterized by its own damping ratio (and natural frequency), the HDI is effective in those modes where the mode shape involves the HDI, i.e. when the VE polymer composite experience shear strain (as explained in chapter 3). Therefore the positioning of such interface is of vital importance. The latter is illustrated in paper V where a HDI has been implemented on a palletized active alignment workholding system. The experimental modal analysis (EMA) shows that the HDI is effective only for certain modes (see Figure 32), those where the structure shows a relative displacement between the two sides of the damping interface.
Another important issue to consider when designing a HDI is the mechanical impedance between coupled structural elements which control the energy flow path through the structure. It is important that most of the energy flows through the damper. If the energy has an alternative path of propagation with lower mechanical impedance (Figure 33(a)), the energy will by-pass the damper [68, 68]. For this reason, the damping interface should be the only structural component in the energy flow path, see Figure 33(b).

Figure 33; Energy propagation paths. (a) Bypass through metal-to-metal contact. (b) No bypass, energy flows through damping material.
4.2 Computational model of the damped boring bar

The damped boring bar (DBB) consists of a steel bar on which a certain number of constrained viscoelastic plates are mounted (see Figure 34).

![Diagram of damped boring bar](image)

*Figure 34; Damped boring bar and constrained viscoelastic (VE) plates. The edges of the VE plates are considered elastically restrained against rotation.*

The relatively soft viscoelastic polymer is sandwiched between the structure to be damped and a thin constraining layer with high Young modulus. Direct stresses are resisted primarily by the constraining layer, while the viscoelastic layer acts primarily in transverse shear. In flexural vibration the shearing strain in the viscoelastic layer dissipates energy and thereby reduces vibration.

The computation model of the damped boring bar (DBB) is accomplished in three steps:

1. Computation of the motion equation of the VE plate
2. Computation of the flexural motion of the DBB
3. Computation of the rotational motion of the DBB
4.2.1 Computation of the motion equation of the VE plate

The analysis of flexural vibrations of annular, polar VE plates with edges elastically restrained against rotation is of interest since ideal supports or clamps are practically impossible to obtain in practice. The computation is performed for the transversal motion of the thin VE-plate. The axisymmetric case, when the material properties, load and boundary conditions are independent of $\theta$, is considered. The axisymmetric motion of a circular plate of radius, $a$, of an annular plate, in polar coordinates $r, \theta$ is governed by the equation

$$-rac{\partial}{\partial r} (r Q_r) + \rho h \frac{\partial^2 w}{\partial t^2} = 0$$

(11)

where $b < r < a$, $b$ and $a$ are the inner and outer radii, respectively, of the annular plate, $w$ is the transvers displacement, $h$ is the thickness of the plate, $\rho$ is the mass density per unit volume of the plate and $Q_r$ is the shear force

$$Q_r = \frac{1}{r} \left[ \frac{\partial}{\partial r} (r M_{rr}) - M_{\theta \theta} \right]$$

Where $M_{rr}$ and $M_{\theta \theta}$ are the moments per unit length of the orthotropic plate

$$M_{rr} = \int_{-h/2}^{h/2} \sigma_{rr} z dz \quad M_{\theta \theta} = \int_{-h/2}^{h/2} \sigma_{\theta \theta} z dz$$

(12)
The transversal displacement $w$ is a function of $r$, $\theta$ and $t$, $w(r, \theta, t)$. In order to express the governing equations in terms of the displacement, $w$, the bending moments in Equation (12) must be transformed with the help of the constitutive and kinematics equations respectively.

\[
\begin{bmatrix}
\sigma_{rr} \\
\sigma_{\theta\theta} \\
\sigma_{r\theta}
\end{bmatrix} = \frac{E}{1-\nu^2}
\begin{bmatrix}
1 & \nu & 0 \\
\nu & 1 & 0 \\
0 & 0 & \frac{1-\nu}{2}
\end{bmatrix}
\begin{bmatrix}
(\varepsilon_{rr} - \alpha \Delta T) \\
(\varepsilon_{\theta\theta} - \alpha \Delta T) \\
2\varepsilon_{r\theta}
\end{bmatrix}
\]

Regarding the VE polymer the in-plane shear deformation can be neglected, and the stress–strain relationship can be expressed as

\[
\begin{bmatrix}
\sigma_{rr} \\
\sigma_{\theta\theta}
\end{bmatrix} = \frac{E}{1-\nu^2}
\begin{bmatrix}
1 & \nu \\
\nu & 1
\end{bmatrix}
\begin{bmatrix}
(\varepsilon_{rr} - \alpha \Delta T) \\
(\varepsilon_{\theta\theta} - \alpha \Delta T)
\end{bmatrix}
\] (13)

where $\Delta T$ is the temperature rise from an undeformed state, $\alpha$ is the coefficient of thermal expansion and $\nu$ is the Poisson's ratio. From Equations (12)-(13) it can be further derived

\[
M_{rr} = \int_{-h/2}^{h/2} \sigma_{rr} z dz = D(\varepsilon_{rr}^* + \nu \varepsilon_{r\theta}^*) - M_T
\]

\[
M_{\theta\theta} = \int_{-h/2}^{h/2} \sigma_{\theta\theta} z dz = D(\nu \varepsilon_{rr}^* + \varepsilon_{\theta\theta}^*) - M_T
\]

where $M_T$ is the thermal moment and $D$ represents the bending stiffness expressed by

