Regenerative Chatter Vibration in Indexable Drills: Modeling and Simulation

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Amir Parsian
This dissertation is dedicated to my wife Maria.
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Abstract

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An indexable insert drill is a drill which uses cutting inserts to make holes. Undesirable sound generated by this type of drill has always been considered as a problem in workshops. The focus of this thesis is to investigate the mechanism behind these vibrations, to model it and to provide guidelines for reducing the sound in future drill designs. Primary investigations show that the main sound-generating mechanism is a self-induced vibration due to a coupled torsional-axial deformation in the drill which leads to the torsional-axial chatter vibration. The first step of simulating regenerative chatter vibrations in a drill is to model the static cutting forces. In this thesis, a model is proposed to estimate static cutting forces in indexable drills by dividing the cutting edges into small elements. Since, using this model, forces can be calculated separately on each insert, it is possible to consider insert differences in estimation of the cutting loads. Torsional-axial coupling has been discussed and subsequently a time-domain model is proposed to simulate chatter vibrations. The resulting model is a system of delay differential equations with variable delays. The delay varies with time and is dependent on the state of the system. Variations in the time-delay, tool jump-out and backward motions of inserts have been included in the proposed time-domain simulation. A set of experiments was conducted to verify the model. Finally, a number of different strategies to alleviate the problem of chatter vibration are explored and their feasibilities for use in future products are discussed.
List of Publications

**Paper A.** A Mechanistic Approach to Model Cutting Forces in Drilling with Indexable Inserts
Published at “Procedia CIRP, 2014, 24, 74 - 79” – Authors: Amir Parsian, Martin Magnevall, Tomas Beno, Mahdi Eynian

**Paper B.** Time-Domain Modeling of Torsional-Axial Chatter Vibrations in Indexable Drills with Low Damping
Presented at “The 4th International Conference on Virtual Machining Process Technology” in Vancouver, Canada, 2015 – Authors: Amir Parsian, Martin Magnevall, Tomas Beno, Mahdi Eynian

**Paper C.** Time domain simulation of chatter vibrations in indexable drills
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**Paper D.** Sound Analysis in Drilling, Frequency and Time Domains
Published at “Procedia CIRP, 2017, 58, 411 - 415” – Authors: Amir Parsian, Martin Magnevall, Mahdi Eynian, Tomas Beno

**Paper E.** Modeling of torsional-axial chatter vibrations in drilling: a review
Authors: Amir Parsian, Martin Magnevall, Mahdi Eynian, Tomas Beno

**Paper F.** Minimizing the negative effects of coolant holes on torsional rigidities of drills
Authors: Amir Parsian, Martin Magnevall, Mahdi Eynian, Tomas Beno
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Appended Publications

Paper A. A Mechanistic Approach to Model Cutting Forces in Drilling with Indexable Inserts

Paper B. Time-Domain Modeling of Torsional-Axial Chatter Vibrations in Indexable Drills with Low Damping

Paper C. Time domain simulation of chatter vibrations in indexable drills

Paper D. Sound Analysis in Drilling, Frequency and Time Domains

Paper E. Modeling of torsional-axial chatter vibrations in drilling: a review

Paper F. Minimizing the negative effects of coolant holes on torsional rigidities of drills
1 Introduction

1.1 Background

According to the World Bank, in the last decade around 15.5-17.5 percent of the world’s GDP was generated by manufacturing. In 2012, around ninety million jobs were provided directly and indirectly by the manufacturing sector in the European Union\(^1\). Growing demand of human societies for a higher standard of living through employing new technologies, combined with eco-friendly and energy-efficiency obligations, are driving the manufacturing sector to be more innovative and efficient and at the same time more responsive to changes.

Machining processes are important manufacturing methods that are widely used to make high quality products with close tolerances. While different manufacturing methods have been developed to cope with the constantly growing complexities of industrial products, machining plays an indispensable role in producing many manufactured components. At the end of the last century Merchant [1] highlighted the importance of machining by mentioning that 15 percent of all product values are due to machining, and in production of engineering components the machining process is used more than any other manufacturing process [1]. It is predicted that machining will continue to play its important role in the coming decades [2].

Although a large variety of machining processes have been developed over time, it is possible to categorize them into two main types namely conventional, or traditional, and non-conventional machining. While conventional machining processes e.g. turning, drilling and milling are processes in which cutting edges remove material from workpieces through movement of tools relative to workpieces, the term non-conventional machining refers to other cutting processes for example water jet, electrochemical machining, electrical discharge machining, etc.

Hole making with drills is a conventional machining operation widely used in manufacturing. Holes are made in manufactured parts for different reasons, for

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\(^1\) Advancing Manufacturing paves way for future of industry in Europe; European Commission-MEMO/14/193, 19 March 2014
example where screws or pins are to be inserted, or channels must be made to convey fluids. According to estimations made by Sandvik Coromant, drilling accounts for 15% of the market for metal cutting tools.

Depending on cutting parameters, workpiece material, quality requirements and economic considerations, different types of drills can be used such as solid carbide, exchangeable-tip and indexable insert drills. Indexable drills utilize inserts to make holes. Two typical inserts for drilling are shown in Figure 1.1a. Due to exchangeability of the inserts, using them improves the economy of machining. Inserts are made of more durable materials in comparison to the drill-body which makes it possible for this type of drill to tolerate severe cutting conditions. In most cases, each insert has several cutting edges that are used in turn by rotating, indexing, the inserts. Having edges on the inserts, rather than on the tool-body, makes it possible to use the tool-body for a longer time simply by indexing the inserts until all cutting edges are worn-out and then the insert is replaced. Most indexable insert drills have two inserts; namely the periphery and the central. As shown in Figure 1.1b, inserts are mounted at different radial distances to the drill rotation axis and each insert cuts a part of the hole.

![Figure 1.1: (a) Typical central and peripheral inserts, (b) Inserts on a typical indexable drill (source: www.sandvik.coromant.com)](image)

The drilling process with indexable drills can generate a disturbing squealing and high-pitched sound which limits the usage of such tools. The cause of this sound, which is undesirable in the workplace and detrimental to product quality needs to be investigated and mitigated in a systematic way.

### 1.2 Primary Investigations of the High Pitch Sound

One approach to understand more about the underlying mechanisms responsible for causing the sound is to look at its acoustic spectrogram. The first step is to measure the sound. The experimental setup is shown in Figure 1.2.
Figure 1.2: The experimental setup and a flow diagram of the sound measurement procedure

In the spectrogram, the time and the frequency are shown on the horizontal and vertical axes respectively. Colors are used to show the amplitude of the components. As an example, an indexable insert drill with diameter of 24 mm and length of 96 mm is used to make a hole at feed rate of 0.15 mm/rev and cutting speed of 100 m/min. The measured sound and its spectrogram is given in Figure 1.3. As can be seen from the figure, the measured signal is dominated by a periodic component and its higher harmonics. This appears in the form of three distinct horizontal lines associated with integer multiples of 3.8 kHz which is considered as the fundamental frequency. The line which represent this fundamental frequency is the main component of the sound and appears shortly after the drilling process starts.
Studies of different drill sizes and cutting parameters reveals that a wide range of indexable insert drills follow a similar pattern; where the vibration is dominated by a frequency component around 1.5-6 kHz and its higher harmonics. Comparing this with the normal equal loudness level contours, which is plotted according to ISO 226:2003 [3], shows that this type of vibration occurs in a frequency range where the human auditory system is sensitive, which amplifies its negative effects in the workplace. A typical normal equal loudness level contour is shown in Figure 1.4. More detailed discussions about the sound measurement contents in drilling with indexable drills is presented in Paper D [4].
Since the source of the sound is vibration, it is essential to explore the different types of vibration which can appear in drilling operations. In machining, vibrations can be free, forced or self-induced [5]. Technical definitions of these types of vibrations are found in ISO-2041-2009 [6]. Regardless of the type, vibrations are usually an unwanted phenomenon in machining, because they cause noise, poor surface quality, out of tolerance products and shorten the tool life. Therefore, except cases such as vibration assisted machining [7, 8, 9], substantial efforts are made by both academia and industry to maximize process stability in machining operations to make them less prone to vibration.

