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Ultrasonic vibrations for single point diamond turning applications

"The development of a tunable cutting tool"

by

Jeroen A.A. van Assen

Report Number: PE 2002-081

A project of the University of North Carolina at Charlotte, Center for Precision Metrology

In cooperation with
The Technical University of Eindhoven, department of Mechanical Engineering, division Precision Engineering

August 2001-March 2002
Preface

This report has been written as a partial requirement for the degree of Master of Science. To understand the way this report is set up, it is important to know the history of it.

At the beginning of August 2001, the author started a research at the Technical University of Eindhoven (TU/e), department Mechanical Engineering, The Netherlands. This literature research took a look at the history of "ultrasonic vibration cutting". Its main purpose was to prepare the author for a research, larger in scale at the University of North Carolina at Charlotte (UNCC), Center for Precision Metrology, United States of America. The author started in the beginning of October 2001 with his research at the UNCC. He ended his research at the end of March 2002.

This report is divided into two parts. The first part is mainly written in the Netherlands and covers the results of the literature research. The second part publishes the results of the research done at the UNCC.

To get a total picture of "ultrasonic vibration cutting", the two Parts should be read as a whole. Nevertheless, they can be read separately.

Jeroen A.A. van Assen
Abstract

Ultrasonic vibration cutting is a relatively new technique used to cut hard or brittle materials like ceramic, glass and different sorts of metals on a Single Point Diamond Turning (SPDT) machine. In normal Single Point Diamond Turning it is only possible to cut soft materials like brass and aluminum. Cutting materials like steel results in excessive tool wear. Researchers all over the world have done extensive research in the field of ultrasonic vibration cutting and some even succeeded in cutting hardened steel without excessive tool wear. Though the technique seems to be promising nobody understands its fundamentals exactly.

The main reason for this is the lack of data of this process. Generally researchers used tools vibrating at a constant resonant frequency to perform the cutting. The experiments were not able to investigate the relationship between cutting frequencies and stroke on the one hand and tool wear and cutting forces on the other hand.

To do fundamental research in the field of ultrasonic vibration cutting, a tool is necessary which has a variable cutting frequency, stroke and vibration mode. If these parameters can be changed any time during an experiment important relationships can be derived which might reveal the well-kept secrets of ultrasonic vibration cutting.

A new technique will be presented to build a tool, which will be able to fulfill these requirements. The technique basically consists of a flexure system with an extremely high natural frequency and an in comparison low stiffness, driven by a set of piezoelectric elements which are actuated by a pulse generator. If the response of the PZTs would be optimal, an impulse in the system would result in a perfect impulse output. In that way several PZTs can be combined in series to virtual multiply the drive frequency to obtain a higher output frequency. This theory in combination with an adjusted pulse width input and the new flexure system leads to a new technique, which can be used in the future to actually do cutting experiments.
Acknowledgment

First of all, I would like to thank the University of North Carolina at Charlotte, Center for Precision Metrology, and especially Dr. R.J. Hocken for the invitation to come and work at the research center. The hospitality during my stay here was enormous.

I would like to thank my advisor in the U.S., Dr. James. F. Cuttino for his continued support and guidance in the execution and completion of this project. His insights in this research were extremely valuable to me. I would also like to thank my advisor in the Netherlands, Prof. Dr. Ir. P.H.J. Schellekens, for the opportunity to do this research in the United States.

I would like to thank the Technical University of Eindhoven, The Schuurman-Schimmel van Outeren foundation, The Dr. Hendrik Muller’s vaderlandsch foundation and my parents for their financial support. Without all of them this would never have happened.

I would like to acknowledge John F. Brien for his assistance in the design of the electronics, used in this research.

Above all, a final thanks to everybody who supported me during my stay in the United States. Like my parents and girlfriend for their support while being away from home for half a year, and of course Jerald Overcash and Stefan Rakuff, and many other students for the necessary entertainment.

Thanks, everybody! I have had a great time.

Jeroen van Assen
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Part I: Ultra precision single point diamond turning

A literature research conducted at the Technical University of Eindhoven, Faculty of Mechanical engineering, Precision Engineering, The Netherlands.

Period: August-September 2001
Advisor: Prof. Dr. Ir. P.H.J. Schellekens
Formulation of the problem

There has been an increasing demand, from industries all over the world, for high precision products made from (hardened) steel. Grinding these products takes a lot of time and effort, therefore some researchers tried to cut (hardened) steel by applying single point diamond turning. It seemed not possible to cut (hardened) steel with diamond tools because of excessive wear rates of the diamond tool tip. This lead to the development of a new cutting technique. Applying ultrasonic vibrations on the tool head, researchers stated that it seemed possible to cut hardened steel using the new technique. Wear rates of the diamond tool tip reduced significantly. Nevertheless, the technique is still in an experimental phase.

Aim of the research

Obtaining relevant information and knowledge about ultrasonic vibration cutting by an extensive literature research and by experiences with the new technique in industries. Once the necessary information is obtained, look at several aspects of ultrasonic vibration cutting in more detail.
1 Introduction

1.1 General introduction
Over the last 200 years, progress towards greater precision has led to tolerances less than 1 micron and surface roughness less than $R_{\text{max}} 0.1 \mu m$. Therefore the progress of machining accuracy, which has improved by at least one order of magnitude in the past few decades, has been considerable. The main pressure for this development is the demand for greater miniaturization, for better systems performance and for better quality control. To satisfy these demands a whole new range of techniques had to be developed, because it is not possible to extend the conventional machining processes as grinding [1]*.

1.2 History
The new technology, which is referred to, is Single Point Diamond Turning (SPDT). Parallel developments in Europe and the USA starting in the 1940s have led to widespread use of SPDT. For much of this period, quality of the parts produced was limited largely by machine performance, which was the focus of most development efforts. More recently there has been increased interest in understanding the details of tool-workpiece interactions. [2]
The ultra-precision machining community had in the early days generally accepted the premise that only certain materials are "diamond turnable". In practice diamond turnable materials are those where the tool wear rate is low enough that reasonable areas of surface can economically be produced. It is well known that non-ferrous metals as aluminum and copper are turnable. Some decent research has been conducted in this field. The ability of turning non-ferrous metals has nowadays been widespread. It has been applied to machine various light metals such as aluminum for computer disks and copper for mirrors of laser equipment.
The potential need for ultra-precision machining of steels was quite high. However the SPDT technique could not be applied to steels because of excessive wear of the diamond cutting tool. Cutting distances were in the order of tenths of a meter. In order to solve this problem, Casstevens [3] tried turning of high carbon steel in a carbon-saturated atmosphere. Masuda et al. and Moriwaki et al. employed single crystal and sintered CBN tools (Cubic Borium Nitride) in turning steel [4+5]. In order to reduce the frictional force between the tool rake and the chip efforts have been made to develop new lubricant technologies, rotary tools, new tool geometry, free cutting materials, cryogenic atmospheres [2] and so on.
All these attempts were not necessarily successful to satisfy both the practically long tool life and the optical quality surfaces. [6]
Besides the cutting of steel ductile cutting of brittle materials such as ceramics and glasses was and probably still is recognized as a necessary technology with important applications. Just like steel, researchers recognized that is not possible to cut brittle materials with an ordinary SPDT technique. In cutting brittle materials it is necessary to keep the depth of cut and the feed rate below a certain critical value, which govern the transition from plastic flow along the tool edge to brittle fracture. It is, however, difficult to keep the depth of cut and the feed rate at extremely low level over a wide range of cutting region even with ultra-precision machine tools. [7]

*[Nr.] Refers to the source used. The number refers to the number in the literature list.
Because of the need to cut steel and brittle materials a new technique was investigated by Kumabe et al. [8]. As regards the wear of the diamond cutting tool, he reported that the steel could be machined with a diamond tool by applying an ultrasonic vibration to the tool. Although the accuracy of machining did not reach the level of ultra-precision, it seemed to be promising to solve the problem of the wear of the diamond tool. [6]

After this discovery several research institutes used the technique to cut steel and glass. First there were experiments conducted by Weber [9] and Moriwaki et al. [6] applying ultrasonic vibration on the cutting of stainless steels and glass.

The experiment of Weber showed:
1. It is possible to increase tool life up to 20 times while cutting glass by superimposing the cutting operation with ultrasonic vibrations.
2. Ultrasonic super position is able to prevent the formation of built-up edges while cutting steel.

The experiments of Moriwaki showed:
1. The feasibility of ultra-precision diamond machining of the steel family.
2. An optical quality mirror of stainless steel with surface roughness of 0.026 μm can be obtained.
3. High quality surfaces are obtained stable, up to a cutting distance of 1600 m.

This kind of research was a breakthrough in the ability of machining the steel and glass family [7]. Because little was understood of the actual mechanism of tool wear during ultrasonic operation, research was conducted. Li investigated the effect of tool vibration on tool wear in ultrasonic vibration cutting [10], he concluded that ultrasonic vibration decreases tool crater wear but increases tool flank wear, this will be explained in detail later. Kim looked at the characteristics of chip generation during extremely low cutting velocity [11].

In the last decade an improvement of the traditional ultrasonic vibration technique emerged. Ultra-precision machining of hardened steel is required for precision molds and dies for various plastic and steel products, etc. They are normally ground and polished subsequently, however the diamond turning has many advantages of high geometrical accuracy, good surface quality, high machining efficiency and capability to machine complicated shapes. In order to realize the diamond cutting of hardened steel, an elliptical vibrating tool was developed. The development of this tool and the experimental phase were conducted by Moriwaki et al.[12+13+14]. After five years (start in 1994) they were able to cut hardened steel with a flexible microstructure.

Still there must be admitted that actually little is understood of the real material removal process. Although, as mentioned, several researchers in the past have done extensive research in this field there is still a lot to be done in the future.
2  Understanding the Single Point Diamond Turning Process of ferrous materials

In order to obtain the demanded high precision surfaces one has to fulfill a certain amount of criteria. The geometrical accuracy of the products is determined by:

1. The machine layout (Chapter 3)
   - Static and dynamical behavior
   - Numerical control
   - Thermal behavior etc.
2. The cutting process (Chapter 2)
   - Wear and geometry of the tool
   - Vibrations etc.

In this Chapter the attention is focused at the cutting process. Because the cutting process of ferrous materials is a rather complicated subject, not every detail is discussed.

2.1 Different types of tool wear

Tool wear performance is an important factor in cutting difficult-to-machine materials and precision machining. There are usually two types of tool wear in metal cutting (Fig 2.1). One is crater wear, which occurs in the form of a small depression at the tool rake face. Another is flank wear, which appears as a land immediately below the cutting edge. In general, the crater wear affects the chip formation process, the flank wear affects the machined workpiece quality and a combination of both affects the strength of the tool cutting edge.