\[
D = \frac{Eh^3}{12(1-\nu^2)}
\]

and

\[
\varepsilon_{rr} = z \varepsilon_{rr}^*
\]

\[
\varepsilon_{\theta\theta} = z \varepsilon_{\theta\theta}^*
\]

\[
\varepsilon_{rr}^* = -\frac{\partial^2 w}{\partial r^2}
\]

\[
\varepsilon_{r\theta}^* = \frac{1}{r} \left( \frac{\partial w}{\partial r} + \frac{1}{r} \frac{\partial^2 w}{\partial r^2} \right)
\]
Now, neglecting the thermal moment $M_T$, the bending moments can be expressed in terms of the deflection $w$ using the relations

$$M_n = -D \left( \frac{\partial^2 w}{\partial r^2} + \frac{v_r}{r} \frac{\partial w}{\partial r} \right)$$

$$M_{\theta \theta} = -D \left( v_\theta \frac{\partial^2 w}{\partial r^2} + \frac{1}{r} \frac{\partial w}{\partial r} \right)$$

The equation of motion (11) can now be written in terms of deflection $w$

$$\left[ D \frac{\partial^4 w}{\partial r^4} + 2A_1 \frac{\partial^3 w}{\partial r^3} + A_2 \frac{\partial^2 w}{\partial r^2} + A_3 \frac{\partial w}{\partial r} \right] = -\rho h \frac{\partial^2 w}{\partial t^2}$$

where

$$A_1 = \frac{1}{r} \left( D + r \frac{\partial D}{\partial r} \right)$$

$$A_2 = \frac{1}{r^2} \left[ -D + r(2 + \nu) \frac{\partial D}{\partial r} + r^2 \frac{\partial^2 D}{\partial r^2} \right]$$

$$A_3 = \frac{1}{r^3} \left( D - r \frac{\partial D}{\partial r} + r^2 \nu \frac{\partial^2 D}{\partial r^2} \right)$$

The transversal deflection $w(r, \theta, t)$ can be separated into two functions

$$w(r, \theta, t) = w_0(r, \theta) \psi(t)$$

where $w_0$ is a function of only $r$ and $\theta$. Then equation (15) can be rewritten as

$$\frac{1}{\rho h w_0} \left[ D \frac{\partial^4 w_0}{\partial r^4} + 2A_1 \frac{\partial^3 w_0}{\partial r^3} + A_2 \frac{\partial^2 w_0}{\partial r^2} + A_3 \frac{\partial w_0}{\partial r} \right] = -\frac{1}{\psi} \frac{\partial^2 \psi}{\partial t^2}$$

According to Gupta [69], equation (16) can be written as a system of two equations

$$D \frac{\partial^4 w_0}{\partial r^4} + 2A_1 \frac{\partial^3 w_0}{\partial r^3} + A_2 \frac{\partial^2 w_0}{\partial r^2} + A_3 \frac{\partial w_0}{\partial r} - \omega^2 \rho h w_0 = 0$$

$$\frac{d^2 \psi}{dt^2} + \omega^2 \psi = 0$$

where $\omega$ is a constant to be determined. Ramaiah and Kumar [70] have derived the expressions for maximum strain energy, $V_{max}$, and maximum kinetic energy, $K_{max}$, in the form
Design and Implementation of passive control of machining vibration

\[
V_{\text{max}} = \frac{1}{2} \int_0^b \left( D \left( \frac{\partial^2 w_0}{\partial r^2} \right)^2 + 2 \frac{D_1}{r} \frac{\partial^2 w}{\partial r^2} B_1 + \frac{D_\theta}{r^2} B_1^2 + 2D_{r\theta} \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial w_0}{\partial \theta} \right)^2 \right) \right) r dr d\theta \quad (18)
\]

and

\[
K_{\text{max}} = \frac{1}{2} \omega^2 \int_0^b \rho h w_0^2 r dr d\theta \quad (19)
\]

where

\[
B_1 = \left( \frac{\partial w_0}{\partial r} + \frac{1}{r} \frac{\partial^2 w_0}{\partial \theta^2} \right)
\]

\[
D_1 = \nu D
\]

\[
D_{r\theta} = \frac{G h^3}{12}
\]

\[D_{r\theta}\] is shear rigidity and \(G\) is shear modulus.

By using variational calculus, this requires that

\[
\delta (V_{\text{max}} - K_{\text{max}}) = 0 \quad (20)
\]

satisfying relevant boundary conditions. Considering the mode shape of the form

\[w_0(r, \theta) = w(r)\cos(\theta)\]

then equations (18) and (19) can be written as

\[
V_{\text{max}} = \frac{1}{2} \int_0^a \left( D \left( \frac{\partial^2 w_1}{\partial r^2} \right)^2 + 2 \frac{D_1}{r} \frac{\partial^2 w_1}{\partial r^2} B_1 + \frac{D_\theta}{r^2} B_1^2 + 4D_{r\theta} \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial w_1}{\partial \theta} \right)^2 \right) \right) r dr
\]

and

\[
K_{\text{max}} = \frac{\pi}{2} \omega^2 \int_0^a \rho h w_1^2 r dr
\]

Applying the Rayleigh-Ritz method, the mode shape \(w_1(r)\) in the radial direction is expressed as

\[w_1(r) \approx W_N(r) = \sum_{j=1}^{N} C_j \phi_j(r) \quad (21)\]

in which \(\phi_j(r)\) are chosen admissible functions satisfying the geometric boundary conditions such that they are linearly independent and form a complete set. \(C_j\) are linear parameters to be determined from the variational condition (20). The method leads to a set of linear, homogeneous, simultaneous equations in the \(C_j\)’s. The
condition for non-trivial solutions of this set of equations results in a characteristic equation for the determination of eigenvalues

\[ \det(V_{ij} - \omega^2 \rho h T_{ij}) = 0 \]  

(22)

where \( V_{ij} \) and \( T_{ij} \) are symmetric functions in \( i \) and \( j \). Now, the transvers deflection \( W_N(r) \) can be determined from (21).