Free vibrations are the result of a transient excitation. During free vibration, the system vibrates at its natural frequencies and the response will decay over time. Forced vibrations are caused by periodic changes in the excitation and the response of the system will be harmonic with the fundamental frequency related to the time interval between the successive changes in the excitation. The frequency of this harmonic response is independent of the natural frequencies of the system. In contrast, the vibration associated with high pitched sound in drilling with indexable insert does not exhibit such signs. Its dominant frequency is always close to and slightly less than the first torsional-axial frequency of the clamped drill, according to Paper D [4] and will not decay as long as the drilling operation continues. This is a typical sign of a self-induced type of vibration known as regenerative torsional-axial chatter [10].

In this type of chatter, the length of the tool periodically changes during the drilling operation, which can be observed at the bottom of the hole. Typical chatter marks are shown in Figure 1.5.
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Figure 1.5: Chatter marks at the bottom of the hole which indicates periodic changes of the length of the drill. It is clear that the drill exhibits multiple periodic changes in length during a single revolution.

1.3 Literature Survey

In general usage, chatter refers to a quick and repeated high pitched sound. In the field of metal cutting the term chatter refers to noisy vibrations arising from different mechanisms. While some authors use the term to refer to vibrations made by both self-excited and forced vibrations [11, 12], others have used chatter only for self-induced vibrations [5, 13, 14, 15, 16, 17, 18, 19]. Throughout this thesis the term chatter refers only to the vibrations arising from a self-induced mechanism which is also called regenerative chatter vibration.

Regenerative chatter vibrations lead to excessive sound which is troublesome in many workshops. They cause poor surface finish and lower accuracy, and shorten the tool life [15]. As the relative distance between the tool and the workpiece varies due to dynamic forces, a wavy surface is imprinted on the workpiece. During the next cut, these imprinted waves in combination with current motions of the tool-workpiece system cause variations in uncut chip thickness and therefore cutting forces which consequently affect the dynamic motions of the system. This interaction can lead to the development of regenerative chatter vibrations. Details about the mechanism of regenerative chatter can be found in the literature [20, 21].

Taylor who is referred as "the father of metal cutting science" [22], gave one of the earliest explanation for the chatter problem in his book on metal cutting [23]. Taylor in [23] explained chatter as being a result of changes in cutting pressure which excite one of the resonance frequencies of the system. In 1946, Arnold [24] described self-induced vibrations as the result of changes in cutting speed which lead to changes in cutting forces and consequently vibrations. Later, more detailed investigations by other researchers, including Tobias and Fishwick [25, 26] and
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Tlusty and Polacek [27] show that self-induced vibrations are usually the result of a feedback effect originating from previous cuts. Other pioneering studies on chatter vibrations in machining were conducted by Hahn [28], Doi and Kato [29]. The chatter phenomenon has been investigated for different metal cutting operations such as turning [11, 30], milling [31, 32, 33], boring [30, 34, 35, 36] and drilling [37, 38, 39, 40, 41, 10]. A comprehensive review on chatter vibrations is given by Quintana et al. in [42].

Surveying the literature shows that the main sources of vibrations in drilling are whirling, lateral chatter and torsional-axial chatter. Whirling vibrations occur at frequencies which are odd multiples of the spindle frequency and are not within the scope of this study. The interested reader is referred to studies done by Bayly et al. [43], Fujii, Marui and Ema [44, 45, 46].

Lateral chatter occurs in drills where a high ratio between length and diameter results in low bending stiffness and occurs close to the first and second bending mode of the clamped drill. This has been studied by Arvajeh and Ismail [47, 48], Roukema and Altintas [37, 38] and Ahmadi and Altintas [39] Ahmadi and Savilov [40]. As mentioned before, the dominant frequency of the vibration in drilling with indexable insert drill is close to, and less than, the first torsional-axial resonance frequency of the clamped drill [4] and not related to the bending modes.

A breakthrough in drill design was the introduction of the twist drill, by Morse, in 1863 U.S. Patent 38,119 [49]. The usage of helical flutes is also common in indexable drill design. A drill with helical flutes can be represented by a pre-twisted beam with a non-circular cross-section which cause a coupling between torsional and axial deformation [10, 50, 51, 52].

Bayly et al. in [10] have considered the variations in length of the drill as the source of torsional-axial chatter vibration. In their model the length of the drill changes because of the axial force and, via the previously mentioned coupling, due to cutting torque. They proposed a single degree of freedom model which can predict the onset of chatter.

Although, based on the coupling property, different studies have been performed on torsional-axial chatter in drilling [37, 38, 39, 40], the behavior of the indexable drill during chatter vibration has not been the main focus. Instead the objective has been on the prediction of the onset of chatter, and usually for solid carbide drills. Indexable drills are usually larger in diameter than solid carbide and exchangeable-tip drills and therefore the sound generated from these is more pronounced. Torsional-axial chatter vibration is the main source of sound in indexable insert drilling. A small helix angle in combination with its asymmetric
design are characteristic features of the indexable drill that are essential to consider.

Based on Sandvik Coromant’s estimation, indexable insert drills constitute approximately 53% of the market for drilling tools. This means that almost 8% of the total market for metal cutting tools is for indexable insert drills. Regarding vibration, the torsional-axial chatter vibration is the main problem in these drills and its reduction is a significant driver for improvement in the design. Despite this, the amount of research dedicated to this problem in indexable drills is very limited.

With the long-term objective to optimize drill design, a deep understanding of the regenerative chatter phenomenon is important to build computationally efficient simulation models which accurately describe the core physics of the dynamics in the drilling process for indexable drills. This is an important field of research and highly valuable for the industry.

1.4 Aim of the Project and Research Questions

The main goal of this project is to understand the underlying mechanism that causes torsional-axial chatter and consequently unacceptable sound levels while machining using indexable drills. The focus of this study is to explain why and how chatter vibration is developed in indexable drills.

Note that vibration in metal cutting can have a serious detrimental effect on the surface finish and the quality of the workpiece. In case of torsional-axial chatter in indexable drills, the effect on the surface finish of the hole wall is marginal.

This thesis will study how differences in an indexable drill in comparison to other type of drills affects their behavior. The thesis will provide a description of the relationship between design parameters and dynamic behavior of drills and therefore, contribute to a more efficient product development process. Furthermore, this project aims to provide a guideline to developing the best strategy to reduce the level of sound in drilling with indexable drills.

1.5 Research Methodology

The objective is to reduce the level of vibration as much as possible and develop a systematic approach to compare different designs. In this thesis, a combination of experimental, analytical and numerical techniques is used to model the dynamic behavior of indexable insert drills. The methodology can be visualized by the coordinated approach as presented in Figure 1.6.
To take full advantage of the cross-benefits between simulations and experiments in the early product development stages, a coordinated approach provides a systematic way to assess and improve the product by a combined use of theoretical and physical models [53].

This study starts with experimental tests in the form of sound measurement to investigate the type of the vibration phenomena, described in detail in Paper D [4].

Prediction of static forces in drilling is an important part of this thesis because of their relevance to the dynamic behavior of indexable drills. This aspect is addressed in Paper A [54].

After developing a force model, the next requirement for simulating the dynamic behavior of the system is to identify the parameters which affect the system. A global damping ratio is considered in the model which is obtained from an impact test. The impact test is performed by clamping the drill in a holder which is mounted on a heavy, rigid table. The damping ratio obtained, together with Young's modulus, Poisson's ratio and densities of the tool materials are used in a three-dimensional finite element model to obtain frequency response functions. These frequency response functions (FRFs) are used to obtain dynamic
parameters by using modal analysis techniques. Equations of motions of the system are solved by Runge-Kutta methods to obtain the responses. This is addressed in Paper B [55] and Paper C [41]. A more detailed description of the workflow is shown in Figure 1.7. Based on the knowledge obtained, an optimization study was conducted targeting the positioning and the sizes of the coolant channels, Paper F.