2.2 Cutting forces

This part is partially obtained from: fundamentals of machining and machine tools by Boothroyd [15]

All metal-cutting operations can be likened to the process shown in fig 2.2. This figure represents a special case of cutting. This two-dimensional problem is known as orthogonal cutting, hereby is the cutting edge of the tool arranged to be perpendicular to the direction of relative work-tool motion (feed direction). This model lends itself to research applications where it is desirable to eliminate as many of the independent variables as possible.
However single point diamond cutting is a three-dimensional non-orthogonal process. This means that widely used relatively simple models of orthogonal cutting cannot be used without modifications.

The cutting tool basically consists of two surfaces intersecting to form the cutting edge. The surface along which the chip flows is known as the rake face, or simply as the face, and the surface ground back to clear the new or machined workpiece surface is known as the flank or clearance face. The depth of the individual layer of material removed by the action of the tool is known as the undeformed chip thickness. One of the most important variables in metal cutting is the slope of the tool face, and this slope, or angle, is specified in orthogonal cutting by the angle between the tool face and a line perpendicular to the new work surface as shown in fig 2.3. This angle is known as the rake angle.

The tool flank plays no part in the process of chip removal, however, the clearance angle can affect the rate at which the cutting tool wears. [15]

In orthogonal cutting the resultant force $F_r$ applied to the chip by the tool lies in a plane normal to the tool cutting edge (fig 2.4). This force is normally determined from the measurement of two orthogonal components: one in the direction of cutting ($F_c$) and one in the direction of thrust ($F_t$).
If the resultant tool force $F_r$ is resolved in a direction parallel to the shear plane, the force $F_s$ required to shear the work material and form the chip is obtained. This force is very important in predicting the machinability of a specific material. Note: if the shear angle increases, the force required to form a chip will be reduced. If the resultant force $F_r$ is resolved in a direction parallel to the tool face, the friction force $F_f$ is obtained.

The resultant force in metal cutting is distributed over the areas of the tool that contact the chip and workpiece. As the tool edge "plows" its way through the work material, the force that acts on the worn tool cutting edge forms only a small proportion of the cutting force at large values of the undeformed chip thickness. At small values of the undeformed chip thickness, however, the force that acts on the tool edge is large and cannot be neglected. Because of the high stresses acting near the tool cutting edge, deformation of the tool material may occur in this region. This deformation would cause contact between the tool and the new workpiece surface over a small area of the tool flank. Thus when sharp cutting tools are used, a frictional force may act in the tool-flank region. Neither the force acting on the tool edge or the force that may act on the tool flank contributes to removal of the chip, and these forces will referred to as the plowing force $F_p$. The existence of the plowing force results in certain important effects and can explain the so-called size effect. This term refers to the increase in specific cutting energy at low values of undeformed chip thickness. It is thought that the plowing force $F_p$ is constant and therefore becomes a greater proportion of the total cutting force as the chip thickness decreases. So in diamond turning with very low undeformed chip thickness the specific cutting energy will be very high.

### 2.3 Tool wear mechanisms

This part is partially obtained from: Cryogenic diamond turning of Stainless steel by Evans [2] and Chemical aspects of tool wear in single point diamond turning by Paul and Evans [20]

Wear of single crystal diamond turning tools may occur by a number of mechanisms, which may take place simultaneously or interactively.
1. Adhesion and the formation of a built-up edge; Under some conditions the friction between the chip and the tool is so great that the chip material welds itself to the tool face. The resulting pile of material is referred to as a built-up edge. An increasing built-up edge increases tensile forces on the diamond edge, which eventually fractures, and the process starts again.

2. Abrasion; It is common experience in diamond turning soft metals that hard particles causes chipping on the tool edge and a degradation of surface finish. These hard particles may be highly strain-hardened fragments of an unstable built-up edge or hard constituents in the material itself.

3. Tribothermal; Increased temperatures at the tool-work interface may cause thermal degradation of the diamond. Diamond is an unstable form of carbon and reverts to graphite at elevated temperatures (above approximately 1000 K).

4. Tribochemical; Three types of chemical wear are possible: oxidation, diffusion and catalyzed graphitization. First of all diamond oxidizes rapidly at high temperatures (above 900 K). It is also possible that carbon diffuses from the diamond tool into the ferrous work-piece. Solid state diffusion occurs when atoms move from a region of high atomic concentration to one of low concentration. This process is dependent on the existing temperature.

A number of the above mentioned mechanisms might be involved in tool wear in the single point diamond turning of steels. More than one mechanism may contribute, and the balance of mechanisms is clearly a strong function of specific operating conditions.

Some extensive research has been done to find the dominant mechanism. Problems in comparing the different researches are the operating conditions. Each research handles its own cutting conditions like feed and undeformed chip thickness.

Possible dominant wear mechanisms:

*ad 1.* One theory reports the formation of a built up edge when diamond turning low carbon steel. A built-up edge was formed on the diamond tool and the resulting surface finish was poor but the cutting edge of the diamond remained sharp and even. In this experiment material removal rates were high compared to modern diamond turning practice.

Another research observed that above a critical sliding speed metal is smeared over the surface of the diamond. The diamond is untouched, the rubbing interface is metal on metal and adhesion prevents diamond wear. Below the critical speed there is no apparent metallic transfer and the diamond wears rapidly.

One explanation for the high wear rates of diamond tools in machining steel that has been proposed is based on an unstable built up edge. If one of these mechanisms were dominant, evidence of the built-up edge should be seen on the tool, no such evidence has been found by Evans, Wilks or Casstevens [2]+[3].

*ad 2.* Watson points out that inclusions in steels may have Vickers hardness up to 2 orders of magnitude greater than the iron matrix. Inclusion hardness, however, is still less than diamond hardness. Thus the mechanism for inclusion removal is probably microcleavage. The wear of the diamond during microcleavage is a fracture along the preferred cleavage plane rather than abrasion.

Contrary to expectation diamond shows fatigue behavior. The fatigue mechanism may contribute to tool wear when machining steels of relatively
low hardness. In this case, the carbides will introduce relatively high frequency cyclic stresses.

ad 3. In order to estimate the contribution of tribothermal wear to the wear of the diamond tool, one has to measure the temperature of the rake face. There seems to be a tendency to accept the fact that tool temperatures are well below the critical temperature of graphitization [2]. This is also reported by Lucca and Seo. They did some experiments on electroless nickel and steel. If thermal graphitization would be the primary wear mechanism, wear rates should be similar. But they were not. In fact the wear rates were some orders of magnitude different. Some other researchers [16] mentioned that the pressure upon a certain face of the diamond tool could interact with the graphitization of the diamond. At pressures of 3500 Mpa (as can be estimated on the rake face) diamond is the stable configuration of carbon up to 1000 K. On the other hand, the mean pressure on the flank wear land is significantly lower. At typical flank face pressures, diamond is meta-stable at any temperature. They concluded that graphitization is therefore likely to occur at the flank face of the tool. Note: this proceeding was talking about *thermal* graphitization. So this means graphitization initiated by temperature, not by tool-workpiece contact!

ad 4. In explaining observations of chemical wear several different approaches have been used. Empirical correlations relate melting points and crystal structures to materials that excessively wear diamond tools. There is also some correlation with hardness. However careful examination of the data shows unexplained anomalies for each of these approaches. Paul proposed that unpaired d electrons in the workpiece allow carbon-carbon bond breaking in diamonds followed by metal-carbon complex formation, leading to chemical wear. Chemical reaction rates can be understood in terms of mechanisms that lead from reactants through intermediate transition complexes to product formation. The rate of reaction is related to the ease of formation of the intermediate complex and is affected by the reacting species, temperature, pathways and environmental conditions. In this way carbon atoms can revert to the diamond lattice, form graphite structures, react with other materials in the environment or diffuse into the metal if there are vacancies available.

Until this moment it is very difficult to say which mechanism is the dominant one. Though the wear mechanisms are not understood completely, researchers have found some variables by experiment that seem to be of high importance. Probably the most important one is the temperature. Temperature increases the rate of chemical reactions exponentially. The temperature at the diamond tool head is directly deviated from the cutting forces. Therefore some extensive research has been done to find the optimal cutting conditions for various materials. Factors like cut depths, feed and rake angle were investigated.

Note that any wear of the diamond tool will increase tool forces and hence temperature; this in turn will increase the rate of tribochemical and tribothermal wear mechanisms.
3 Necessary machining conditions for ultra-precision turning

Not only understanding the cutting process, as discussed in Chapter 2, but controlling the total package of machining conditions is an important factor. In order to achieve earlier mentioned accuracy’s, the static, dynamic and thermal behavior of the machine’s components should be analyzed.

3.1 Machine statics and dynamics

The force-loop between the workpiece and the tool has to be one of very high stiffness. Because of rake –and flank forces, the machine will deviate from its wanted path. Besides its static behavior, the machine dynamics are extremely important. The machine, cutting tool and workpiece form a structural system with a complicated dynamic characteristic. Under certain conditions vibrations of the structural system may occur, and as with all types of machinery, these vibrations may be divided into three basic types [15]:

1. Free or transient vibrations: resulting from impulses transferred to the structure through its foundation as from rapid reversals of masses, such as machine tables. The structure is deflected and oscillates in its natural modes of vibration.
2. Forced vibrations: resulting from periodic forces within the system, such as unbalanced rotating spindles, or transmitted through the foundations from nearby machinery.
3. Self-excited vibrations: usually resulting from a dynamic instability of the cutting process. This phenomenon is commonly referred to as machine tool chatter.

It is important to limit vibrations of the machine tool structure as their presence results in poor surface finish, cutting-edge damage and irritating noise. These deviations to vibrations may be reduced by [1]:

1. Eliminating or reducing the exciting forces
2. Avoiding the coincidence of frequencies of exciting forces and natural frequencies of the machine tools.
3. Increasing the stiffness and the damping of the machine elements.

In order to do this; it is necessary to know the sources, amplitudes and frequencies of the vibrations as well as the properties of the machine elements.

These demands have, in the past, led to the development of high accuracy lathes. The precision lathe usually comprises a spindle with hydrostatic or aerostatic bearings, hydrostatic or aerostatic linear slides, an air and/or oil supply system, a motor for driving the spindle and a position control system.

Any remaining errors should be minimized by introducing better metrology and control systems.
4 Applying ultrasonic vibration

In Chapter 2 we concluded that wear of the diamond tool is the limiting factor in turning steel. In the past, several methods for reducing tool wear in diamond turning of steel have been proposed, i.e. cryogenic turning by cooling the tool-workpiece system with liquid nitrogen and turning in inert and other gas atmospheres. These experiments were valuable for understanding the wear process, but the techniques are not suitable for precision turning of steel.

As regards to the wear of the diamond cutting tool, Kumabe [8] reported that steel can be machined with a diamond tool by applying an ultrasonic vibration.