To account for the behaviour the rheologic operator \( \Delta \) is introduced to multiply the right side of Equation (15). The time equation becomes then viscoelastic

\[ \frac{d^2 \Psi}{dt^2} + \omega^2 \Delta \Psi = 0 \]  

(23)

For a Maxwell model the \( \Delta \) operator is

\[ \Delta = \frac{G}{\eta} + \frac{d}{dt} \]

and Equation (23) becomes

\[ \frac{d^2 \Psi}{dt^2} + \omega^2 \frac{d\Psi}{dt} + \omega^2 s \Psi = 0 \]

(24)

The differential Equation (24) has the characteristic equation

\[ \lambda^2 + \omega^2 \lambda + \omega^2 s = 0 \]

The roots of the characteristic equation are

\[ \lambda_{1,2} = -\frac{\omega^2}{2} \pm \sqrt{\left(\frac{\omega^2}{2}\right)^2 - \omega^2 s} \]

or

\[ \lambda_{1,2} = -\frac{\omega^2}{2} \pm i \sqrt{\left(\frac{\omega^2}{2}\right)^2 - \omega^2 s} = -\frac{\omega^2}{2} \pm i\Omega \]

The solution of (24) can be expressed by

\[ \Psi(t) = \exp\left(-\frac{\omega^2}{2} t\right) \left(C_1 \cos \Omega t + C_2 \sin \Omega t\right) \]

\( C_1 \) and \( C_2 \) are constant that are determined from the initial conditions, \( \Psi' = 1 \) and \( d \Psi/dt = 0 \) at \( t = 0 \).
Using the above procedure, characteristics of the VE polymer can be determined and the effect of the VE polymer parameters on the dynamic behaviour may be analysed. The computation approach has been implemented in a FE-program, where the main part concerns VE-material behaviour evaluation. Viscoelastic polymers have a time-dependent response even if the loading is constant. Linear viscoelasticity is adopted in all calculations and the stress is considered to vary linearly with strain and strain rate. It is also assumed that the viscous part of the deformation is incompressible, so that the volume change is purely elastic. To ensure incompressibility, the Cauchy stress tensor, $\bar{\sigma}$, is expressed in the following form

$$\bar{\sigma} = -pl + \tilde{s}$$

where $p$ is the volumetric stress and $\tilde{s}$ is the deviatoric part of the stress tensor. The general linear dependence of the stress deviator on the strain history can be expressed by the hereditary integral

$$\tilde{s}(t) = \int_{-\infty}^{t}\Gamma(t-\tau)\frac{d\tilde{\varepsilon}}{d\tau} d\tau$$

where $\Gamma(t-\tau)$ is the relaxation shear modulus function and $\tilde{\varepsilon}$ is the deviatoric part of the strain.

A so-called generalized Maxwell model is implemented where the relaxation function is evaluated by Prony series

$$\Gamma(t) = G + \sum_{m=1}^{M} G_{m}\exp\left(-\frac{t}{\tau_{m}}\right)$$

Hence, $\tau_{m}$ is the relaxation time constants of the spring-dashpot pairs in the same branch, and $G_{m}$ represents the stiffness of the spring in branch $m$.

In frequency domain, the deviatoric parts of stress and strain are represented as

$$\tilde{s} = \text{real}(\tilde{s}\exp(j\omega t))$$
$$\tilde{\varepsilon} = \text{real}(\tilde{\varepsilon}\exp(j\omega t))$$

and

$$\tilde{s} = 2(G' + jG'')\tilde{\varepsilon}$$

where $G'$ is the storage modulus and $G''$ is loss modulus

$$G' = G + \sum_{m=1}^{M} G_{m}\frac{(\omega \tau_{m})^{2}}{1+(\omega \tau_{m})^{2}}$$
$$G'' = \sum_{m=1}^{M} G_{m}\frac{\omega \tau_{m}}{1+(\omega \tau_{m})^{2}}$$
Figure 36 and Figure 37 show the storage shear modulus and loss factors values at different time intervals.

**Figure 36; Storage shear modulus values for the VE-material at different time intervals.**

**Figure 37; Loss factors values for the VE-material at different time intervals.**

The computed storage shear modulus and loss modulus as frequency functions are illustrated in Figure 38.
4.2.2 Computation of the flexural and rotational motion of the DBB

This section presents the analysis of the effect of the damping plates on the boring bar. The energy introduced in the elastic structure can be very large during internal turning where the ratio between the length of the bar and its diameter is large. Therefore, it is important to design a system to dissipate a part of this energy. As described above, the composite plates, formed by a VE-material constrained between two Al-plates are mounted on the boring bar. The damped plates on the bar form a complex system that will be studied in this section. First the dynamic analysis of the undamped boring bar is presented. The bar is considered to be constrained as shown in Figure 39.
Figure 40 and Figure 41 show the mode shapes of the boring bar for the flexural and torsional modes respectively.