Figure 1.7: The workflow shows the different activities in this thesis and how they are related.
Despite differences in design, a common feature of the hole making process by drilling is that material is removed from the workpiece by simultaneous angular and axial motions of the cutting edges. Thus, each infinitesimal segment on the cutting edge, relative to the workpiece, follows a helical path during the drilling operation. Assuming the length of the drill varies sinusoidally due to vibration, each segment leaves a wavy surface when it cuts the workpiece. In the next passage of the cutting edge, another wavy surface is generated and if these two waves are in phase as shown in Figure 2.1a, the thickness of the uncut chip remains constant. On the other hand, if these waves are not in phase as shown in Figure 2.1b, the uncut chip thickness varies, and this causes variations in cutting forces and induces vibrations.

In a more precise way the above mechanism can be represented by the block diagram of the chatter mechanism as shown in Figure 2.2 [11, 20].

where $\phi$ is the transfer function of the structure, $C$ is a cutting force coefficient which estimates the load generated per unit of chip thickness. $T$ is the time-delay between subsequent cuts. The above block diagram can be represented by the following equation [20]:

\[
\text{Dynamics of the Structure}\quad F(s) \quad \phi \quad y(s)
\]
The above equation is a delay differential equation, DDE, and the approach to a solution depends on the behavior of the time-delay. Changes in $C$ can transform it from a stable system to an unstable system. For a system with a single degree of freedom, SDoF, with a constant time-delay, $T$, the critical value of $C$ is obtained by [20]:

$$C_{\text{lim}} = -\frac{1}{2\text{Re}(\phi(f_c))}$$  \hspace{1cm} (2.2)

where $f_c$ is the chatter vibration frequency and Re is a function which returns the real part of transfer function. For orthogonal turning, $C$ is equal to $K_c b$, where $K_c$ is the cutting constant and $b$ is the depth of cut. $K_c$ and $b$ and therefore $C$ are all positive values. Therefore, in this case chatter occurs at frequencies higher than the natural frequency where the real part of the frequency response function, $\phi$, is negative.

Drilling is in several aspects different from orthogonal turning. The differences are described in the following sections.

### 2.1 Frequency of Chatter

In drilling, chatter usually occurs at a frequency below the natural frequency [10]. This can be further understood by looking at the SDoF system proposed by Bayly et al. [10] for torsional-axial chatter vibration in drilling. In their work $C$ is obtained from the following equation:

$$C = (C_2 + \theta_{Np} R_{av} C_1) b$$  \hspace{1cm} (2.3)

$C_1$ and $C_2$ are positive numbers which represent cutting constants in the axial and tangential directions and $C_1$ is larger than $C_2$. $R_{av}$ is the average radius of cut. $\theta_{Np}$ represents the coupling between torsional and axial deformation and is a negative number. In contrast to orthogonal turning, in this case $C$ is usually a negative number which leads to chatter below the natural frequency [10]. Although this was concluded for the SDoF model, this behavior is common and appears in more detailed models for torsional-axial chatter.

This can also be seen by rewriting Eq. (2.1) as follows:

$$m\ddot{y} + c\dot{y} + (k + C)y = C(h_0 + y(t - T))$$  \hspace{1cm} (2.4)

The left-hand side of Eq. (2.4) is a system in which the stiffness is changed by $C$. If $C$ is negative, the total stiffness of this system is less than $k$ and therefore the natural frequency is below $\sqrt{\frac{k}{m}}$. Combining this and the history of the system, in
the form of $y(t - T)$, affects the frequency of the chatter vibration. It is important to note that since $|a| \gg |b|$ the system has a static stability.

### 2.2 Depth of Cut

There is limited, or no control over the depth of cut in drilling. Here, the depth of cut is the difference between the radius of the hole and the radius of the pilot hole. In most practical cases, pilot holes are either short or are avoided, which leaves little room to influence the depth of cut. This eliminates the possibility of applying the most common chatter avoidance strategy; namely reducing the depth of cut. At the same time, it highlights the importance of modeling chatter vibrations instead of only the prediction of the chatter threshold.

### 2.3 Variable Time-delay

Generally, even if the spindle speed is constant, time-delay varies in almost all metal cutting operations because of the tangential (the direction of cutting) component of vibrations. However, in many turning and milling processes this variation is negligible in comparison to the constant part of the delay. In the model proposed by Bayly [10], the same assumption is made for drilling operations. The effect of the tangential vibration on the threshold of the chatter is limited. However, as shown in Paper C [41], the variation in time-delay as a result of the tangential vibration has a significant impact on modeling the drill vibration during chatter. Tangential vibration in drilling, which can also be called torsional vibration, makes Eq. (2.1) a so-called state-dependent delay differential equation. This state-dependency makes it difficult to solve the governing equation analytically. Therefore, a numerical approach is used for simulation in the time-domain. This numerical approach is explained in Paper B [55] and Paper C [41].

### 2.4 Limit on the Magnitude of Vibrations

In chatter vibrations, the amplitude of vibration can grow so much that the tool jumps out of the cut [56]. As mentioned in [56], forces are zero when the tool is out of cut which is a source of significant nonlinearity in the system. In the case of drilling with indexable drills, the tool does not usually jump out of the cut. This can be seen in high-speed filming of the process or by looking at the chips, for example as shown in Figure 2.3.

However, as shown in Paper C [41], the amplitude of vibration increases until the speed of tangential vibration surpasses the spindle speed. Under such conditions, there are moments in time when contact forces between “just-cut” waves and the insert prevent the vibration amplitude from growing further. This behavior
should be included in the model. These changes in contact forces in combination with the state-dependency of the time-delay increase the complexity of the system behavior. To address this in an appropriate manner, time domain simulation was used in this thesis for modeling the vibrations. The model is described in detail in Paper C [41].

![Figure 2.3: Typical shapes of the chips of the central insert (upper) and the peripheral insert (lower)](image)

### 2.5 Asymmetries in Indexable Drills

Most indexable insert drills have asymmetric design. One reason for this asymmetry is that the flutes are different for the central and peripheral inserts. The main reason why flutes are not designed with the same geometry is that they carry chips with different shapes. Typical chips for central and peripheral inserts are shown in Figure 2.3. This does not mean the shapes are limited to the ones shown in this figure, but it shows that in the same drilling process the different inserts generate different chip shapes, and therefore the flutes need to be designed with this in mind.

In addition to the asymmetry of the flutes, inserts are mounted at different locations on the drill. Because of this, the inserts have different axial displacements when the drill is subjected to a torque. Since the drill cross-section is not circular, it warps in twisting and therefore different parts of the cross-section will have different axial displacement.

Cutting at different speeds, in combination with differences in location and cutting width of the inserts, leads to differences in cutting forces, as described in Paper A [54].
2.6 Lower Helix Angle in Indexable Drills

The changes in the length of the drill at each point on the cross-section can be estimated by summation of the warp of the cross-section for a prismatic structure plus the influence of the helix angle. The latter is relatively large at greater helix angles. For example, in a long solid carbide drill the elongation of the drill close to its tip can be considered as a constant value over its cross section. Since the helix angle is relatively small in indexable drills in comparison to solid carbide drills, the effect of it is less. Figure 2.4 shows the helix on typical solid carbide and indexable drills.

Figure 2.4: Lower helix angle in indexable drills

A consequence of the lower helix angle is that the difference in the axial deformation is more pronounced for inserts which are not placed symmetrically. The axial deformation of the cutting edges for a solid carbide twist drill is shown in Figure 2.5, a typical indexable drill in Figure 2.6, and the similar indexable drill with straight flutes in Figure 2.7. As shown in these figures, the relative differences in axial displacement across the cutting edges becomes more pronounced as the helix angle reduces.
Figure 2.5: Axial displacement of the cutting edges in the first torsional-axial mode for a solid carbide drill.

Figure 2.6: Axial displacement of the cutting edges in the first torsional-axial mode for an indexable insert drill with a total pre-twist of 130 degrees.
Figure 2.7: Axial displacement of the cutting edge in the first torsional-axial mode for an indexable insert drill with straight flutes.