4.1 "Linear" ultrasonic vibration cutting

In the first report about ultrasonic vibration cutting Kumabe noted that there are three possible directions in which the vibration can take place: a) the cutting direction; b) the thrust direction; and c) the feed direction. Kumabe reported that the application of ultrasonic vibration only in the cutting direction was practical (Fig. 4.1).

Fig 4.1: orthogonal cutting with ultrasonic vibrated tool in the cutting direction

H. Weber et al. [9] conducted a research, where they tried to cut glass and metal with an ultrasonically vibrated tool. The tests in turning were performed by a cutting tool system who’s cutting edge carried out a linear ultrasonic vibration in the cutting direction within the 20 kHz range.

The turning operation with ultrasonic vibration results in a periodic interruption of the cut. Weber mentioned that the wear of the tools is much more influenced by ultrasonic vibration than the cutting force. Note that Weber used tungsten carbide as cutting tool. At the end of his research Weber concluded that the tool life increased up to 20 times by superimposing ultrasonic vibrations when compared with the normal turning of glass. The basic reason was according to Weber to be seen in stimulating the fracturing process typical of the chip formation in glass. Weber also observed that cutting steel at low surface speed in compliance with an ultrasonically vibrated tool reduced surface roughness.

Moriwaki used a high oscillation frequency of 40 kHz, in way of reducing vibration marks on the finished surface. And indeed after some experiment he concluded that the roughness in the cut direction was lower that in the feed direction (10 μm/rev). Moriwaki was the first researcher who showed that ultra-precision diamond machining of steel was possible by applying ultrasonic vibration.

Several studies were conducted and overall they concluded the same. Compared to conventional cutting, the friction coefficients at the cool-chip interface and tool flank-work interface are lower, the chip formation process is facilitated, and consequently the cutting forces and temperature are lower, the machined surface finish is better and the residual stresses at the machined workpiece surface are less [10]. As research proceeded little was understood about the effect of ultrasonic vibration on tool wear. However, since tool wear performance is an important factor in cutting of difficult-to-machine materials, the problem is to be considered. Li did some research on this problem.

4.1.1 Analyses on tool wear

This part is partially obtained from: The effect of tool vibration on tool wear in ultrasonic vibration cutting by Xiaoping Li [10].

The mechanics of the tool cutting edge vibrated in the cutting direction for turning is considered as follows:

In ultrasonic vibration cutting, due to the tool vibration, the actual cutting velocity of the vibrated tool cutting edge relative to the workpiece is:

\[ V = V_0 + A \cdot \omega \cdot \cos(\omega \cdot t) \]  [4.1]

where \( V_0 \) is the cutting speed and \( A \) the amplitude of the vibration. Compared to the cutting speed \( V_0 \), \( V \) can be much higher at one moment and much lower in the next, depending on the vibration frequency and amplitude.

When:

\[ \frac{A \cdot \omega}{V_0} > 1 \]  [4.2]

the tool cutting edge will be separated frequently from the chip, which would not happen in conventional cutting. Due to this separation, there is a discrepancy between the net cutting time and the time for cutting. If separation occurs, the net cutting time is less than the time for cutting.

In conventional cutting the distance of the tool sliding over the machined workpiece surface is:

\[ L_0 = \pi \cdot D \cdot n \cdot t \]  [4.3]

where \( D \) is the diameter of the workpiece, \( n \) is the number of revolutions per time unit and \( t \) is the cutting time used. In ultrasonic vibration, an extra sliding distance \( \Delta L \) is introduced. Therefore the actual sliding distance \( L \) is the sum of \( L_0 \) and \( \Delta L \).

\[ L = L_0 + \Delta L \]  [4.4]

\[ \Delta L = 2 \cdot A \cdot f \cdot t \]  [4.5]

\[ f = \frac{\omega}{2 \cdot \pi} \]  [4.6]
Li investigated tool crater and tool flank wear. In ultrasonic vibration cutting the tool rake face is separated each cycle from the chip (if equation 4.2 is fulfilled). Therefore, the crater wear should be less compared to conventional cutting. Another effect of the separations on the crater wear process is that during each of the separations cutting fluids or surrounding air can access the tool-chip contact region, which never occurs in conventional cutting at normal conditions. Also, since the time interval for each of the tool-chip separations increases with increasing tool vibration amplitude or frequency, it can be expected that the tool crater wear rate decreases with increasing the vibration amplitude or frequency.

In ultrasonic vibration cutting, the wear process of the tool flank might be intensified by the vibration. The distance of the tool sliding over the machined workpiece surface can be much longer compared to conventional cutting. This might increase abrasive wear on the tool flank.

The foregoing review was proved by experiments by Li. Thus ultrasonic vibration cutting reduces tool crater wear but increases tool flank wear.

Another problem when cutting hardened steel arises at the tool edge and is called chipping. Jin and Murakawa [17] proposed that chipping and unusual wear of the cutting edge could be reduced successfully by means of “inclined” tools. Jin considered that the main cause of chipping on the tool edge during ultrasonic vibration cutting is when the vibrating tool leaves the cutting point of the workpiece, the flank of the tool collides or rubs against the surface of the workpiece. Thus, it is considered that the flank of the tool collides with the workpiece because of the elastic recovery (strain due to thrust force on tool) of the workpiece and/or the tool.

To prevent contact between the flank face and the workpiece, an attempt was made to make the vibrating direction incline toward the depth-of-cut direction from the cutting direction by tens of degrees (angle \( \theta \), see fig. 4.2)

Jin named this way of cutting an “inclined” ultrasonic vibration cutting method and concluded that stable finish cutting can be carried out without tool chipping, if hardened steel is cut using an inclined vibration direction at a level of about 30°.
4.1.2 Development of an ultrasonic vibration cutting tool system

Most conventional ultrasonic vibration cutting tools use a longitudinal vibration mode or a bending vibration mode. Both the overhang lengths in longitudinal and bending vibration mode are much longer than in conventional turning because there must be a certain distance from both the fixed point and the node of vibration to the tool tip. An advantage of the bending vibration mode: the overhang length at same rigidity in a longitudinal vibration mode is longer than one in bending mode.

The longitudinal vibration tool Moriwaki used was of a minor quality (Fig 4.3). Besides the vibration in the cutting direction there was a lateral vibration superimposed due to the unbalanced weight of the tool tip at the end of the horn. The tool is also not rigid enough to prevent chipping as mentioned above. A bolted Langevin type transducer is employed to generate the ultrasonic vibration in the axial direction. The amplitude of vibration is magnified by three times as the vibration is transmitted through the stepped horn. Jin and Murakawa developed an ultrasonic vibration cutting tool system with a greater stiffness in both the thrust and feed direction. Because of the high stiffness, the researchers hoped that an actual one way linear movement of the tool would be created as well as no deviation of the tool due to thrust forces, preventing chipping.

A cutting tool system was fabricated as shown in Appendix A. For improving the rigidity of the tool, the longitudinal vibration mode is applied. The tool is strongly attached to a tool post at four nodal points of the tool shank. The resonance frequency of this tool is 21 kHz and the amplitude is 15 μm.

The researchers concluded that this tool has not yet reached the level of practical application. The chipping was, though less, still observed. In terms of the cause of this, it is considered that the improvement of the rigidity of the tool is not complete as a countermeasure for preventing contact between the flank face of the tool and the cut surface of the workpiece. Inclining the tool didn't appear to be the right solution. In the following paragraph a new method is proposed to prevent chipping at the tool edge.
4.2 Elliptical ultrasonic vibration cutting

The ultrasonic vibration cutting, as mentioned before, has been successfully applied to ultra-precision diamond cutting of difficult-to-cut materials, including steel and glass materials. Generally the tool is vibrated in the cutting direction, so that the tool is separated from the chip in each cycle of the vibration. When cutting hardened steel chipping of the cutting edge determined the tool wear generally.

Shamoto and Moriwaki [12+13+14] proposed to vibrate the cutting tool in a plane including the cutting direction and the thrust direction in an elliptical vibration mode (see fig 4.4). In this way the tool also has a velocity component in the chip flow direction in each cutting cycle after it penetrates into the workpiece. The cutting takes place after re-entering of the cutting edge into the workpiece, and the chip is mainly pulled up and formed while the tool moves up in the chip flow direction. The tool moves down without cutting while it is separated from the chip. As a consequence, the frictional force between the tool rake and the chip is effectively reduced by reversing the frictional direction, and the reversed frictional force assists the chip to flow out.

There should be noted that the instantaneous cutting direction is not constant in this way of cutting. Also the clearance angle is changing all the time. Because this clearance angle is changing during cutting there is a critical value to be found. The nominal clearance angle needs to be greater than this critical value.

Shamoto and Moriwaki did the first experiments cutting copper with a high speed tool at a rather low frequency [13]. They concluded: a) the chip thickness and the cutting forces are reduced significantly by applying the elliptical vibration as compared with the conventional vibration cutting. b) The shear angle is increased, as the frequency of vibration is increased. Thus the cutting method is quite effective to control the nominal friction between the rake face and the chip. c) The nominal clearance angle needs to be larger that the ordinary cutting so that the instantaneous clearance angle is always positive.

In a second article about the cutting of copper using elliptical vibration cutting [14], Shamoto and Moriwaki concluded that applying elliptical vibration cutting suppresses the formation of burrs at the edges of chips. Because cutting forces are further reduced in comparison to conventional vibration cutting, geometrical accuracy is better in the elliptical vibration cutting. Note: at the cutting frequency of 20 kHz used in this experiment, the surface roughness (0.02 μm R_max) is larger than that obtained by ordinary cutting!
In 1999 Shamote and Moriwaki [12] applied elliptical vibration cutting to single diamond turning of hardened steel. The conventional linear vibration cutting of hardened steel requires careful adjustment of the vibration direction inclined to the nominal cutting direction, so that the clearance face does not interfere with the cut surface. Although the conventional vibration cutting can also realize ultra-precision turning of hardened steel, the same or better surface can be obtained by the elliptical vibration cutting without such careful adjustment. The researchers concluded that elliptical vibration cutting has superior performance: low cutting force, high quality finish and long tool life (See Appendix B).

Chipping in elliptical mode occurred at 2300 m, while in conventional vibration mode chipping was observed after 300 m., so the main tool failure mechanism has been postponed drastically.

4.2.1 Design of an ultrasonic elliptical vibrated tool system

To use in their experiments, Shamote and Moriwaki developed several elliptical vibrating tools. In the first two experiments 2 different tools were developed. The first one was only capable of handling frequencies up to 6 Hz. The second one used resonant bending modes to make an elliptical pattern on the end of the tool. The horn was resonating at 20 kHz, actuated by two piezoelectric plates glued to the lateral surfaces of the vibrator as shown in fig. 4.5.

When two piezo’s at resonance frequency perform an orthogonal vibration, an ellipse or a circle is obtained.