Figure 40; Undamped boring bar, flexural mode X-dir, $f_0 = 847\text{Hz}$ (a) and Undamped boring bar, flexural mode Y-dir, $f_0 = 841\text{Hz}$ (b).

Figure 41; Undamped boring bar, torsional mode 4852 Hz.

Table 2; Natural frequencies and mode type, undamped boring bar.

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Mode type</th>
</tr>
</thead>
<tbody>
<tr>
<td>841</td>
<td>flexural</td>
</tr>
<tr>
<td>847</td>
<td>flexural</td>
</tr>
<tr>
<td>4852</td>
<td>torsion</td>
</tr>
</tbody>
</table>

Table 3 illustrates how the natural frequencies change when applying three constrained VE plates (see Figure 43). As one might expect, the natural frequencies drop, basically due to the drop in stiffness. Although this scenario would be unrealistic for the final design of the boring bar, it can be useful for studying the behaviour of such a design approach.
Table 3; Natural frequencies, boring bar with 3 constrained damped plates.

<table>
<thead>
<tr>
<th>Frequency [Hz]</th>
<th>Mode type</th>
</tr>
</thead>
<tbody>
<tr>
<td>221</td>
<td>flexural</td>
</tr>
<tr>
<td>263</td>
<td>flexural</td>
</tr>
<tr>
<td>3533</td>
<td>flexural</td>
</tr>
<tr>
<td>3602</td>
<td>flexural</td>
</tr>
<tr>
<td>3927</td>
<td>torsion</td>
</tr>
<tr>
<td>5244</td>
<td>flexural</td>
</tr>
</tbody>
</table>

Figure 42; Effect increasing VE-material loss factor on Power dissipation ratio (PDR) on a bar with three constrained VE plates, blue $\eta = 0.4$, red $\eta = 1.0$.

A first consideration might be made on the power dissipation. Power dissipation ratio (PDR) is calculated as the ratio between the power dissipation in the VE polymer and the total power dissipation in the rest of the structure. As can be observed from Figure 42, for lower loss factor ($\eta=0.4$) the PDR increases with the frequency while higher loss factor ($\eta=1.0$) has an opposite effect. Meaning this that a higher loss factor of the VE polymer would not bring any considerable improvement of the overall power dissipation of the plate package composed of the three constrained VE plates.
Figure 43; Boring bar with three constrained VE polymer plates.

Figure 44; Power dissipation density (W) in the VE polymer (blue) and total (red).

Figure 44 illustrates that the power dissipation density of the VE polymer is considerably higher than the one of the rest of the structure, making this an indication of the effectiveness of this concept.

As stated in previous chapters, VE polymers damping effectiveness is related to strain, thus the behaviour of the strain energy has to be studied as well. Figure 45 illustrates how the strain energy is distributed on the three VE polymer plates; the middle plate has to withstand the largest quantity.
Design and Implementation of passive control of machining vibration

Figure 45; Integral of strain energy on the three VE polymer plates illustrated as function of frequency.

An index of the effectiveness of the concept proposed here for the boring bar is given by the strain energy ratio (SER), i.e. the strain energy in the VE polymer versus the total strain energy in the rest of the system. Figure 46 illustrates how this varies with frequency. It appears that, up to 3500 Hz, the strain energy stored in the VE polymer corresponds to 30% - 40% of the total.

Figure 46; Strain energy ratio (SER) of the DBB with three constrained VE polymer plates.
Looking further into the results of the model, it can be observed that the stress normal to the plate’s surface is not evenly distributed on the VE polymer plate (see Figure 47). This suggests that to attain a more even repartition one should apply a pre-stress to the plates, allowing a more efficient use of the VE polymer.

Another study has been carried out in order to take into account the possible ways to clamp the DBB. Two scenarios have been taken into consideration: clamping either (i) the aluminium plates or (ii) the VE polymer plates. As previously stated, this is modelled as an elastic constraint (see Figure 48).

![Figure 47: Stress (in z-direction, i.e. normal to the surface) on the VE polymer plate.](image)

![Figure 48: Illustration of the two possible ways to constrain the plate package: (a) Elastic constrain on the aluminium plate and (b) on the VE polymer plate.](image)
As Figure 49 and Figure 50 illustrate, applying the constraint on the aluminium plates generates a generally lower total displacement. There might be two reasons for such behaviour: (i) aluminium possesses a considerably higher stiffness than the
VE polymer, and (ii) the free VE polymer has to undergo higher strain, and, as explained in chapter 4, this is the condition that allows the VE polymer to be more effective as damping material.

![Figure 51; PDR for DBB with 3 constrained VE polymer plates (blue) and 20 constrained VE polymer plates (red).]

A more realistic scenario for the realization of the DBB is given if the number of constrained VE polymer plates is considerably increased. The study of the PDR for a DBB where 20 plates are applied shows that there is not an appreciable difference between the two scenarios, see Figure 51. This result suggests that the PDR does not change with increasing of the number of constrained VE polymer plates applied.

Table 4; Turning bar with 30 constrained VE plates.