Note that even when the drill has straight flutes, the warp due to the non-circular cross-section is enough to make the drilling process unstable. Figure 2.8 shows three drills with diameter 22 mm and available cutting length of 22, 44 and 66 mm. All these drills have straight flutes. A sound measurement result is presented in Figure 2.9. It can be seen that chatter occurs for the two longer drills.

Figure 2.8: Straight-flutes drill, diameter: 22 mm, available cutting length: 22, 44 and 66 mm
2.7 Single Cutting Edge

In most indexable drills, two inserts are used for cutting; central and peripheral. The inserts are placed in a way that the overlap between the two inserts is very small and negligible. This means that drilling with indexable inserts can be considered as a single edge cutting operation.

2.8 Time-Domain Model

To develop a model for predicting vibration in drilling with indexable inserts, one can start with a general form of delay differential equation which governs regenerative chatter vibration in metal cutting.

\[ \mathbf{M} \ddot{\mathbf{u}} + \mathbf{C} \dot{\mathbf{u}} + \mathbf{K} \mathbf{u} = \mathbf{f}(\mathbf{u}, \mathbf{u}_{T_1}, \mathbf{u}_{T_2}, \mathbf{u}_{T_3}, \ldots) \]  

(2.5)

where \( \mathbf{M} \), \( \mathbf{C} \) and \( \mathbf{K} \) are mass, damping and stiffness matrices. \( \mathbf{f} \) is the force vector and is a function of the positions of the inserts in the current, \( \mathbf{u} \), and the past cut(s), \( \mathbf{u}_{T_i} \).
Since torsional-axial chatter occurs very close to the first torsional natural frequency of the drill, and due to the directions of the cutting forces and the orthogonality of the mode shapes, it is reasonable to assume that deflection of the drill is very similar to its first torsional mode. This assumption is also made in the work done by Bayly et al [10]. Note that the damping ratio is low in most cases, less than one percent. Figure 2.10 shows the deformation of a typical indexable drill in the first torsional mode.

Figure 2.10: Deflection of the drill in its first torsional mode

As can be seen in Figure 2.10, this mode is a combination of torsional and axial motions. Note that in general, axial and torsional deformation can be coupled to bending deformation if the center of shear is not located at the cross-sectional center of mass [50]. In the drills studied in this thesis, the assumption is that the coupling between torsion and bending is small and has a negligible effect on the chip thickness. Considering torsional and axial deformations for two inserts, in total four degrees of freedom for the system need to be included in the dynamic model.

The right-hand side of Eq. (2.5) is the force vector. Since it is assumed that bending is uncoupled from torsional deflection, bending forces are not included in the model. Therefore, the force vector only includes the torque and axial forces of each insert. In addition, a linear force model is used to estimate the cutting forces from the thickness of the chip. More details and discussion about the choice of force model are given in Paper E.

To estimate the chip thickness, in addition to the current positions of the insert, its position in the previous cuts (given in \(u_{T_1}\)) also needs to be considered. More precisely, chip thickness for each insert is calculated from the following equation (ref: Paper C [41]):
When the uncut chip thickness is estimated, the cutting force is calculated and the
displacement of the system in the next time step is estimated. One method to
approximate the response of a dynamic system is a numerical integration of the
corresponding differential equations. Due to computational advantages of the
fourth order Runge-Kutta method, known as RK4, this method is used in this
work for solving the system of equation of motion. To apply this method a new
vector, \( \mathbf{q} \), is introduced as shown in Eq. (2.7).

\[
\mathbf{q} = \begin{bmatrix} \mathbf{u} \\ \dot{\mathbf{u}} \end{bmatrix}
\]

(2.7)

Following the RK4 method, \( \mathbf{q} \) is calculated as shown in Eq. (2.8).

\[
\mathbf{q}[n] = \mathbf{q}[n-1] + (k_1 + 2k_2 + 2k_3 + k_4) \frac{\Delta t}{6}
\]

(2.8)

\( k_1-k_4 \) are obtained as shown in Eq. (2.9).

\[
\begin{align*}
k_1 &= g(t[n-1], q[n-1]) \\
k_2 &= g(t[n-1] + \frac{\Delta t}{2}, q[n-1] + \frac{\Delta t}{2} k_1) \\
k_3 &= g(t[n-1] + \frac{\Delta t}{2}, q[n-1] + \frac{\Delta t}{2} k_2) \\
k_4 &= g(t[n-1] + \Delta t, q[n-1] + \Delta t k_3)
\end{align*}
\]

(2.9)

where

\[
\dot{\mathbf{q}} = g(t, \mathbf{q})
\]

(2.10)

As mentioned previously, the amplitude of vibration increases until the speed of
vibration reaches the spindle speed, which cause a sudden change in the forces.
This yields nonlinear forces and affects the simulation routine. The modeling of
these nonlinear forces is described in Paper C [41].
3 Static Cutting Force

Drilling with indexable inserts in a conventional metal cutting operation involves removing a thin layer of workpiece material in each tool revolution. This generates forces known as cutting forces. These forces vary as the tool starts to penetrate the workpiece. However, in most drilling cases when the tool is fully engaged the average and RMS force values do not change significantly until the drill emerges from the workpiece. As an example, a measured thrust force is shown in Figure 3.1.

![Figure 3.1: An example of an experimentally measured thrust force](image)

The amplitude and direction of cutting forces depends on a number of different parameters; for example, workpiece material, cutting speed, insert geometry, uncut chip thickness and chip width. The relationship between these parameters and cutting force is of interest to the machining industry because it allows prediction of the cutting force. Prediction of the cutting force is important because cutting force cause deflection in the machine-tool-workpiece system, and it affects tool life and power consumption. A model which is capable of predicting cutting force accelerates the tool design phase by reducing the number of prototypes, and improves planning for drilling operations by suggesting improved cutting parameters.

One common technique for modeling static cutting force is to divide the cutting edge into smaller elements and using experimental, analytical or numerical approaches to calculate elemental forces. Combining these forces gives the total force. Armarego and Cheng in [57, 58] proposed an elemental approach to calculate cutting force by dividing the cutting edge into small elements. A similar
approach was used by Watson in [59] to predict cutting force produced by drill lips in twist drills. Predicting cutting force by dividing the cutting edges into smaller elements gives the possibility of modeling a wide range of cutting shapes. It is therefore a common practice and has been used in other machining applications such as turning, milling and boring.

Kaymakci et al. in [60] presented a unified model which uses the approach of dividing cutting edges into smaller elements to predict cutting force in several metal cutting operations including drilling. Later this model was used by Parsian et al. in [54] to propose a matrix equation which relates the geometries of the cutting edges, workpiece material and cutting forces in drilling with indexable insert drills. The proposed relationship can be used to predict cutting force by using a set of cutting coefficients. Using this model, cutting coefficients can be obtained from a small set of drilling operations by using the fact that most indexable drills, due to their asymmetric design have an unbalanced lateral force. Therefore, the model uses the lateral force in combination with the thrust force and the torque to obtain the required coefficients.

3.1 Mechanistic Modeling

One approach to model cutting forces is by a mechanistic method which can be found in the literature [20]. In this method forces are expressed as functions of equivalent uncut chip width ($L_e$) and uncut chip thickness ($h_e$) using a set of coefficients. By obtaining these values from a set of experiments, it is possible to predict forces in different uncut chip thicknesses and uncut widths. The cutting coefficients are obtained experimentally. The first step to estimate these coefficients is to measure cutting forces at different feed rates and then to calculate the average forces at each feed rate and finally to fit a function (usually a linear polynomial) to the discrete points. If a linear polynomial is used, the relationship will have a generic form as shown in Eq. (3.1).

$$F = a \cdot h_e \cdot L_e + b \cdot L_e$$  \hspace{1cm} (3.1)

While the equivalent uncut chip width and thickness are dependent on geometries, the cutting coefficients incorporate the effect of many parameters which influence the cutting forces, for example workpiece material and the edge radius of the insert. Although using these coefficients is convenient because of the simple relationships between them and cutting forces, there are many restrictions on their wider application since they are restricted to specific cutting conditions and geometrical features of the inserts. If any of these factors change the coefficients must be re-estimated using a new set of experimental data. Changes include for example a different workpiece material, change of coating or changes in the cutting-edge geometry. The dependency of cutting coefficients on
insert geometries is a drawback when using these coefficients in the design process because the coefficients can be calculated only when the tool has been produced and used to measure the cutting forces. This is not efficient from a tool design point of view because a lot of prototyping is required before an optimized design can be achieved. To overcome this problem, the approach of dividing the cutting edges into smaller elements is used as described in Paper A [54] to reduce the dependency of the cutting coefficients on the geometries and the positions of the cutting edges.