This can be explained mathematically:

\[ x(t) = a \cos(\omega \cdot t) \]
\[ y(t) = b \cos(\omega \cdot t + \theta) \]
\[ r(t) = x(t) + y(t) \]

If the phase difference \( \theta \) between the piezo’s is equal to \( \frac{1}{2} \) phi, a circle is the result (see appendix E). When the phase difference decreases to zero, the circle will transform into an ellipse, which becomes smaller at decreasing phase difference. If the phase difference is equal to zero a straight line is the result. The importance of this conclusion is the need of direct control of the phase difference between the two piezo’s.

The vibration amplitude is magnified by the step horns. The horn is supported rigidly at the two nodal points not to disturb the vibration. Note that the system has suppressed to many degrees of freedom.
4.3 Appliance of ultrasonic vibration cutting in industries

The information used in writing the preceding pages was found in literature. In order to get an idea of the appliance of ultrasonic vibration cutting in industries, two visits have been paid to leading research laboratories in the Netherlands and Germany. In the Netherlands, Philips N.V. research laboratories and in Germany, the IPT fraunhofer institute (department production-technology) have been visited. At both the institutes research has been done in the field of ultrasonic vibration cutting. At Philips, traditional (linear) ultrasonic vibration cutting is used for machining products made from hard to cut materials (steel). The technique has proved itself and is nowadays used in various production-plants of Philips to cut high precision products. At the IPT fraunhofer institute no direct information was obtained because the dissertation of the research was not available yet and the researcher had left the institute.


## 5 Vibration of beams

In order to create ultrasonic vibrations on the tool edge, all researches until now have used wave propagation in beams. To understand these wave propagations, some basic knowledge has to be present. In this chapter therefore, the flexibility of beams is considered and two types of vibrations will be discussed: longitudinal and transverse vibrations. The mass of the beam is assumed to be distributed evenly in the model. These systems are called infinite-dimensional, continuous or distributed parameter systems.

### 5.1 Bending vibrations

A straight elastic beam possesses both the mass and stiffness to resist bending. During transverse vibration, the beam flexes perpendicular to its own axis to alternately store potential energy in the elastic bending of the beam and then release it into kinetic energy of transverse motion.

Figure 5.1 illustrates a beam with the transverse direction of vibration indicated (deflection, \( w(x,t) \) in the \( y \)-direction). The beam is of rectangular cross section \( A \) with width \( h_y \), thickness \( h_z \) and length \( L \). Also associated with the beam is a stiffness \( EI \), where \( E \) is Young's elastic modulus for the beam material and \( I \) is the cross-sectional area moment of inertia about the \( z \)-axis.

![fig. 5.1: bending vibrations of a beam](image)

A model for bending vibration may be derived from examining the force diagram of an infinitesimal element of the beam as indicated in fig. 5.1. From mechanics one obtains:

\[
M(x,t) = EI \frac{\partial^2 w(x,t)}{\partial x^2} \tag{5.1}
\]
Ultrasonic vibrations for single point diamond turning applications

Assuming the deformation to be small enough such that the shear deformation is much smaller than \( w(x,t) \) (which is generally true when \( L/h_x > 10 \) and \( L/h_y > 10 \), so for long slender beams), a summation of forces in the \( y \)-direction yields:

\[
\left( V(x,t) + \frac{\partial V(x,t)}{\partial x} \right) - V(x,t) + f(x,t) \, dx = \rho A dx \frac{\partial^2 w(x,t)}{\partial t^2}
\]

Here \( V(x,t) \) is the shear force and \( f(x,t) \) is the total external force applied to the element per unit length. The term on the right side of the equality is the inertial force of the element. Next the moments acting on the element are summed. This yields:

\[
\left( M(x,t) + \frac{\partial M(x,t)}{\partial x} \right) - M(x,i) + \left( V(x,t) + \frac{\partial V(x,t)}{\partial x} \right) dx + (f(x,t) \, dx) \frac{dx}{2} = 0
\]

Since \( dx \) is very small, \( (dx)^2 \) is assumed to be almost zero, so that this moment expression yields:

\[
V(x,t) = -\frac{\partial M(x,t)}{\partial x}
\]

Substitution of equations [5.1] and [5.4] into [5.2] yields:

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

If no external force is applied so that \( f(x,t) = 0 \) and if \( EI \) and \( A \) are assumed to be constant, equation [5.5] simplifies so that free vibration is governed by:

\[
\frac{\partial^2 w(x,t)}{\partial t^2} + c^2 \frac{\partial^4 w(x,t)}{\partial x^4} = 0
\]

\[
c^2 = \frac{EI}{\rho A}
\]

A rod or thin beam can oscillate transversely in an infinite number of ways, satisfying the fundamental equation [5.6]. There are, however, certain types or modes of oscillation in each of which every section of the beam executes simple harmonic motion in the same period and same phase, but of different amplitudes. These modes are called normal modes and are expressed by:

\[
w(x,t) = u(x) \sin(\omega t + \alpha)
\]

Substituting equation [5.8] into [5.6] yields:

\[
\frac{\partial^4 u}{\partial x^4} - \frac{\rho A \omega^2 u}{EI} = \left[ D^4 - \lambda^4 \right] u = 0
\]

\[
\lambda^4 = \frac{\rho A \omega^2}{EI}
\]

This latter differential equation does not involve \( t \) and, therefore, the assumption of equation [5.8] is verified.

The result is a fourth order differential equation with constant coefficients, factorizing equation [5.9] yields:
Ultrasonic vibrations for single point diamond turning applications

\[
\begin{align*}
[D^2 + \lambda^2 D]u & = 0 [5.11] \\
[D - j\lambda][D + \lambda][D - \lambda][D + \lambda]u & = 0 [5.12]
\end{align*}
\]

Hence:

\[u = ae^{j\lambda x} + be^{-j\lambda x} + ce^{\lambda x} + de^{-\lambda x} [5.13]\]

\[u = A \cos \lambda x + B \sin \lambda x + C \cosh \lambda x + D \sinh \lambda x [5.14]\]

The value for \(\lambda\) and three of the four constants of integration, A, B, C and D, can be determined from the four boundary conditions. This is already done by D. Blevins [23], he presented a technique to calculate the natural frequencies and bending modes very quickly.

The mode shapes and natural frequencies of a slender beam are a function of an integer index (n), which may be associated with the number of half-waves in the mode shape. For each \(n\) there is a natural frequency and mode shape. The natural frequency in Hertz can generally be expressed in the form:

\[f_n = \frac{\lambda_n^2}{2\pi^2} \sqrt{\frac{EI}{\rho A}}; \quad n = 1, 2, 3, \ldots [5.15]\]

Where \(\lambda_n\) (table 5.1) is a dimensionless parameter, which is a function of the boundary conditions applied to the beam. For a free-free beam, \(\lambda_n\) is the solution of the function:

\[\cos \lambda \cosh \lambda = 1 [5.16]\]

With this method, also the mode shape of a free-free beam can be calculated:

\[u(x) = \cosh\left(\frac{\lambda_n x}{L}\right) + \cos\left(\frac{\lambda_n x}{L}\right) - \sigma_n \left[\sinh\left(\frac{\lambda_n x}{L}\right) + \sin\left(\frac{\lambda_n x}{L}\right)\right] [5.17]\]

The dimensionless parameter \(\sigma_n\) (table 5.1) is also a function of the boundary conditions applied to the beam and can, for a free-free beam, be calculated with equation [5.18]:

\[\sigma_n = \frac{\cosh \lambda_n - \cos \lambda_n}{\sinh \lambda_n - \sin \lambda_n} [5.18]\]

<p>| Table 5.1: natural frequencies and mode shape parameters for a free-free beam |
|-----------------|---------|---------|</p>
<table>
<thead>
<tr>
<th>n</th>
<th>(\lambda_n)</th>
<th>(\sigma_n)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.730</td>
<td>0.9825</td>
</tr>
<tr>
<td>2</td>
<td>7.853</td>
<td>1.0007</td>
</tr>
<tr>
<td>3</td>
<td>10.996</td>
<td>0.9999</td>
</tr>
<tr>
<td>4</td>
<td>14.137</td>
<td>1.0000</td>
</tr>
<tr>
<td>5</td>
<td>17.279</td>
<td>0.9999</td>
</tr>
<tr>
<td>&gt;5</td>
<td>((2n+1)\pi/2)</td>
<td>(\approx 1)</td>
</tr>
</tbody>
</table>

Now one is able to calculate the natural frequencies and natural modes of a beam (See appendix C and D) as can be seen in figures 5.2 and table 5.2.
The slender beam model given above is often referred to as the Euler-Bernoulli beam equation. The assumptions used in formulating this model are that the beam is:

1. Uniform along its span, or length, and slender.
2. Composed of a linear, homogeneous, isotropic elastic material without axial loads.
3. Such that plane sections remain plane.
4. Such that the plane of symmetry of the beam is also the plane of vibration so that rotation and translation are not coupled.
5. Such that rotary inertia and shear deformation can be neglected.

The model presented above ignores the effects of shear deformation and rotary inertia. It is safe to ignore the shear deformation as long as the height and the width of the beam are small compared with the length of the beam. As the beam becomes shorter, the effect of shear deformation becomes evident.

Timoshenko proposed a new beam model, which takes shear deformation and rotary inertia into account:
THE EFFECT OF TOOL VIBRATION ON TOOL WEAR IN ULTRASONIC VIBRATION CUTTING


ABSTRACT: Theoretical and experimental studies on the effect of tool vibration on tool wear in ultrasonic vibration cutting are presented. Analyses are given on the mechanics of ultrasonic vibration cutting, from which the relationships between the vibration and machining parameters are described. The special mechanisms of tool wear in ultrasonic vibration cutting are discussed using the analyses. Experimental tests for tool wear in cutting with and without the vibration at various cutting conditions are conducted based on the analyses. Compared to conventional cutting, it is found that in ultrasonic vibration cutting the tool wear is much less, but the tool flank wear is larger. The effect of tool ultrasonic vibration on tool wear is more significant when a water-based coolant is applied to the cutting region.

KEYWORDS: metal cutting, ultrasonic applications, tool ultrasonic vibrations, tool wear, turning tests, coolant applications

INTRODUCTION

The unique feature of ultrasonic vibration cutting is that the tool cutting edge is oscillated ultrasonically during cutting. It has been found that ultrasonic vibration cutting has many advantages. Compared to conventional cutting, the friction coefficients at the tool chip interface and tool flank work interface are lower [1][2], the chip formation process is facilitated [3], and consequently the cutting forces and temperature are lower [1][4]. The machined workpiece surface finish is better [5] and the residual stresses at the machined workpiece surface are less [6]. Due to having these advantages, ultrasonic vibration cutting is particularly suitable for machining of difficult-to-machine materials [2][3][4][7][8], and for finishing processes [5] and super-precision machining [8][9]. Therefore, despite the additional cost is required for the ultrasonic vibration equipment, ultrasonic vibration cutting has been widely used in machining industry.