<table>
<thead>
<tr>
<th>Eigenfrequency (Hz)</th>
<th>VE Strain energy density (J)</th>
<th>Strain energy density (J)</th>
</tr>
</thead>
<tbody>
<tr>
<td>454</td>
<td>1.11E+07</td>
<td>2.69E+07</td>
</tr>
<tr>
<td>483.79</td>
<td>1.09E+07</td>
<td>3.08E+07</td>
</tr>
<tr>
<td>4471.25</td>
<td>4.53E+07</td>
<td>2.31E+09</td>
</tr>
<tr>
<td>6776.5</td>
<td>3.46E+08</td>
<td>1.11E+10</td>
</tr>
</tbody>
</table>

By further increasing the number of applied constrained VE polymer plates to 30 and studying the behaviour of the strain energy, one may observe that a considerably large part is stored again in the VE polymer, see Table 4. This result puts in evidence the effectiveness of the design.

Ultimately the expected result of this computational model is an indication on the most suitable amount of VE plates with respect to both damping and stiffness.
Figure 52 illustrates the frequency response to a given dynamic excitation for DBB’s with different amounts of applied VE polymer constrained plates. The diagram shows that stiffness grows with the amount of VE constrained plates; however, damping has its optimum (within the range computed) when applying 50 plates.

4.3 Chapter conclusions

The ultimate objective when designing machine tool components capable of withstanding cutting instability in a passive manner is to enhance both stiffness and damping. On the other hand, these two properties are intrinsically linked one another and the enhancement of one, usually compromises the other [56].

Hydrostatic tool clamping allows maximising the static stiffness as it reduces the effective overhang of the tool. This clamping technique has also the advantage of generating a homogeneous and controllable clamping force. However, it does not create any vibration dissipation. For this purpose a damping system has to be designed.

The HDI implemented on the tool side consists of a certain number of constrained viscoelastic (VE) plates. This chapter has introduced a computational approach for investigating the effectiveness of the proposed design and the major result is the computation of the optimal amount of plates composing the HDI.
This section describes the methodology used to evaluate the proposed concept and the results of such an evaluation.

5.1 Methodology

In order to understand the principle for design of efficient damping systems it is necessary to understand the dynamic behaviour of machining systems. Machining systems may be represented by a closed loop system comprising of the machine tool elastic structure (ES), i.e. the machine tool structure including tool, tool holder, workpiece etc., and the cutting process (CP), i.e. turning, milling etc., see Figure 53 [9, 38, 53]. The interaction between the machine tool’s elastic structure and the cutting process describes the behaviour of the machining system. This behaviour directly affects the process accuracy.

In Figure 53, $F(t)$ is the instantaneous cutting force, $F_0(t)$ is the cutting force nominal value, $x(t)$ is the relative displacement between cutting tool and workpiece, $\Delta d(t)$ is the total deviation of the relative displacement $x(t)$. $P(t)$ and $P_d(t)$ are disturbances such as tool wear, thermal dilation of the elastic structure, variation of rigidity of the elastic structure during a machining process, variation of cutting parameters and so on. Critical factors for the optimization of designs for damped structural components for machine tools are the modal parameters (frequency and damping...
These parameters, that also control the stability of the cutting process, can be extracted from the analysis of the interaction between the two subsystems. Traditional evaluation of machining system dynamic behaviour has invariably been approached in the following steps:

1. Identification of the dynamic properties of elastic structure of machine tools.
2. Identification of the characteristics of cutting process, i.e. the dynamic parameters describing the transfer function of the subsystem represented by cutting process in Figure 53.

The first step is experimentally carried out through experimental modal analysis (EMA) and the second by machining tests, where the acoustic signal or the vibration acceleration is recorded. In some cases these signals are analysed using model based identification algorithms [39] in order to extract the operational dynamic parameters (ODP) [37, 40], This methodology is further discussed in Paper IV where it has been used for the evaluation of a machine tool system with rotating tools. Surface roughness has otherwise been utilized as evaluation criterion.
5.2 Results

The designed components have all undergone experimental modal analysis. The EMA on the turret (Figure 54) validates the conclusions drawn by the modelling activity in chapter 3:

- The 5-layers and 8-layers treatments on steel plates are the ones showing better dynamic and static stiffness, as compared to aluminium solutions.
- There is no major difference between these two treatments on steel, both in terms of static and dynamic stiffness.
- The static stiffness is slightly affected in a negative way (as expected) by the treatments when compared to the conventional turret (Figure 54), but a considerable improvement in dynamic stiffness has been attained, e.g. about 60% lower compliance at the first natural frequency for the 5-layers treatment on steel turret.

![Compliance diagram for 8-layers treatment on aluminium, 5-layers treatment on aluminium, 5-layers treatment on steel, 8-layers treatment on steel and conventional turret.](image)

Therefore this solution seems to be preferable, also considering its easier implementation. The effect of this improved design for the turret on machining results will be presented later in this section.

Pertaining to the boring bar presented in paper I and III, particular interesting observations can be made by looking at Figure 55, where a comparison of the
compliance of the damped and conventional boring bar mounted in a conventional screw clamp and in a hydrostatic clamp is presented. The DBB displays lower compliance than the conventional one when both are mounted in the conventional screw clamp (curve 1 vs. curve 3 in Figure 55). However, the highest dynamic stiffness is displayed by the DBB mounted in the hydrostatic clamp (curve 2 in Figure 55) confirming the expectations described in chapter 4.