### 3.2 Dividing the Cutting Edges

This is an approach to separate the effect of cutting edge profile from the cutting coefficient. As shown in Figure 3.2, the edges for the central and the peripheral inserts can be different. Dividing the cutting edges into smaller elements provides a possibility to deal with variations in edge profile in a systematic way.

![Figure 3.2: (a) central insert, (b) peripheral insert](image)

The method consists of dividing the cutting edges into small elements, computing the forces on each element and finally combining the elemental forces to obtain the total cutting force.

The force on each element is calculated using a set of cutting coefficients which are the same for all elements on a specific edge. Friction and normal forces on element $i$, $\Delta F_{u}^{i}$ and $\Delta F_{v}^{i}$ respectively, are calculated on the surface of the element using four cutting coefficients as shown in Eqs. (3.2) and (3.3) [60]. The forces are shown in Figure 3.3a.

$$\Delta F_{u}^{i} = K_{uc} \cdot h^{i} \cdot L^{i} + K_{ue} \cdot L^{i}$$

$$\Delta F_{v}^{i} = K_{vc} \cdot h^{i} \cdot L^{i} + K_{ve} \cdot L^{i}$$

When the forces are calculated on each element, they are transformed into a global coordinate system ($\Delta F_{x}^{i}$, $\Delta F_{y}^{i}$ and $\Delta F_{z}^{i}$ as shown in Figure 3.3b and summed up to estimate the total cutting force on the tool.
3.3 Total Force

As shown in Paper A [54], when using an indexable drill with two inserts, the process of summation of elemental forces to obtain the total force can be summarized in a matrix equation as shown in Eq. (3.4).

\[
\mathbf{C}_{8 \times 8} \mathbf{a}_{8 \times 1} = \mathbf{b}_{8 \times 1} \tag{3.4}
\]

where \( \mathbf{C} \) is a square matrix which contains geometric information about the edge profiles, \( \mathbf{a} \) is a vector representing cutting coefficients of inserts (\( K_{uc}, K_{ue}, K_{vc} \) and \( K_{ve} \)) and \( \mathbf{b} \) is a vector representing linear coefficients of the forces.

3.4 Modeling Forces in Backward Rotation

In addition to causing a variable delay, angular vibrations cause another phenomenon which is backward rotation. The incremental relative rotation between the cutting edge and workpiece at each time interval is the summation of the rigid body motion and the torsional vibration. At some instance, the speed of backward rotation due to torsional vibration is greater than the forward rigid body motion and therefore the edge experiences a backward angular motion. In such moments, the cutting edge does not cut on its rake face. The method proposed in Paper B [55] is used in this thesis to simulate the forces under such conditions.

The basic assumption is that if the cutting edge touches the surface with its flank face during backward rotation, it rubs over the surface and therefore the part of the force function which is dependent on the uncut chip thickness does not contribute to force generation. One approximation is to use the edge coefficients when the direction of speed is negative, as is proposed in Paper B [55].
Additionally, during backward rotation the direction of torque will be opposite to that during forward rotation. Considering these two modifications, the axial force and the torque for an insert at the $n^{th}$ time step can be calculated as shown in Eqs. (3.5) and (3.6) [41]. More details are presented in Paper C [41].

\[ T_i^n = \begin{cases} a_i^t h_i^n + b_i^t & \text{if } h_i^n > 0 \text{ and } \Delta \theta_i^n + \Omega \geq 0 \\ -b_i^t & \text{if } h_i^n > 0 \text{ and } \Delta \theta_i^n + \Omega < 0 \\ 0 & \text{if } h_i^n \leq 0 \end{cases} \]  

\[ F_z^n = \begin{cases} a_i^f h_i^n + b_i^f & \text{if } h_i^n > 0 \text{ and } \Delta \dot{\theta}_i^n + \Omega \geq 0 \\ -b_i^f & \text{if } h_i^n > 0 \text{ and } \Delta \dot{\theta}_i^n + \Omega < 0 \\ 0 & \text{if } h_i^n \leq 0 \end{cases} \]  

In the above equations, $\Omega$ is the angular speed of the spindle in radians per second.

### 3.5 Applications of the Model

Using the proposed model, the contribution of each insert to the cutting force can be estimated, as required for a dynamic simulation. The main reason for developing this force model was to improve chatter simulation. However, other benefits of the model are:

- It gives the distribution of cutting forces on the edges. This improves the simulation of drill deflection.
- It makes it possible to obtain the contribution of each insert and use this in designing insert seats.
- The forces can be used to choose an appropriate machine, fixture, adapter, etc.
- Forces can be used for power consumption estimation.
- Forces at the entry and exit can be estimated.

An unbalanced lateral force causes a deflection which can, depending on its direction result in a larger or smaller hole than the nominal value. In some cases, this lateral force can be so large that the hole dimension exceeds acceptable tolerances. Therefore, prediction of this force is beneficial in the tool design process.

Another advantage of the model is that it makes it possible to calculate the forces at the entry and exit of the hole. How the drill enters the workpiece plays an important role in drilling operations and its final quality. Usually these drills are designed in a way to have a small lateral force in the case of full edge engagement. However, this does not mean that the lateral forces are small at the moment of entry and exit. During entry and exit, inserts are partially engaged, which in some
STATIC CUTTING FORCE

cases results in large unbalanced forces. Therefore, a good estimation of the magnitude of these forces is crucial in the design of indexable drills.
4 Dynamic Parameters

When a mechanical system is excited with a dynamic force, it will respond with a dynamic motion. The response is a function of the loading and dynamic parameters of the system. Identification of these dynamic parameters is a prerequisite to obtain a system model. In the case of torsional-axial chatter vibration, dynamic parameters associated with the first torsional resonance frequency of the clamped drill must be estimated.

To extract dynamic parameters, namely modal mass, stiffness and damping, different methods including analytical, experimental modal analysis, three-dimensional numerical methods or extended two-dimensional numerical methods can be used. Each of these has its own advantages and disadvantages.

The experimental approach might seem more accurate because it does not have many of the simplifications that are made during derivation and numerical solution of mathematical models. However, these experimental methods are also based on many assumptions which can have significant effects on their accuracy. For example, given the hammer impact test which is the most common experimental method, the exact point measured is uncertain. Another disadvantage with the experimental approach is the need for physical drills to conduct the experiments. From a product development point of view, this means a significant cost for prototyping and evaluating each modification.

Three-dimensional numerical approaches such as finite element have a good accuracy for modal parameter estimation because the drill is relatively weak in comparison to most machine tools. Therefore, assuming a fixed-free boundary condition for the drill gives accurate results. However, the damping ratio still needs to be estimated experimentally.

Despite its benefits, the three-dimensional finite element method, 3D-FEM, is computationally demanding, which can be a disadvantage for optimization applications. A faster approach is the so-called three-dimensional spectral-Tchebychev, 3D-ST, which is presented by Filiz et al. in [50, 61] and Bediz et al. in [62]. In this technique, a mapping is constructed to relate the simply connected cross-section of the beam to a simpler rectangular domain. In indexable drills, the cross-section is not simply connected because of coolant holes. Coolant holes affect the torsional and the torsional-axial rigidities of the drill, as has been discussed in detail in Paper F.