However, since tool wear performance is an important factor in cutting of difficult-to-machine materials and precision machining, the effect of the tool ultrasonic vibration on tool wear is a problem to be considered. There are usually two types of tool wear in metal cutting (see Fig. 1(b)): One is crater wear, which occurs in the form of a small depression at the tool rake face. Another is flank wear which appears as a land immediately below the cutting edge. In general, the crater wear affects the chip formation process, the flank wear affects the machined workpiece integrity, and a combination of both affect the strength of the tool cutting edge. Therefore, a knowledge of the effect of the vibration on tool crater and flank wear is necessary for the planning and tool selection as well as choosing optimum cutting conditions in ultrasonic vibration cutting. In this paper, theoretical and experimental studies on the effect of tool ultrasonic vibration on the tool wear performance will be presented.

THEORETICAL ANALYSES

A schematic illustration of turning with an ultrasonically vibrated tool is shown in Fig. 1. There are three possible directions in which the ultrasonic vibration is applied.
to the cutting edge during cutting, namely the cutting direction, the feed direction and the thrust direction. It has been reported [10] that only the application of ultrasonic vibration in the cutting direction is practical. The mechanics of the tool cutting edge vibrated in the cutting direction for turning is considered as follows.

\begin{align*}
\text{Ultrasonic Vibration at the Tool Cutting Edge} \\
\text{Consider the ultrasonic vibration of the tool cutting edge in a co-ordinate system as shown in Fig. 1, in which the origin is at the cutting edge of the tool without vibration. The vibration of the tool cutting edge can be described by its position} \\
y = A \sin \omega t \\
\text{and the velocity} \\
dy \over dt = A \cos \omega t \omega \\
\text{where } A \text{ is the vibration amplitude and } \omega \text{ is the angular frequency.}
\end{align*}

\begin{align*}
\text{Influence of the Vibration on the Cutting Speed} \\
\text{In ultrasonic vibration cutting, due to the tool vibration, the actual cutting velocity of the vibrated tool cutting edge relative to the workpiece is} \\
V = V_0 \pm A \cos \omega t \\
\text{where } V_0 \text{ is the velocity of the tool cutting edge without vibration, which is equal to the cutting speed used, Eq. 3 indicates that in ultrasonic vibration cutting the relative cutting velocity } V \text{ is a combination of the cutting speed } V_0 \text{ and the vibration velocity } A \cos \omega t. \text{ Since the vibration velocity varies in its direction, the relative cutting velocity also varies between its maximum and minimum during cutting. Compared to the cutting speed } V_0, V \text{ can be much higher at one moment and much lower in the next, depending on the vibration frequency and amplitude. When the product of the amplitude and frequency is so large that} \\
\frac{A \omega}{V_0} > 1 \\
\text{the value of } V \text{ will be positive at one moment and negative in the next. Consequently, the tool cutting edge will be separated frequently from the chip, which would not happen in conventional cutting. The frequency of the separation is equal to the tool vibration frequency} \\
f = \frac{\omega}{2\pi}. \\
\text{Influence of the Vibration on the Net Cutting Time} \\
The net cutting time } T_s \text{ is defined as the time interval within each cycle of tool}
\end{align*}

each cycle of vibration, as described in the last section. When a separation occurs, there is a discrepancy between the net cutting time } T_s \text{ and time for the vibration cycle } T. \text{ The discrepancy can be determined from the difference between the time } t, \text{ at which the separation ends and the time } t_s, \text{ at which the separation begins. At the beginning of the separation, the relative cutting velocity is } 0, \text{ therefore, } t_s \text{ can be determined from Eq. 3, and is given by}
\begin{align*}
t_s &= \frac{1}{2\pi} \cos^{-1} \left( -\frac{V_0}{2\pi f} \right) \\
r,s \text{ can be determined by considering that at } t \text{ the positions of the tool cutting edge and the chip, which is again in contact with the tool, are equal, i.e.} \\
A \sin \omega t_r &= A \sin \omega t_s - V_0 (t - t_s) \\
\text{where} \\
\omega t_r &= 2\pi \frac{l_t}{f} \text{ and } \omega t_s = 2\pi \frac{l_s}{f} \\
\text{From Eq. 6} \\
\frac{V_0}{f} &= -\frac{2\pi}{t} \cos \left( 2\pi \frac{l_t}{f} \right) \\
\text{Substituting Eq. 8 into Eq. 7 gives}
\sin \left( 2\pi \frac{l_t}{f} \right) - 2\pi \frac{l_s}{f} \cos \left( 2\pi \frac{l_t}{f} \right) = -2\pi \frac{l_s}{f} \cos \left( 2\pi \frac{l_s}{f} \right) \\
\text{Using Eq. 9 } t_s \text{ can be calculated from } t_r \text{ and } T. \text{ The net cutting time during each cycle of the tool vibration is then given by}
\begin{align*}
T_s &= T - (t - t_r) \leq T \\
\text{Eq. 10 shows that in ultrasonic vibration cutting the net cutting time might be less than the time for cutting, depending on whether a separation between the tool and chip occurs, as described in Eq. 4.}
\end{align*}

\begin{align*}
\text{Influence of the Vibration on the Distance of the Tool Sliding over the Machined Workpiece Surface} \\
\text{In conventional cutting, the distance of the tool cutting edge sliding over the machined workpiece surface is} \\
L_{\text{cut}} &= \pi D t \\
\text{where } D \text{ is the diameter of the workpiece, } \pi \text{ is the rotation rate of the workpiece and } t \text{ is the cutting time used. In ultrasonic vibration cutting, due to the tool vibration, an extra sliding distance } A \text{ is introduced. Therefore, the actual sliding distance of the cutting edge in ultrasonic vibration cutting is}
\end{align*}
where

\[ \Delta l = 2Af \]

Eq. 12 indicates that in ultrasonic vibration cutting the actual distance of the tool cutting edge sliding over the machined workpiece surface is longer compared to conventional cutting.

Influence of the Vibration on the Feed Motion

In a cutting process, the relationship between the feed velocity \( u \) (m/s), feed rate \( S \) (mm/min) and the workpiece rotation rate \( n \) (rev/min) is

\[ u = S \cdot n \]  \hspace{1cm} (13)

The relationship between the actual cutting velocity \( V \) and the workpiece rotation rate is

\[ V = \pi \cdot D \cdot n \]  \hspace{1cm} (14)

where \( D \) is the diameter of the workpiece. From Eq. 13 and Eq. 14, the relationship between the feed rate and actual cutting velocity can be determined to be

\[ S = \frac{u \pi D}{V} \]  \hspace{1cm} (15)

For conventional cutting, Eq. 15 is written as

\[ S_0 = \frac{u \pi D}{V_0} \]  \hspace{1cm} (16)

Where \( S_0 \) is the feed rate in cutting without tool vibration. For ultrasonic vibration cutting, substituting Eq. 3 into Eq. 15 gives the feed rate

\[ S_t = \frac{u \pi D}{V_0} \left( 1 + \frac{k_1}{k_2} \cos \omega t \right) \]  \hspace{1cm} (17)

Consider Eq. 13 and 14 in conventional cutting,

\[ u = S_0 \cdot n = S_0 \frac{V_0}{\pi D} \]  \hspace{1cm} (18)

Therefore, the feed rate in ultrasonic vibration cutting can be written as

\[ S_t = \frac{S_0}{1 + \frac{k_1}{k_2} \cos \omega t} \]  \hspace{1cm} (19)

Eq. 19 indicates that in ultrasonic vibration cutting the feed rate varies as a function of the vibration frequency and amplitude. The ratio of \( S_t \) to \( S_0 \) is

\[ \frac{S_t}{S_0} = \frac{1}{1 + \frac{k_1}{k_2} \cos \omega t} \]  \hspace{1cm} (20)

In each tool vibration cycle, the minimum of the ratio is

\[ \left( \frac{S_t}{S_0} \right)_{\text{min}} = \frac{1}{1 + \frac{k_1}{k_2}} < 1 \]  \hspace{1cm} (21)

and the maximum is

\[ \left( \frac{S_t}{S_0} \right)_{\text{max}} = \frac{1}{1 - \frac{k_1}{k_2}} > 1 \]  \hspace{1cm} (22)

That is, in each tool vibration cycle, the feed rate of the vibrated tool, \( S_t \), varies from the minimum which is less than \( S_0 \) to the maximum which is larger than \( S_0 \).

Influence of the Vibration on Tool Crater Wear

Tool crater wear occurs usually by three possible mechanisms: adhesion, abrasion and diffusion. In conventional cutting, the tool rake face and chip material are always in intimate contact during cutting. The direct cause of the crater wear is the chip flow along the tool face at extremely high pressure and temperature. In ultrasonic vibration cutting, the tool rake face and the chip material can be separated. During each of the separations, the crater wear process ceases. Therefore, the wear should be less compared to conventional cutting. Another effect of the separations on the crater wear process is that during each of the separations, cutting fluids or surrounding air can access the tool-chip contact region, which never occurs in conventional cutting at normal cutting conditions. This action provides protection of the tool surface and prevents adhesion between the tool and chip. According to these differences, it can be expected that the tool crater wear in ultrasonic vibration cutting is less compared to conventional cutting. Also, from Eq. 10, since the time interval for each of the tool-chip separations increases with increasing tool vibration amplitude or frequency, it can be expected that the tool crater wear rate decreases with increasing the vibration amplitude or frequency.

Influence of the Vibration on Tool Flank Wear

In conventional cutting, the mechanisms of the tool flank wear are usually adhesion, abrasion, diffusion and fatigue at the tool flank-work interface. In ultrasonic vibration cutting, the wear processes caused by abrasion and fatigue might be intensified by the vibration. Firstly, according to Eq. 12, the distance of the tool cutting edge sliding over the machined workpiece surface can be much longer compared to conventional cutting. This might increase the extent of rubbing from the hard particles contained in the workpiece material on the tool flank face and thus cause more abrasive wear at the tool flank. Secondly, as described in Eq. 19, the actual feed rate varies frequently from the maximum to the minimum during cutting. This variation can create frequent changes in the stresses on the tool flank face, resulting in fatigue wear. Roshchupkin and Radchenko [11] described and experimentally demonstrated the change in stress in the material below a surface as an asperity from a rubbing surface passes over.
the asperity, tensile stresses elongate the material. This change in sign of the stress as an asperity passes a given point can cause fatigue failure of the material below the surface. The wear particles are created by cracks, formed below the surface, spreading and moving up to the surface.