Based on this result a comparative SLD for the DBB mounted in the hydrostatic clamp and the conventional boring bar mounted in the screw clamp has been computed assuming that the tools would machine exactly the same workpiece having a hypothetical cutting force coefficient of 1500 N/mm$^2$ (see Figure 56). This comparison already gives an indication of the achievable higher stability limit for the combination of damped tool and hydrostatic clamp. As Figure 56 illustrates, the asymptotical depth of cut limit is almost three times larger for the damped boring bar.
More accurate information on the effectiveness of these improvements can be extracted only after machining tests. Preliminary tests were carried out in form of internal turning on 34CrNiMoS6 (SS 2541) bars with an outer diameter of 150 mm, inner diameter of 48 mm and a length of 170 mm. The tests were performed in a SMT Swedturn 300 (see Figure 57) at three different depths of cut \( (a_p) \), 1 mm, 2 mm and 3 mm, keeping a constant cutting speed \( (v_c) \) at 120 m/min and feed \( (f) \) at 0.15 mm/rev. The boring bars had a section diameter \( d=25 \) mm and the overhang was set to be \( L=125 \) mm (i.e. \( L/d=5 \)). Both conventional boring bar and DBB were mounted in the hydrostatic clamp (on standard turret).
The sound recorded from machining tests has been processed to extract the dynamic characteristics of the process-machine interaction. The sound recorded by the microphone can be shown to have good correlation to the vibration generated during machining [29] and also to have the practical benefit of not interfering in the working zone.

The effect of the tool’s improved dynamic stiffness on the machining process is shown in Figure 58, where the acoustic signals produced by machining with the conventional and the damped tool respectively, are compared. Figure 58 shows the comparison when machining at a depth of cut \((a_p)\) of 1 mm. Machining with conventional tool showed the typical signs of instability, i.e. high and irregular amplitudes, these signs are entirely absent in the signal generated by the DBB.

![Figure 58](image)

*Figure 58; Acoustic signal produced during machining of SS2541 with \(a_p = 1\) mm, \(v_c = 120\) m/min and \(f = 0.15\) mm/rev. Comparison between the signal produced when machining with DBB (red) and with a conventional boring bar (blue), respectively.*

The acoustic signals have also been analysed through model based identification in order to quantify the ODPs (operational frequency, OF, and operational damping ratio, ODR). Figure 59 shows the damping ratio variations as the tool enters the workpiece (at the left end of the figure) and exits close to the chuck (at the right end of the picture).
The operational damping ratio for the conventional boring bar goes towards zero during the machining of the whole length of the workpiece. The DBB worked in stable conditions throughout the whole length of the workpiece with a tendency of stability increase as the tool approaches the chuck.

The effect of this improvement of dynamic properties directly reflects on the surface quality produced on the workpiece as illustrated in Figure 60. The DBB is capable of generating surface roughness as low as 50% of the one generated by the conventional boring bar.
conventional boring bar. The absence of results for the conventional boring bar at $a_p = 3$ mm is explained by the impossibility of running tests at those parameter settings due to excessive instability, while the damped tool still was capable of machining in stable conditions. Figure 61 shows the surface profile taken after machining at $a_p = 1$ mm; the conventional tool (Figure 61(a)) is not able to perform in stable conditions and therefore the surface profile is disturbed by chatter marks.

![Figure 61; Surface roughness scan. (a) Conventional tool, (b) damped tool; the scale is the same for both scans. The scans were performed after machining at $v_c = 120$ m/min, $f = 0.15$ mm/rev and $a_p = 1$ mm.](image)

The effectiveness of the novel turret design in machining conditions has been evaluated in internal turning of SS2541 steel. The boring bars (damped and conventional) were tested and compared when mounted in both conventional and damped turret (see Figure 57). Surface roughness ($R_a$) has been used as criterion. The chosen cutting parameters are summarized in Table 5. Figure 62 summarises the results obtained by machining with the different combinations of tools (damped and conventional) and turrets (conventional and HDI treated steel).
**Table 5: Cutting data settings.**

<table>
<thead>
<tr>
<th>#</th>
<th>$n$ [rpm]</th>
<th>$a_p$ [mm]</th>
<th>$f$ [mm/rev]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1040</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>2</td>
<td>1040</td>
<td>2</td>
<td>0.15</td>
</tr>
<tr>
<td>3</td>
<td>1040</td>
<td>3</td>
<td>0.15</td>
</tr>
<tr>
<td>4</td>
<td>800</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>5</td>
<td>800</td>
<td>2</td>
<td>0.15</td>
</tr>
<tr>
<td>6</td>
<td>800</td>
<td>3</td>
<td>0.15</td>
</tr>
<tr>
<td>7</td>
<td>560</td>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>8</td>
<td>560</td>
<td>2</td>
<td>0.15</td>
</tr>
<tr>
<td>9</td>
<td>560</td>
<td>3</td>
<td>0.15</td>
</tr>
</tbody>
</table>

**Figure 62: Machining test results, surface roughness.** Comparison between the conventional tool clamped in conventional turret and in the damped turret, as well as the damped tool clamped in the conventional turret and in the damped one. The conventional tool clamped in the conventional turret could not perform at setting 7, 8 and 9 due to excessive instability.
From Figure 62 it appears evident that the ordinary configuration (conventional boring bar and conventional turret) suffers from grave instability when machining at settings 7, 8 and 9 (see Table 5 for details). A considerable improvement is attained when either the conventional boring bar is mounted in the damped turret or the DBB is employed instead of the conventional one in the conventional turret; in both cases it becomes possible to machine even at those cutting data settings that generated instability. In addition to this the overall surface roughness is somewhat improved even for those cutting parameters where signs of instability are not as manifest. Further improvement of the resulting surface quality is attained when employing the combination of damped boring bar and damped turret.
Chapter 6: Discussion and conclusions

6.1 Discussion

With the advances in material technology, allowing, for instance, engines to withstand higher combustion pressure and consequently improved performance, comes challenges to productivity. These materials are, in fact, more difficult to machine with regards to tool wear and especially machine tool stability [25]. To cope with this, the usual strategy has been to lower the cutting parameters to a safe level with regards to stability. The research work presented here addresses this problem and proposes what is thought to be the most viable solution.