Extended two-dimensional numerical approaches offer other alternatives to estimate the required dynamic parameters. Being faster compared to other...
methods is the main advantage of these approaches. A disadvantage is their lower accuracy, specifically with large helix angles [63, 50]. These approaches are usually based on the Saint-Venant theory for torsion, modified to include the helical shape of the structure.

The Saint-Venant theory offers promising results for estimation of torsional rigidity of the prismatic bars. It can be formulated in one the following forms [64, 65]:

\[
\nabla \cdot (G \nabla \Psi) = 0
\]
\[
\tau_{nz} = n_x \tau_{xz} + n_y \tau_{yz} = 0 \text{ on } \partial \Gamma
\]

(4.1)

\[
\nabla \cdot \left( \frac{1}{G} \nabla \Phi \right) = -2\theta
\]
\[
\Phi = \text{Constant on } \partial \Gamma
\]

(4.2)

where \( G \) is the shear modulus, \( \theta \) is the twist caused by the applied torque, \( \Gamma \) is the domain, \( \partial \Gamma \) is the boundary of the domain. \( \Psi \) is the warp function and \( \Phi \) is called the Prandtl stress function. After estimating one of these over the domain, the torsional rigidity can be calculated according to [64, 65].

Eqs. (4.1) and (4.2) have analytical solutions for a homogeneous material and for simple geometries such as an ellipse and rectangle which can be found in the literature, for example in [66]. For a homogenous material, Eq. (4.1) becomes the Laplace equation; therefore, if the second order partial derivatives are continuous, \( \Psi \) then is a harmonic function [67] and complex analysis can be used to solve the equation over more complicated geometries. This method has been discussed in for example [67]. In the case of the drill geometry, the only options are usually numerical which are given for example in [64, 65]. Figure 4.1 shows the cross-section of a typical indexable drill and the corresponding numerically computed Prandtl stress function.
After calculation of $\Psi$, or $\Phi$, the torsional-axial rigidity can be estimated by including the twist angle, for example by the method proposed by Hodges in [52]. However, the accuracy of this method reduces as the helix angle increases. Therefore, its application should be limited to the cases where the helix angle is small. In case of indexable insert drills, the helix angles are usually small in comparison to solid-carbide or exchangeable-tip drills. Therefore, it is expected that the extended 2D methods provide acceptable results in most cases. This allows faster parametric studies. An example of such a study is presented in Paper F, where the effect of coolant hole positioning on torsional and torsional-axial rigidities is evaluated.

The suggestion is to use these extended 2D techniques for optimisation and parametric studies; and for a detailed time-domain simulation to use 3D-FEM to extract mode-shapes and resonance frequencies. The damping ratio can be estimated by experimental approaches, as a global parameter. In this work, frequency response functions, obtained from 3D-FEM are used to describe the dynamic parameters of the system.

4.1 Frequency Response Functions

If a linear dynamic system is subjected to a pure sinusoidal force, the responses of the system are sinusoidal motions which have the same frequencies as the input signal but might have different amplitudes and phases [68]. For a pair of excitation and response, at each frequency of excitation, the difference between phases and the ratio between amplitudes of output and input signals can be represented in the form of a complex number. A frequency response function (FRF), contains these complex numbers expressed as a function of the excitation frequency. Each frequency response function is associated with an input, an excitation force at $q,$
and an output, a measured displacement, speed or acceleration at \( p \), which can be mathematically represented as follows if there is only a single mode in the model:

\[
H_{pq}(\omega) = \frac{u_p(\omega)}{F_q(\omega)} = \frac{1}{k_{pq}} \cdot \frac{\omega_n^2}{\omega_n^2 - \omega^2 + 2j\zeta \omega_n \omega}
\]  

(4.3)

\( \omega_n \) is the undamped natural frequency, \( \zeta \) is the damping ratio and \( k \) is the modal stiffness. A basic assumption in using FRFs is that the system is time-invariant which means its characteristics remain constant over time. In drilling, due to chips carried in the drill flutes the system dynamics varies over time, but because of the marginal mass of the chips compared to the mass of the drill, it is assumed that these variations are not significant and the system is assumed to be time-invariant.

Usually more than one FRF is required to describe the dynamics of a system. Therefore, a starting point for identification of dynamic parameters is to decide how many and which FRFs are required. The number of FRFs required depends on the number of degrees of freedom that need to be considered. The aim is to choose only the FRFs which are needed to describe chatter in the drill with an acceptable accuracy. The model for torsional-axial chatter in indexable drills has four degree of freedoms; torsional and axial motions for the two inserts. Therefore, 16 FRFs must be estimated. Due to reciprocity, this number can be reduced to ten. The following figure shows the magnitude of two of these FRFs; where the input is the axial force and the torque generated by the central insert and the output is the axial displacement of the central insert. The diameter of the drill is 24 mm and the length is 96 mm. After extracting all required FRFs, modal parameters are estimated and used in the time domain simulation presented in Chapter 5 and Paper C [41].
Figure 4.2: Magnitude of two FRFs for a drill with diameter of 24 and length of 96 mm. The output is the axial displacement and the input is the torque and the axial force of the peripheral insert.
5 Simulation and Measurement

Having all the necessary computational tools, presented in previous chapters, it is now possible to simulate the vibration produced by regenerative chatter. An indexable insert drill with two inserts is used both for simulation and measurement. The diameter of the drill is 24 mm and the length of the drill 96 mm. Eqs. (5.1) to (5.4) are used to express axial forces and torques as functions of chip thicknesses, Paper C [41].

\[
\begin{align*}
F_z^c[n] &= -2178 \, h^c[n] - 1279 \\
F_z^p[n] &= -2040 \, h^p[n] - 1201 \\
T^c[n] &= -29.5 \, h^c[n] - 2.3 \\
T^p[n] &= -81.4 \, h^p[n] - 6.4
\end{align*}
\]

The simulation is done for a drilling operation at a feed rate of 0.1 mm/rev and cutting speed of 200 m/min.

To verify the results a drilling operation using the same cutting conditions was performed and the tangential speed of a point on the edge of the central flute was measured with a laser. The measurement-setup is shown in Figure 5.1.

![Measurement setup](image)

Figure 5.1: Measurement setup

The laser beam makes an angle of \( \theta \) which is larger than 90 degrees with the radial direction of the drill as shown in Figure 5.2. Therefore, the measured speed must be divided by \( \sin(\theta) \) to obtain the tangential speed of the vibration.
Note that the simulation results on the edge of the drill need to be scaled by the mode shape to obtain the simulated tangential speed at the measurement point. The simulation methodology is described in Paper C [41]. Some of the results are shown in Figure 5.3 and Figure 5.4.

The model provides the possibility of separating the contribution of forces on each insert to the displacement of the drill. For example, Figure 5.4 shows the axial vibrations of each insert due to the torque and the axial forces generated by the central and peripheral insert.
Figure 5.4: Simulation results for axial vibrations of the inserts, Paper C [41].
6 Investigation of Possible Methods to Suppress Vibrations

To avoid chatter in metal cutting, the depth of cut must be less than a critical depth. The oldest and most common approach is to reduce the depth of cut until the process become stable. In the case of drilling with indexable drills with two inserts, the depth of cut is equal to the radius of the hole. Therefore, the only way to reduce the depth of cut is to first make a smaller pilot-hole, and after that to cut with a drill of the desired diameter. Because of the asymmetric design of indexable drills, the presence of a pilot-hole has a significant effect on the resultant lateral forces. This usually causes excessive bending and severe lateral chatter. As shown in Paper D, even in the short time-period when drilling is starting, a lateral chatter occurs. In the entry phase, first one insert engages the cut and as the second insert starts engaging and the lateral forces decreases, the lateral chatter disappears. This can also be seen from high speed filming of the process. Figure 6.1 shows the chatter marks on the chips because of lateral chatter during entry and torsional chatter when both inserts are engaged. Having a pilot hole creates a large lateral force and consequently lateral chatter. Therefore, the use of pilot holes is not applicable in drilling with these types of drills.