In the preceding analyses, it appears that in ultrasonic vibration cutting the tool vibration can have significant influence on the tool wear performance. It might slow down the tool crater wear process by separating frequently the tool and chip. It might also speed the tool flank wear process by increasing the distance of the tool cutting edge sliding over the machined workpiece surface and creating fatigue stresses at the tool flank.

**EXPERIMENTAL STUDY**

**Experimental Conditions**

The objective of this experimental study was to observe the influence of the tool ultrasonic vibration on the tool wear performance. A conventional lathe machine was used for all the machining tests. The workpiece material used was 0.2\% carbon steel and had a hardness of 140. The tool inserts were made of carbide grade P30. An ultrasonic vibration cutting unit (SONIC IMPULSE SH 150) was used. The tool vibration frequency for the tests was 22 kHz. For all the cutting tests the width of cut was 2 mm and the feed rate taken from the machine was 1 mm/min. The range of cutting speeds used were 0.17, 0.43, 0.65, 1.06, and 1.50 m/s. This speed range is on the low side of cutting speeds in conventional cutting. These lower speeds were used because in ultrasonic vibration cutting, when the tool is vibrated and is frequently separated from the chip, the actual cutting speed will be more than twice the cutting speed. Considering that in ultrasonic vibration cutting separations between the tool and chip can occur, which could facilitate the influence of a cutting fluid on the cutting process, tests for tool wear in conventional and ultrasonic vibration cutting using a water-based coolant were also conducted. For all the tests, the tool crater wear was characterized by measuring the maximum depth of the crater using Taylor Hobson Talysurf 1 profile meter and a chart recorder. The tool flank wear was characterized by measuring the maximum length of the wear land at the tool flank using a Nikon microscope with a graduated eyepiece and a micrometer stage.

**Results and Discussion**

Tool wear on the rake face in both conventional cutting and ultrasonic vibration cutting was observed. When the cutting speed \( \nu \) was 0.67 m/s and the tool vibration amplitude was 25 \( \mu \)m, the tool and chip were frequently separated during cutting. The tool crater wear in ultrasonic vibration cutting was much lower compared to that in conventional cutting, as shown in Fig. 2.

Fig. 3 shows the tool flank wear in cutting with and without tool vibration. The cutting speed was 0.67 m/s. The results corresponding to three vibration conditions: 0.1, 0.3, and 0.5 mm, showed significant differences in tool wear. The tool wear was least at 0.1 mm vibration amplitude. The tool wear was more pronounced at higher vibration amplitudes. The results indicated that ultrasonic vibration cutting is an effective method for reducing tool wear.

![Figure 2: Tool crater wear in conventional cutting and ultrasonic vibration cutting.](image)

![Figure 3: Tool flank wear in conventional cutting and ultrasonic vibration cutting.](image)
1. the tool was not vibrated,
\[
\frac{A_w}{V_0} = 0;
\]
2. the tool was vibrated with amplitude 30 \( \mu m \),
\[
\frac{A_w}{V_0} = \frac{30 \times 22,000 \times 10^{-6}}{0.67} < 1;
\]
3. the tool was vibrated with amplitude 35 \( \mu m \),
\[
\frac{A_w}{V_0} = \frac{35 \times 22,000 \times 10^{-6}}{0.67} > 1.
\]

In the first case, the tool had the minimum flank wear. In the second case, the steady state wear rate was about the same as in the first case, but the wear values were slightly larger. In the third case, the wear values and wear rate were significantly larger compared with the first case. These results indicate that in ultrasonic vibration cutting, the tool flank wear is larger compared with conventional cutting. When the tool vibration does not cause separations between the tool and chip \( (\frac{A_w}{V_0} < 1) \), the wear rate is the same as in conventional cutting. When the tool chip separations occur during cutting \( (\frac{A_w}{V_0} > 1) \), the wear rate is larger compared with conventional cutting.

Figure 4: Influence of a water based coolant on ultrasonic vibration cutting.

Figure 5: Relationship between the cutting speed and the effect of tool vibration on the tool flank wear.

For wear rate are similar to those in dry cutting. However, the wear values in ultrasonic vibration cutting were much larger than those in dry cutting. This indicates that in ultrasonic vibration cutting the effect of the vibration on the tool flank wear is more significant when a water-based coolant is present at the cutting region. A possible reason for this is that the tool vibration permits the coolant more easy access to the tool flank work interface. In conjunction with the high temperature generated by friction between the machined workpiece surface and the vibrated tool flank face, the coolant could accelerate oxidation of the hard particles on the machined workpiece surface. The oxidized particles could be much harder than its original.

Fig. 5 shows a comparison of the variations of the tool flank wear against the cutting speed \( V_0 \) in conventional cutting and ultrasonic vibration cutting. In the conventional cutting cases with the cutting speeds lower than 1 m/s, the wear varying against \( V_0 \) was uncertain. This uncertainty was caused by the tool built-up edge, which was observed during cutting at the speeds 0.17, 0.33 and 0.67 m/s. When the speed was higher than 1 m/s, the built-up edge could not be observed. In cutting at the speeds 1.07 and 1.50 m/s, the wear increased when the speed was increased. It is a common trend that tool wear increases with cutting speed in cutting of steel with a carbide tool at cutting speeds which are high enough to avoid tool built-up edge [12]. In ultrasonic vibration cutting, however, the tool performance was very different. Firstly, it was observed that the tool built-up edge did not occur in all the cases. Secondly, in
shown in Fig. 5. Thirdly, the wear increased with the vibration amplitude, as shown in Fig. 5. It is interesting to see that at each of the vibration amplitudes used, the wear initially increased with cutting speed and then decreased when the speed was further increased. Such as in cutting with the tool vibrated with amplitude 30 µm, at the cutting speed 0.17 m/s, the wear value was increasing with cutting speed. At the higher speed 0.13 m/s, the wear reached its peak value and was decreasing when the cutting speed was increased. As the cutting speed was further increased, the wear values approached those in conventional cutting. These results can be explained using Eq. 1. In the first case, the tool frequently separated from the workpiece but the cutting speed was low. The wear value was, therefore, not large and was increasing with cutting speed. In the second case, the tool was at a higher cutting speed while it frequently separated from the workpiece and there was heavy fatigue stressed. The wear value was therefore the maximum. When the cutting speed was further increased, though the higher temperature caused by the higher speed could increase the tool wear, the wear value should be less compared with those in the first and second cases, because the value of \( \frac{\Delta V}{V_0} \) was less than 1 and was getting smaller and smaller when \( V_0 \) was further increased, which resulted in a shorter distance of tool cutting edge sliding over the machined workpiece surface (see Eq. 12) and less fatigue stresses acting on the tool cutting edge compared to the first and second cases. These results indicate that the effect of the tool vibration on the tool wear varies according to the combination of the cutting speed and the vibration frequency and amplitude. It appears that by choosing suitable combinations of the cutting speed and tool vibration frequency and amplitude, it is possible to have optimum applications of ultrasonic vibration cutting, in which both the best cutting performance and minimum tool wear are achieved.

CONCLUSIONS

Theoretical and experimental studies on the effect of the tool vibration on the tool wear performance in ultrasonic vibration cutting have been presented. It has been found that the tool ultrasonic vibration in cutting reduces the tool crater wear but increases tool flank wear. The influence of tool vibration on tool flank wear varies according to the ratio of the products of the vibration amplitude and frequency to the cutting speed. Applications of a water based coolant in ultrasonic vibration cutting accelerates wear at the tool flank.

References


Ultrasonic vibrations for single point diamond turning applications

\[
EI \frac{\delta^4 w}{\delta x^4} + \rho A \frac{\delta^2 w}{\delta t^2} - \rho l \left( 1 + \frac{E}{\kappa^2 G} \right) \frac{\delta^4 w}{\delta x^2 \delta t^2} + \frac{\rho^2 l}{\kappa^2 G} \frac{\delta^4 w}{\delta t^4} = 0
\]  

[5.19]

where \( G \) is the shear modulus and \( \kappa^2 \) is a dimensionless factor that depends on the shape of the cross-sectional area. The constant \( \kappa^2 \) is called a shear coefficient.

Equation [5.19] is now subject to four initial conditions and four boundary conditions. By assuming a solution the partial differential equation is solvable by the method used earlier.

Which of the two beam models to use is largely dependent on the beam geometry, which modes are of interest, and how many modes are important. If only the first mode is of interest, an Euler-Bernoulli model for this system would be good enough. On the other hand, if the fifth mode is of interest, the Timoshenko model might be worth the extra complexity.

5.2 Longitudinal vibrations

Consider the longitudinal vibration of again an elastic beam of length \( L \), density \( \rho \) and the cross-sectional area \( A(x) \).

Using the coordinate system and the forces on the infinitesimal element as indicated in figure 5.3, one obtains:

\[
F + dF - F = \rho A \frac{\delta^2 w(x,t)}{\delta x^2}
\]

[5.20]

where \( w(x,t) \) is the deflection of the rod in the \( x \) direction and \( F \) denotes the force acting on the infinitesimal element.

From strength of materials yields:

\[
F = E A(x) \frac{\delta w(x,t)}{\delta x}
\]

[5.21]

and substitution into [5.20] and assuming \( A \) constant over \( x \), yields:
Ultrasonic vibrations for single point diamond turning applications

\[
\left( \frac{E}{\rho} \right) \frac{\partial^2 w(x,t)}{\partial x^2} = \frac{\partial^2 w(x,t)}{\partial t^2} \tag{5.22}
\]

The differential equation can be solved in the same manner as described earlier, by assuming a solution. The quantity \( c = \sqrt{E / \rho} \) defines the velocity of propagation of the stress wave in the beam.

For the natural frequency of a free-free beam yields:

\[
\omega_n = \frac{n \pi c}{L} \quad n=1,2,3,\ldots. \tag{5.23}
\]

and for its mode shape:

\[
\cos \left( \frac{n \pi x}{L} \right) \tag{5.24}
\]

### 5.3 Finite element method

The finite element method is a powerful numerical analysis that uses interpolation methods for modeling and solving boundary value problems. The finite element method approximates a structure in two distinct ways. The first approximation made in finite element modeling is to divide the structure up into a number of small elements. Each element's shape is usually very simple which has an equation of motion that can easily be solved or approximated.

The equation of vibration for each individual finite element is then determined and solved. This forms the second level of approximation in the finite element method. The solutions of the element equations are approximated by a linear combination of low-order polynomials.

These solutions, made compatible with the adjacent solution, are then brought together resulting in global mass and stiffness matrices, which describe the vibration of the structure as a whole.

A difficulty with many design and analysis methods is that they work best for systems with a small number of degrees of freedom. Unfortunately, many problems like vibrations have a large number of degrees of freedom. In fact, to obtain accurate results with finite element models, the number of elements and hence the order of vibration should be increased. Note that the exact solution of any beam vibration has an infinite number of natural frequencies, whereas the number of natural frequencies calculated by the finite element method is restricted by the number of elements chosen.