The literature survey presented earlier in this thesis reveals that the issue of machining vibration has been addressed by many, however, most research effort in the subject has been concentrating on the refinement of the stability limits computation based on the theories originally introduced by Tobias [8] and Tlusty [7]. Yet this approach is not widespread within industry due to the limitations still left for researchers to deal with. To begin with, this approach lacks of an absolute stability criterion [22], in addition to this SLD’s are sensitive to the accuracy of the structural analysis; therefore, in order to attain accurate and reliable results, the analysis ought to be carried out by personnel with experience in the field. Further, once the SLD is computed, it only applies to the specific configuration of workpiece (material and geometry) and tool. In addition to this, this approach cannot take into account non-linear behaviours and for structural dynamics variations of the parts and machines along the tool path, this being the most common occurrence in the manufacturing of advanced products. Last but not least, in the cases when the SLD might be accurately and reliably employed, the usual strategy is to select a higher cutting speed in order to be able to machine in stable conditions with a higher depth of cut. On the other hand, this might compromise tool life, as this is strongly dependent on temperature, which in turns increases with cutting speed, thus affecting productivity as the machining process needs to be interrupted more often for tool change and consequently production costs might increase.

Therefore, this thesis proposes to extend the stability limits of the machine system by enhancing the structure’s damping capability. This would allow for maintaining or even improving productivity since the range of stable cutting parameters might be stretched to such an extent that the user is enabled to freely choose the cutting speed for the operation without having to be concerned about stability.

The literature review also reveals that most effort in improving machining performance by changing the machining system structural behaviour (either by
active or passive means) has been concentrating solely on one specific component at the time.

The aim of the research work presented here is therefore to introduce a unified concept based on the distribution of damping within the machining system components exploiting the dynamic properties of the existing joints composing the machine tool structure. Machine tools are generally overdimensioned in terms of strength [22], thus most of the damping capability is generated by the very joints comprising the structure [4, 58]. However, the damping introduced by these joints is generally not consciously designed or controlled.

The fundamental issue when implementing passive damping in the machining system is to evaluate the necessary damping. In fact, as the vibration energy is not constant in machining, the required damping may vary within a large range. Therefore it is of vital importance to have a computational approach to estimate the achievable damping in a machining system. The optimization of damping through a computational approach implies two further issues:

1. Calculate the loss factor of the damped system, given the loss factor of the particular damping material.
2. Design the optimized damped system to achieve the required damping and stiffness.

The selection of the optimal design for the damped structure is problematic as, virtually, there are infinite combinations of damping material shapes, dimensions and constraining set-ups. Once a generic design is selected one may employ a computational approach to calculate how given design parameters affect the overall damping of the system.

This thesis treats the tool-turret and the turret-lathe joints (see Figure 63) and the High Damping Interfaces (HDI) designed for enhancing the damping capability of these joints exploiting the damping properties of VE polymer metal composites.

The tool-turret joint comprises a multilayer package of VE polymer metal composite plates on the tool shaft side (see Figure 64) and a hydrostatic clamp on the turret side. The computational model presented in chapter 4 reveals that an optimal amount of plates can be calculated to achieve a balance between stiffness and damping. The model considers all the plates of the HDI as clamped, modelled as an elastic constraint, though practically this is only possible by utilizing a hydrostatic clamp as this applies a homogeneous and controllable clamping force on the whole plate package. This is demonstrated by the experimental modal analysis comparing the DBB mounted in screw and hydrostatic clamp (chapter 5, Figure 55). The DBB shows its full damping capability only when mounted in the hydrostatic clamp. This fact might be explained by the screw clamp only acting on a limited number of plates, and therefore only these partake of the vibratory movement.
This HDI has an evident effect on the machining performance as the experimental results demonstrates that the DBB mounted in the hydrostatic clamp is able to machine in stable conditions with a depth of cut $a_p=3$ mm while the conventional boring bar works in fully unstable conditions already at $a_p=1$ mm.

The HDI designed for the turret-lathe joint follows a concept of multiple layers of VE polymer metal plates applied in a sandwich configuration between a base and a primary plate as described in chapter 3. The computational model employed to study this design reveals that the effectiveness of the VE polymer metal composite as damping material not only depends on the applied layers, but also on the geometrical dimensions and material properties of the base and primary plate. The model reveals that with regards to both stiffness and damping, steel has to be preferred as material for base and primary plate. In fact, due to design restriction on
the total thickness of the sandwich plate, it is not possible to apply as many layers of VE polymer metal composite as one may wish, as the stiffness of the sandwich structure drastically decreases (see Figure 65).