![Figure 6.1: Chatter marks on the chips; generated by (a) lateral chatter during entry; (b) torsional-axial chatter, when both inserts are engaged.](image)

As discussed in Paper E, although variation in delay because of torsional vibration changes the chattering behavior, it has a limited effect on the stability limit.
Therefore, the governing rules for constant time-delay equations can to some extent be applied to drilling.

**6.1 Chatter Suppression Approaches**

According to Eq. (2.2), one approach to increase the stability limit is by reducing the amplitude of the real part of the frequency response function for frequencies that are close to the natural frequency. In the case of drilling with indexable drills, this means for frequencies close to but smaller than the first torsional natural frequency of the drill. Eq. (2.2) can be rewritten as follows:

\[
C_{lim} = \frac{k}{2} \left[ r_c^2 - 1 + \frac{4\zeta^2 r_c^2}{r_c^2 - 1} \right]
\]  

(6.1)

where \( r_c \) is the normalized chatter frequency \((f_c/f_n)\).

As shown in Eq. (6.1), \( r_c^2 - 1 \) and \( C_{lim} \) must have the same sign on both sides of the equation. This means that in case of negative \( C_{lim} \), \( r_c^2 \) is less than unity which means \( f_c \) is lower than \( f_n \) which is the case in torsional-axial chatter in most drilling operations. Eq. (6.1) offers two important approaches for increasing the critical depth of cut, namely increase the damping and increase the stiffness. These are shown as approaches 1 and 5 in Table 6.1.

One approach to reduce the effect of the sound on the human auditory system is to shift its frequency to above 20 kHz. In some short drills, the chatter frequency is currently around 10 kHz. This is called approach 2 in Table 6.1.

Change in the specific cutting pressure and preventing wave modulation are the other two approaches which are given in Table 6.1. These approaches can be divided into further methods as presented in Table 6.1.

<table>
<thead>
<tr>
<th>Approach</th>
<th>No.</th>
<th>Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Increase modal stiffness</td>
<td>1</td>
<td>1 Reduce the torsional-axial coupling</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>2 Increase elasticity modulus of the drill body</td>
</tr>
<tr>
<td>Shift frequency</td>
<td>3</td>
<td>3 Increase modal stiffness to modal mass ratio</td>
</tr>
<tr>
<td>Reduce ( K_c )</td>
<td>4</td>
<td>4 Reduce ( K_c )</td>
</tr>
<tr>
<td>Prevent the wave modulation</td>
<td>5</td>
<td>5 Use spindle speed selection</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>6 Use a variable pitch angle</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>7 Use spindle speed variation</td>
</tr>
<tr>
<td>Increase damping</td>
<td>8</td>
<td>8 Increase the damping of the tool material</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>9 Increase the process damping</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>10 Use an external damper</td>
</tr>
</tbody>
</table>
Method 1. Reduce the coupling
This can be done by reducing the helix angle as much as possible and by increasing the rigidity of the drill. The first measure on its own is not enough to prevent chatter. Even completely straight flute drills suffer torsional-axial chatter, for example as presented in Figure 2.9 and Paper D.

Increasing the torsional rigidity of drill is limited because of the chip flutes and coolant channels in the cross-section. Paper F proposes an approach to reduce the negative effects of the coolant channels as much as possible by selecting the appropriate locations for them.

Method 2. Increase the modulus of elasticity
This is highly limited by commercially available material and by considering the economic aspect.

Method 3. Increase the modal stiffness to modal mass ratio
The aim of applying this method is to increase the first torsional-axial natural frequency to a value above 20 kHz, so it will not be heard by the operator’s auditory system. This might be achievable in shorter drills by reducing the mass without a significant reduction of the stiffness.

Method 4. Reduce the cutting coefficients
This is difficult and governed by many aspects which are outside the scope of this thesis. A change in design of the insert might give a limited level of improvement. However, during operation the edges become deformed because of wear, which is difficult to control.

Method 5. Use the spindle speed selection method
This is a very popular method in milling and turning and works based on the so-called stability lobe diagrams. This diagram shows the critical depth of cut versus spindle speed. It is usually possible to achieve higher critical depth of cut by choosing an appropriate spindle speed. In most drilling operations with indexable drills, the frequency of vibration is relatively high in comparison to the tooth-path frequency. Therefore, the process lies in the area where these lobes are very packed and leave no practical room to utilize this diagrams. The following figure shows the stability diagram for an indexable insert drill with a diameter of 24 mm and length of 96 mm when the Bayly et al approach [10] is used to obtain the stability lobes. The workpiece material is SS 2541, resonance frequency is 4091 Hz, $aKC$ is -4731 N/mm² and stiffness is 4440 kN/mm.
As can be seen the lobes are packed and the critical depth of cut is very low compared to the depth of cut which is equal to the radius of the drill. To gain a benefit from this technique, either the cutting speed needs to be very high or the first torsional resonance frequency must be low.

Method 6. Use a variable pitch angle
This is a common approach in milling [69, 70, 71] and multi-edge boring operations. The main idea is to adjust the pitch angles between edges in a way that interrupts the chatter mechanisms. A prerequisite of this approach is that more than one edge passes the same cutting path. In indexable insert drills with two insert, on each radial distance only one of the inserts cut the material, except for the small overlap. Therefore, this approach is not possible with current configurations. A possible approach can be to add a small edge just before the main edge to cause variable pitch. However, due to its limits this does not give promising results.

Method 7. Use the spindle speed variation methods
Because of the relatively high vibration frequency in many drills (greater than 3 kHz), spindle speed variation needs a fast response controller system. In addition, rapid speed variations makes a high power-demand on the spindle. In milling, turning and boring usually this approach is used to increase the critical
depth of cut and even a few percentage increase can be considered as an improvement. In drilling, the depth of cut is a fixed value, and improvement is achieved only if the critical depth of cut can be increased to be above this fixed value.

**Method 8. Increase damping of the tool material**
Currently the drills are made of steel with homogenous properties and the damping ratio is low for the steels available. Additive manufacturing may enable production of structures with higher damping through non-homogenous structures.

**Method 9. Increase the process damping**
Process damping occurs when the flank face ploughs into the material. This increases the stability of the process, especially at lower cutting speeds. The effect of this on chatter has been studied in several works including [72, 73, 74, 75]. Process damping is more effective if the chatter marks have a shorter wavelength. The wavelength decreases as the ratio between chatter frequency and spindle rotation frequency increases. If this ratio is above 10 the process is in the so-called “process damping zone” [76]. In the majority of drilling processes with indexable drills cutting occurs in this zone. Therefore, increase in process damping might be considered as a solution. However, by considering the manufacturing tolerances and consequently the risk of increase in lateral forces, it does not seem to be a suitable method for indexable drills.

Here it is important to note that unbalanced lateral forces can occur both in asymmetric indexable drills and symmetric drills. However, in the case of symmetric drills the magnitude of the unbalanced forces is usually smaller and always results in larger holes in comparison to the drill size. In the most common case of indexable drills with two inserts the unbalanced lateral force, depending on its direction, can result in both larger and smaller holes, relative to the drill diameter. The reason is that the wall of the hole is only made by the corner of the peripheral insert. If the resultant force pushes this corner toward the center, the holes becomes smaller than the diameter of the drill. In a symmetric drill, while one edge is pushed toward the center, the other edge is pushed outward. The risk with creating a smaller hole is that it will be difficult to retract the drill from the hole. Therefore, indexable drills are intentionally designed with an unbalanced force to guarantee that the hole becomes slightly larger than the drill. Increasing the effect of the process damping, which already exists, makes it very difficult to control the unbalance force.

**Method 10. Use a tuned mass damper**
Although different external damper system can be considered for improving the stability of the drilling process, only the tuned mass damper, TMD, is studied in
this thesis due to its promising results. A tuned mass damper can significantly increase the damping of a system. The main challenge of its application in drilling is to embed it in the drill. The drills need to go through the hole and at same time deliver coolant to the cutting zone and evacuate cutting chips. This leaves a very limited room for a tuned mass damper. However, because of its effectiveness in combination with improvement in the geometry of the drill body, it is considered as the most feasible candidate among different methods to suppress chatter. Therefore, the next section is dedicated to explaining how this system affects chatter.