### 5.4 Actuators

If a beam with a certain natural frequency has been designed, it has to be actuated to maintain this frequency and a certain stroke. Because the typical natural frequencies of these structures are in the range of several kHz's, the actuator has to be able to vibrate at the same frequency. In the frequency range above 10 kHz, piezoelectric crystals form the best means of transferring electrical into mechanical energy. That's why piezoelectric actuators are normally used. The drive frequency of the piezoelectric plate is matched with the natural frequency of the tool. In this way, only a small amount of actuator power is necessary to create relatively large strokes. For more information about piezoelectric actuators, see Chapter 9 in Part II.
6 Overview using natural vibrating beams in ultrasonic vibration cutting

Research has proved the fact that it is possible to cut (hardened) steel by applying (elliptical) ultrasonic vibration cutting techniques. As concluded in the foregoing chapters, the theory is applicable. Nevertheless there are some major disadvantages when using beams vibrating in a natural mode. These disadvantages are:

1. Changing the vibrating frequency of the tool head involves changing the complete setup. A set of beams has to be present, which all have different natural frequencies. Therefore it is not possible to estimate the influence of the vibrating frequency on the cutting process.

2. Because the beam is vibrating in its natural frequency the output as displacement as a function of the input (power) is high. Although this is a good feature, any damping exerted on the beam from an external source (like a cutting force), will reduce the displacement drastically.

3. The beam is strained by the PZT at some part of the beam. The actual cutting process takes place at the end of the beam. One could doubt if there is direct control. This has an influence on the amplitude of vibration. Is this amplitude really stable during cutting?

These disadvantages are true for linear vibration cutting but linear vibration cutting has proved itself to be efficient, using longitudinal natural frequencies of beams.

Unfortunately that is not the case for elliptical vibration cutting. Applying natural bending modes of beams to induce tool head vibrations seems to be a very inadequate technique for ultrasonic vibration cutting.

This raises the question if there is another way to vibrate a tool at ultrasonic frequencies. A new technique, where the researcher is able to influence several parameters, admitting him to do actual fundamental research.
Conclusion part I

Developments in the past have led to tolerances on products less than 1 μm and surface roughness less than 0.1 μm. A new technique Single Point Diamond Turning (SPDT) was developed to machine non-ferrous metals, like aluminum and copper. This technique is nowadays widespread among manufacturing plants all over the world.

In the high-tech industry there is a demand for high accuracy products made from steel, and preferably hardened steel. Nowadays manufacturing plants have to grind and polish these products to fulfill the customer wishes. This process is rather time consuming, and it is a craftsmanship being able to produce the products with high tolerances.

In the 80's Kumabe reported that it is possible to machine steel, by applying ultrasonic vibrated cutting tool systems. Several researchers picked up his conclusions and started experimenting with this new technique. The results were remarkable. Tool cutting forces reduced significant in comparison with SPDT and tool life increased with a factor 20. It seemed to be possible to cut the steel and glass family relatively easily. The only problem that arose while cutting hardened steel was chipping of the tool cutting edge.

In 1994 Moriwaki did some research on elliptical vibration cutting. In this process the tool cutting edge is not making a linear movement as in the traditional vibration cutting technique, but it is performing an elliptical pattern in time. Moriwaki reported that once again cutting forces dropped and tool life increased in comparison to traditional vibration cutting.

Nowadays a small group of people has succeeded to cut hardened steel by applying ultrasonic elliptical vibrated cutting tool systems. Though the technique is under control, the tool wear mechanisms and the mechanism of cutting forces are not understood completely. This is partially due to the technique that is being used. The technique is not flexible enough to do fundamental research.

Overlooking this paper, a global insight into the world of single point diamond turning and especially vibration cutting has been given. This insight and knowledge about the specific field is necessary to start the research conducted at the UNCC, as published in Part II.
Part II: Design of an ultrasonic vibrating tool

A research conducted at the University of North Carolina at Charlotte, department of Mechanical Engineering, Center for Precision Metrology, The United States.

Period: October 2001-March 2002
Advisor: Dr. James F. Cuttino
Formulation of the problem

In the last few years several researchers have proved it is possible to cut steel with high tolerances on a diamond turning machine, by applying ultrasonic vibration cutting techniques. Although the success of the new technique, little is understood about the actual cutting mechanism. Most of the researches dealt with the tool design problem, not with understanding the actual cutting process. The main reason is that the ultrasonic tools used were not able to change any parameters during the cutting process, prohibiting fundamental research.

Aim of the research

Develop and test an ultrasonic vibrating tool. The tool has to be able to vibrate at different ultrasonic frequencies, strokes and at different modes. It has to be able to produce a linear and an elliptical vibration mode. After the design stage, experiment with the tool during actual cutting tests. Set the tool up, so it can be used in a follow-up research where the influences of the different parameters will be examined.
7 Introduction

Doing research does not only mean getting actual results. Research means achieving your goals and understanding your results. When a process is understood completely, modifications can be made to improve it. When a process is not understood, improving the concept is a manner of intuitive feeling. Sometimes this turns out right, sometimes wrong.

Therefore it is important to understand ultrasonic vibration cutting. Which parameters influence the process, and if they do, how big is their influence? These questions are unanswered until now.

Being able to estimate the influence of these parameters on the cutting process requires a flexible tool design. Flexible means, changeable in a short amount of time, so large setup times can be avoided. Unfortunately, such a flexible device does not exist at this moment, as can be read in Part I.

This research will search for new ideas to come up with such a flexible tool, breaking barriers to finally understand the fundamentals of ultrasonic vibration cutting.
8 General design constraints and criteria

The design of the ultrasonic vibrating tool has to fulfill certain demands. In the following paragraphs a summation is given of the design constraints and criteria.

8.1 Amplitudes and frequencies

The principle of vibration cutting is based on separation of the tool tip and the chip. As has been shown in Part I the following formula has to be fulfilled all the time:

$$\frac{A \cdot \omega}{V_0} > 1$$  \[8.1\]

If this is not the case, the tool tip will not be separated from the chip and tool wear will still be high. So this is the first and probably the most important constraint.

As can be seen there are three variables in the constraint, the amplitude of the movement $A$, the frequency $\omega$ and the cutting speed $V_0$.

The only variable, which is dependent on the cutting process itself, is the cutting speed. So if the cutting speed is estimated, the ratio between frequency and amplitude can be calculated.

For $V_0$ yields:

$$V_0 = \pi \cdot D \cdot n$$  \[8.2\]

A typical range for cutting speeds in single point diamond turning is from 3 to 50 m/min. Table 8.1 shows the minimal requirements of the product of the amplitude and frequency, for several cutting speeds in this range. Note that the values are the absolute minimum. During actual cutting the product has to be at least 20% higher than the cutting speed.

It is obvious that this cutting speed is dependent on the diameter of the material to be machined. If a constant cutting speed should be obtained, the number of revolutions per second of the main spindle has to be increased with decreasing diameter.

<table>
<thead>
<tr>
<th>Cutting speed (m/min)</th>
<th>Frequency (kHz)</th>
<th>Amplitude (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>20</td>
<td>0.4</td>
</tr>
<tr>
<td>3</td>
<td>40</td>
<td>0.2</td>
</tr>
<tr>
<td>10</td>
<td>20</td>
<td>1.3</td>
</tr>
<tr>
<td>10</td>
<td>40</td>
<td>0.7</td>
</tr>
<tr>
<td>25</td>
<td>20</td>
<td>3.3</td>
</tr>
<tr>
<td>25</td>
<td>40</td>
<td>1.65</td>
</tr>
<tr>
<td>50</td>
<td>20</td>
<td>6.6</td>
</tr>
<tr>
<td>50</td>
<td>40</td>
<td>3.3</td>
</tr>
</tbody>
</table>

If the designed tool is flexible in the whole range of cutting speeds, this implies that the tool should be able to vibrate up to 40 kHz with a stroke of 7 μm. When the tool is running at half of its maximum frequency, namely 20 kHz, the stroke has to be twice as high: 13 μm. The frequency and the stroke should be completely adjustable within this range. The minimum working frequency is equal to 1 kHz.
8.2 Type of vibration

There are two ways to cut the material:

1. Linear
2. Elliptical

Ad 1: The basic movement of vibration cutting is the linear one. In this movement high frequencies can be obtained. In addition to high frequencies, high amplitudes can be obtained by applying boosters. Normally longitudinal vibrations are used.

Ad 2: The elliptical movement consists out of two orthogonal movements with a certain phase shift. Until now only one group of people succeeded cutting metals applying elliptical movement. They used a beam with two orthogonal bending modes at a frequency of 20 kHz and maximum amplitudes of 4 μm.

To determine the effect of linear and elliptical motions, a tool has to be designed which has two features. It should be able to conduct a linear motion only and it should be able to conduct an elliptical movement (2 in 1).

8.3 Stiffness

Obtaining high quality finish is a demand. Otherwise the whole idea of single point diamond turning would be disregarded. An optical surface finish with $R_{max}=0.03 \mu m$ is required.

Cutting forces are at maximum estimated to be 3 N (in the thrust direction), because of the plowing force. This means an average stiffness in the thrust direction of $100e6$ N/m. Note that these values are valid in a static situation only! If considering small, high frequency deviations on top of the low frequency deviations of the wanted surface, stiffness should be at least a magnitude 10 higher.

In the cutting direction forces are less than 1 N. Stiffness can be estimated by defining a minimum of chatter. It is important that the tool doesn’t bend away in the thrust direction due to forces in the cutting direction. Bending stiffness has to be sufficient.

In the feed direction cutting forces are less than 1.5 N. When cutting in a linear mode a deviation of the tool tip in the feed direction of 0.1 μm is allowed. This means a stiffness of $15e6$ N/m.

It is very important to understand that the total loop of the machine spindle towards the tool holder must be very stiff and natural frequencies of the machine must be high.

8.4 Measurement and electronics

One has to be able to measure the influences of diverse parameters on the cutting process. This can be done by roughness measurements after the cutting process but also during the cutting process by measuring the cutting forces. A so-called dynamometer is necessary to measure the forces. The range of the dynamometer has to be from 0.1 N to 5 N. It is commercially available.

It is important to know what kind of path the tool tip performs. The problem is: it is difficult to estimate what will happen during cutting. Maybe the tool tip will be
damped and the tool tip will vibrate with smaller amplitude than intended. It is necessary to measure the amplitude and frequency. Maybe it will even be necessary to control these parameters.

In order to get the vibration going a piezoelectric transducer is necessary. Voltages can be rather high, so specific machinery to control it is necessary.

8.5 Portability
This research has the aim to identify the influence of a certain pair of parameters on the cutting process. Therefore portability of the whole system would be nice but it is certain not a direct demand.