Figure 65: Flexural rigidity (a) and loss factor (b) computed for steel primary and base plate constraining the total thickness of the composite plate.

Hence, as the steel primary plate shows a local maximum for the attainable loss factor at a low number of VE polymer plates, this material as to be preferred. The experimental modal analysis carried out on the different configurations for the turret, verifies the result of the model and suggests that the designed HDI composed of five layers of VE polymer metal composite on steel is most efficient in terms of stiffness and damping, if compared to configurations employing eight layers and especially if compared to the case of aluminium as base and primary plate material.

The analysis of the machining test results presented in chapter 5 reveals that employing the improved turret in combination with a conventional boring bar already generates a quantifiable benefit in terms of produced surface roughness within the range of tested cutting parameters. Nevertheless, the ultimate improvement of the machining performance, in terms of surface roughness, is brought by the combined adoption of DBB and damped turret.
6.2 Conclusions

In this section the research questions stated in chapter 1.3 are firstly addressed and, secondly, the industrial implications of this work are discussed.

6.2.1 Addressing the research questions

1. *The theoretical treatment and experimental methodology for exploiting the dynamic properties of existing joints in the machine tool structure to control the overall damping capability of the machining system, possibilities and limitations.*

The overall damping capability of the machine tool structure is basically governed by the structural joints themselves [4, 58]. This thesis introduces the concept of HDI which refers to a new fundamental approach to modeling, analysis and design of machining systems. The HDI concept is employed to develop methods for optimizing the distribution of damping in the machine structural components and for balancing damping and stiffness at system level. The analysis of the computational models illustrated in chapter 3 and chapter 4 shows that a HDI solution that exploits VE polymers damping properties can be employed for enhancing the damping of the given joints and therefore of the machining system. In addition to this, these computational models offer the possibility of consciously design damping in the machining system structure and balance it with regards to the needed stiffness.

The experimental modal analysis presented in chapter 5 confirms that the introduction of the HDI has a significant effect on damping (see Figure 54 and Figure 55) reflecting also on the machining performance.

The limitation of this approach concerns the possibility of optimizing the damping and stiffness at system level, i.e. the distribution of damping and stiffness in joints based on objective criterions. In addition to this, this thesis only treats the designed-in damping approach for continuous chip removal processes. However, the theoretical approach is straightforward for intermittent machining processes.

2. *The theoretical modelling and design approach for improving machining system performance by using single-HDI configuration and further by using multi-HDI configuration (i.e. distributing damping).*

The machining tests illustrated in chapter 5 demonstrate that a significant improvement of machining performance can be attained by solely employing the HDI in the tool-turret joint. The comparison of the acoustic signals recorded during machining (Figure 58) generated by the conventional and damped boring bar clamped in a conventional turret demonstrates that the sole improved tool-turret joint is capable of machining in stable conditions whereas the conventional configuration of tool and turret shows signs of instability already at the lowest tested depth of cut. The ODR (Figure 59) confirms this statement and the measured surface roughness (Figure 60 and Figure 61) quantifies the improvement allowed by the sole HDI between tool and turret to a 50% reduction.
An improvement of machining performance in terms of surface roughness is attainable also by only employing the HDI in the turret-lathe joint (see Figure 62). Nevertheless, it is the combination of the two HDI’s that generates the overall best results in terms of surface quality (see Figure 62).

In conclusion, it is the employment of the full chain of redesigned components (tool-clamp-turret) that enables to generate the lowest surface roughness over the whole range of tested cutting parameters (Figure 62). In addition to this, the improved machining system is not affected by instability at any of the tested cutting parameters.

### 6.2.2 Industrial implications

From the industrial application point of view, the presented approach allows the end user to select the most suitable parameters in terms of productivity avoiding the hassle of tuning the devices, having to acquire a deep knowledge in structural dynamics or having to use additional control systems. In addition to this, the enhanced machine tool system becomes less sensitive to stability issues provoked by difficult-to-machine materials or even fluctuations of the work material properties that might occur in everyday production processes.

### 6.3 Future work

As stated in previous research work [53], to attain a considerable effect on the machining performance, the source of damping should be as near as possible to the cutting zone. This thesis has demonstrated that an enhancement of damping as far as in the turret-lathe joint has a quantifiable effect on the machining performance. On the other hand, the boundaries for improving machining performance by enhancing joint damping have not been identified. For this reason, the natural continuation of this work would be to find these boundaries in order to further improve machining performance to its maximal extent. Practically, components as the spindle, chuck, tailstock, machine bed and guides must be included in the forthcoming research work. This would establish how far in the machine tool structure one may enhance damping and still have a positive quantifiable effect with regards to machining performance. As this thesis only treated the turning machining system, its natural extension would be the investigation of other typologies such as milling and grinding.

In addition to this, extensive studies on the effect of the enhanced damping capability of the machining system on tool life ought to be carried out in order to attain a broader understanding concerning productivity.

Finally, as material technology, regarding alternative damping materials, is advancing, research effort should also be focused on exploring the possibility of employing different materials for enhancing damping capability.
Nec vero probare soleo id, quod de Pythagoreis accepimus, quos ferunt, si quid adfirmarent in disputando, cum ex iis quaeretur, quare ita esset, respondere solitos "ipse dixit"; ipse autem erat Pythagoras: tantum opinio praebidicata poterat, ut etiam sine ratione valeret auctoritas.

Marcus Tullius Cicero

References