6.2 Tuned Mass Damper

In this method, a smaller system is added to the main system to increase the total damping of the system. The mechanism of how this system works to suppress chatter is describe in the literature for example by Sims [77]. If a tuned mass damper is used in the system, equations of motions are written as follows,

\[
\begin{align*}
M_1 \ddot{y}_1 + B_1 \dot{y}_1 + K_1 y_1 &= F + B_2(\dot{y}_2 - \dot{y}_1) + K_2(y_2 - y_1) \quad (6.2) \\
M_2 \ddot{y}_2 + B_2(\dot{y}_2 - \dot{y}_1) + K_2(y_2 - y_1) &= 0 \quad (6.3)
\end{align*}
\]

where \(M_1, B_1\) and \(K_1\) are the dynamic parameters of the system and \(M_2, B_2\) and \(K_2\) are the dynamic parameters of the damper.

Laplace transform of Eqs. (6.2) and (6.3) gives:

\[
\begin{align*}
(M_1 s^2 + B_1 s + K_1)Y_1 &= F + B_2 s(Y_2 - Y_1) + K_2(Y_2 - Y_1) \quad (6.4) \\
M_2 s^2 + B_2 s(Y_2 - Y_1) + K_2(Y_2 - Y_1) &= 0 \quad (6.5)
\end{align*}
\]

Eqs. (6.4) and (6.5) give:

\[
\phi = \frac{Y_2}{F} = \frac{(M_2 s^2 + B_2 s + K_2)}{((M_1 s^2 + (B_1 + B_2)s + K_1 + K_2)(M_2 s^2 + B_2 s + K_2) - (K_2 + B_2 s)^2)} \quad (6.6)
\]

\(\phi\) given in Eq. (6.6) can be used in Eq. (2.2) to evaluate critical depth of cut. This is the frequency domain approach and leads to stability lobes.

It is also possible to perform the evaluations in the time domain. In this case force is written as follows, which is a common way to describe the regenerative effect [20].

\[
F = C (h_0 + y_1^{delayed} - y_1) \quad (6.7)
\]

\(y_1^{delayed}\) is \(y_1(t - T)\). Substituting Eq. (6.7) in Eq. (6.2) gives:

\[
\ddot{y}_1 = -\frac{B_1 + B_2}{M_1} \dot{y}_1 - \frac{K_1 + K_2 + C}{M_1} y_1 + \frac{B_2}{M_1} \ddot{y}_2 + \frac{K_2}{M_1} y_2 + \frac{C}{M_1} y_1^{delayed} + \frac{C h_0}{M_1} \quad (6.8)
\]
Eq. (6.3) can be re-written as follows:

\[
\ddot{y}_2 = -\frac{B_2}{M_2} (\dot{y}_2 - \dot{y}_1) - \frac{K_2}{M_2} (y_2 - y_1) \tag{6.9}
\]

By defining \( q = [y_1 \ y_2 \ \dot{y}_1 \ \dot{y}_2]^T \) as the state vector, and by combining Eqs. (6.8), (6.9):

\[
\dot{q} = Aq + b
\]

\[
A = \begin{bmatrix}
0 & 0 & 1 & 0 \\
0 & -\frac{K_1+K_2+C}{M_1} & \frac{K_2}{M_1} & -\frac{B_1+B_2}{M_1} & \frac{B_2}{M_1} \\
-\frac{K_2}{M_2} & 0 & -\frac{B_1+B_2}{M_2} & 0 & -\frac{B_2}{M_2}
\end{bmatrix}
\]

\[
b = \begin{bmatrix}
\frac{C}{M_1} y_1^{delayed} + \frac{C h_0}{M_1} \\
0 \\
0
\end{bmatrix}
\]

Eq. (6.10) is a delay differential equation and can be solved by the method described in [78].

This approach can be used to obtain the optimized parameters for a tuned mass damper. Sims in [77] proposed the optimum values for TMD as follows:

\[
\left(\frac{f_{TMD}}{f_n}\right)^2 = \frac{\mu + 2 - \sqrt{2\mu + \mu^2}}{2(1+\mu)^2} \tag{6.11}
\]

\[
\zeta_{opt} = \frac{3\mu}{8(1+\mu)} \tag{6.12}
\]

\( \mu \) is the mass ratio [77]. Applying the above equations to the drill discussed in Method 5, for \( \mu = 0.0005 \) gives \( \zeta_{opt} = 0.0137, \frac{f_{TMD}}{f_n} = 0.991 \). The result is shown in Figure 6.3. As is shown the stability limit increases to a value greater than the critical depth of cut of this drill, 12 mm.
Figure 6.3: Stability lobe based on frequency domain analysis when the tuned mass damper is used.
7 Conclusions

The main objective of this thesis was to investigate and model the mechanisms which cause vibration and undesirable sound while drilling with indexable insert drills, and to provide suggestions for reducing or eliminating this unwanted phenomenon. An initial investigation based on measurement and analysis of the sound shows that it is caused by a torsional-axial regenerative chatter mechanism. Details and interpretation of the sound measurement results are presented in Paper D [4].

The vibration of the drill was described with a system of delay differential equations. The time-delay of the system is variable due to angular vibrations of the drill. A time domain approach was used to cope with dependency of the time-delay on the state of the system. The model is described in Paper B and Paper C [55, 41]. The proposed time domain model was used to simulate the drill vibrations during torsional-axial chatter. The simulations show that a variation in cutting torque is the main contributor. Torque variations affect the length of the drill because of torsional-axial coupling. Torsional-axial coupling is mainly a result of the non-circular cross-section and the twist in the drill body. Because of the direction of the twist, an increase in the cutting torque will elongate the drill. In most cases, this behavior causes chatter vibrations to occur at frequencies below the natural frequency.

The simulation also shows that the level of vibration is limited by the spindle speed. The amplitude of the angular vibration increases until it surpasses the angular speed of the spindle. This creates time-intervals where the motion of the cutting edge is opposed by the force created by just-cut surface. Measurements confirm the simulation results and show that the angular speed of the spindle limits the amplitude of the vibration.

The time domain simulation provides an in-depth knowledge into the cutting process; for example, the magnitude of the force generated by each insert; how the force varies and how the edges move during the cutting operation.

Since adding a pilot-hole causes a large lateral force and consequently results in severe lateral chatter, reducing the depth of cut to avoid chatter is not applicable when machining with indexable insert drills. Other approaches to eliminate chatter including spindle speed selection, spindle speed variation, increase in the modal stiffness, shift in the frequency and increase in the damping are discussed with details in the previous chapter. The method of using a tuned mass damper is a promising candidate for this application. The most challenging aspect is embedding a tuned mass into the drill body. This will need further investigation.
CONCLUSIONS
8 References


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Regenerative Chatter Vibration in Indexable Drills: Modeling and Simulation

An indexable insert drill is a drill which uses cutting inserts to make holes. Undesirable sound generated by this type of drill has always been considered as a problem in workshops. The focus of this thesis is to investigate the mechanism behind these vibrations, to model it and to provide guidelines for reducing the sound in future drill designs. Primary investigations show that the main sound-generating mechanism is a self-induced vibration due to a coupled torsional-axial deformation in the drill which leads to the torsional-axial chatter vibration. The first step of simulating regenerative chatter vibrations in a drill is to model the static cutting forces. In this thesis, a model is proposed to estimate static cutting forces in indexable drills by dividing the cutting edges into small elements. Since, using this model, forces can be calculated separately on each insert, it is possible to consider insert differences in estimation of the cutting loads. Torsional-axial coupling has been discussed and subsequently a time-domain model is proposed to simulate chatter vibrations. The resulting model is a system of delay differential equations with variable delays. The delay varies with time and is dependent on the state of the system. Variations in the time-delay, tool jump-out and backward motions of inserts have been included in the proposed time-domain simulation. A set of experiments was conducted to verify the model. Finally, a number of different strategies to alleviate the problem of chatter vibration are explored and their feasibilities for use in future products are discussed.