8.6 Overview design criteria
1. The product of amplitude and frequency of the vibrating tool has always to be higher than the actual cutting velocities.
2. The vibrating frequency should be adjustable in the range from 1 to 40 kHz.
3. The stroke should be adjustable from 0 to 13 µm in the range of 20 kHz, and from 0 to 7 µm in the range of 40 kHz.
4. The tool has to be able to produce a linear motion as well as an elliptical motion.
5. The stiffness of the instrument in the thrust direction has to be in the order of 100e6 N/m. Stiffness in the feed direction has to be in the order of 15e6 N/m. Bending stiffness in all directions has to be sufficient to order not to influence the cutting process.
6. Placement, attachment and preloading the PZT has to be optimal. If it is not done right, high voltages will be necessary to reach high amplitudes. High voltages means extra security, and instruments to produce these voltages will be expensive.
7. A so-called dynamometer is necessary to measure the cutting forces. The working range of the dynamometer has to be from 0.1 to 5 N.
8. It is necessary to measure the amplitude and frequency of the tool; maybe it will be necessary to control the tool by a feedback loop.
9 Piezoelectric Technology

In the ultrasonic vibration cutting designs where researchers made use of the natural frequencies of certain bodies, piezoelectric devices were used to provide that body with the necessary energy to keep vibrating at its natural frequency. The piezoelectric device was working at a frequency close to the natural one of the body. So this frequency was fixed.

The design criteria stated that the amplitude and the drive frequency of the tool should be variable. This means changing the voltage and the drive frequency of the piezoelectric device. What does this mean to the appliance of a piezoelectric actuator?

9.1 Introduction

The word “piezo” is derived from the Greek word for pressure. In 1880, Pierre Curie discovered that pressure applied to a quartz crystal creates an electrical charge in the crystal. Later they also verified that an electrical field applied to the crystal would lead to deformation of the material [30]*.

Piezoelectric materials can be used to convert electrical energy into mechanical energy and vice versa. While normally everybody is speaking about piezoelectricity, we should speak about ferroelectricity. The word ferroelectricity can namely be split up into two main groups: Piezoelectricity and electrostriction. These two groups should be distinguished strictly. The piezoelectric effect is defined as a primary electromechanical coupling effect (i.e. the strain is proportional to the electric field), while the electrostriction is a secondary coupling in which the strain is proportional to the square of the electric field [32+33]. Though often confused, it is clear that there is a big difference between the two types of ferroelectric materials. For specific reasons explained later, only piezoelectric materials will be discussed.

Since the ferroelectric effect exhibited by natural materials such as quartz and Rochelle salt is very small, polycrystalline materials such as barium titanate and lead zirconate titanate (PZT) have been developed. These ferroelectric ceramics become piezoelectric when poled. PZT ceramics are the most widely used materials for actuator applications today. At temperatures below the so-called Curie temperature, PZT crystallites exhibit mostly tetragonal structures. Due to their permanent electrical and mechanical asymmetry, these types of unit cells exhibit spontaneous polarization and deformation. Because of the random distribution of these units in the ceramic material, a ferroelectric poling process is required to obtain the associated piezoelectric properties. During poling, an intense electric field is applied to the piezo ceramic. When the field is removed, the electric dipoles stay roughly, but not completely in alignment. The material now has a semi permanent polarization (fig 9.1).

Fig 9.1: Electric dipoles: before and after poling

* [Nr.] Refers to the source used. The number refers to the number in the literature list.
In such ionic (polarized) crystals, when an electric field \( E \) is applied, cations are attracted to the cathode and anions to the anode due to electrostatic interaction. This results in an electric displacement \( D \).

\[
D = \varepsilon \cdot \varepsilon_0 \cdot E
\]  

[9.1]

Here, \( \varepsilon_0 \) is the vacuum permittivity \( (=8.854 \times 10^{-12} \text{ F/m}) \), \( \varepsilon \) is the materials relative permittivity (also called dielectric constant) [30].

The polarization of the ceramic can be disregarded by exceeding the mechanical, electrical and thermal limits of the material. If heated above the Curie temperature, the PZT crystallite unit cells take on a different structure. When cooled, the electric dipoles have again randomized orientations and the material does not regain its macroscopic piezoelectric properties [33].

\[9.2\] Piezoelectric coefficients

Because of the anisotropic nature of PZT ceramics, piezoelectric effects are dependent on direction. To identify directions, the axes 1, 2 and 3 will be introduced (corresponding to the classical x, y and z orthogonal axis set). The axes 4, 5 and 6 identify rotations associated with the axes 1, 2 and 3. The polar, or 3 axis, is taken parallel to the direction of polarization within the ceramic. Piezoelectric coefficients with double subscripts link electrical and mechanical quantities. The first subscript gives the direction of the electrical field associated with the voltage applied. The second subscript gives the direction of the mechanical strain. Several piezoelectric material constants may be written with a superscript, which specifies either, a mechanical or electrical boundary condition [34]. The superscripts are T, E, D and S signifying:

- T= constant stress
- E= constant field
- D= constant electrical displacement
- S= constant strain

As mentioned, piezoelectric materials are characterized by several coefficients:

- \( d_{ij} \): Strain coefficients [m/V] or Charge output coefficients [C/N]; strain developed per unit of electric field strength applied [V/m].
  \[
d_{ij} = \frac{\varepsilon}{E}
\]  

[9.2]

- \( g_{ij} \): Voltage coefficients or Field output coefficients [Vm/N]; open-circuit electric field developed [V/m] per applied mechanical stress [N/m²].
  \[
g_{ij} = \frac{E}{\sigma}
\]  

[9.3]

Or strain developed per applied charge density.

- \( K \): The relative dielectric constant is the ratio of the permittivity of the material, \( \varepsilon \), to the permittivity of free space, \( \varepsilon_0 \).
  \[
K = \frac{\varepsilon}{\varepsilon_0}
\]  

[9.4]

- \( k_{ij} \): Coupling coefficients [\(^2\)]. The coefficients are energy ratios describing the conversion from electrical to mechanical energy. The ratio of energy stored to energy applied is equal to \( k^2 \). The coupling coefficient is only indicating the overall efficiency of a certain ceramic, not of the total piezoelectric element.
There are also some coefficients characterizing the piezoelectric element as a whole, so dependent of material as well as dimensions:

- **C**: Capacitance [F] is calculated by multiplying the relative dielectric constant by the permittivity of free space and the electrode surface area, and then divided by the thickness separating the electrodes.

\[
C = \frac{K \cdot \varepsilon_0 \cdot A}{t}\]  

[9.6]

At frequencies far below resonance, piezoelectric ceramic elements are fundamentally capacitors. Consequently, the voltage coefficients \(g_{ij}\) are related to the charge coefficients \(d_{ij}\) by the dielectric constant \(K\).

\[
d_{ij} = K \cdot \varepsilon_0 \cdot g_{ij}\]  

[9.7]

- **\(\delta\)**: Dielectric loss. The efficiency of a piezoelectric element depends on the mechanical and dielectric loss as well as the coupling coefficient. The dielectric loss is usually the most significant factor and is the ratio of the effective series resistance (real part of the total impedance) to the effective reactance (imaginary part of the total impedance).

\[
\tan(\delta) = \frac{\text{series resistance}}{\text{series reactance}}\]  

[9.8]

Ceramics with a low \(\tan(\delta)\) should be employed for piezoelectric elements which are to be run continuously at high power levels.

- **\(Q_m\)**: The mechanical quality factor is defined as the ratio of the energy supplied per cycle to the energy dissipated per cycle.

Combining all the coefficients, the behavior of a complete piezo actuator can be modeled. The best way to model a piezo actuator seems to be to model it as an equivalent electrical circuit. Using these models it is possible to estimate the performance of the piezo at a certain frequency quite well. But problems arise when trying to model a piezo through its entire frequency range. Different models have to be used for different frequencies. Because this research is not interested in predicting the behavior, but just in monitoring it, these models will not be used.
9.3 Behavior of the piezoelectric actuator

To monitor the output/input of a piezo actuator, several properties of actuators have to be defined [32+33]:

- Voltage ranges: Two main types of piezo actuators are available: low-voltage devices requiring 150 V or less and high-voltage devices requiring about 1000 V for full extension. The maximum operating voltage depends on the ceramic material used and the thickness of the individual layers (parameters like insulation are also important). The high-voltage ceramics are normally called “hard” ceramics. Low-voltage ceramics are called “soft” ceramics. There are some differences: a soft ceramic has high activity in conjunction with high losses. Opposed to this, hard ceramics have comparatively less activity, in association with low loss. Main differences can be seen in the figure 9.2.

- Stroke (displacement): Displacement of PZT ceramics is primarily a function of the applied electric field strength (E), the piezoelectric material used and the length L of the actuator. The displacement of an unloaded single-layer piezo actuator can be estimated by the following equation:
  \[ \Delta L = E \cdot d_f \cdot L_0 \]

For piezo stacks generally a polarity is stated: it defines the polarity of the maximum voltage applicable. This means that certain stacks can be driven in a bipolar operation, improving the stroke and force generation capability of the actuator.

Using PZTs, always keep in mind that the stroke/voltage diagram of a piezo actuator shows a distinct hysteresis. Typical high strain actuator materials show a hysteresis of 10-15%.

- Maximum applicable forces: PZT ceramics can withstand pressures up to a few hundred MPa before it breaks mechanically. This value must not be approached because depolarization occurs at pressures on the order of 25% of the mechanical limit.

- Stiffness: High stiffness values are required to minimize the variation of a stack’s length under varying load. Typically stiffness of low voltage actuators is higher than high voltage actuators. There must be noticed that the stiffness of an actuator differs under different conditions. A PZT element with open electrodes for example, appears to be stiffer than one with shorted electrodes. This is due to the electric charging of the element. The main conclusion is that
Hook's law is not obeyed, so no single stiffness value can reasonably be given. For example, stiffness determined using very low-amplitude displacements is almost twice as high as measured using large-amplitude displacements.

- Force generation: An actuator is only capable of generating force at the cost of the actuator's stroke. Maximum force generation results in zero stroke.

- Resonant frequency: In general the resonant frequency is a function of its stiffness and effective mass. For a relatively long slender actuator, the first resonance mode is in the axial direction of the PZT. For stacks with larger diameters, attention must be paid to the resonance mode of the stack's diameter.

The first axial resonance frequency for an unloaded situation is equal to:

\[ f_0 = \left( \frac{1}{2 \cdot \pi} \right) \sqrt{ \frac{k_e}{m_{eff}} } \quad [9.9] \]

Note: the theoretical value will differ from the practical one. This is due to non-linear spring behavior of the PZT ceramics. When adding a mass M to the actuator the resonant frequency will drop. In addition to this; the resonance frequency can be kept high by improving the stiffness of the element.

The stiffness of the element can be improved by preloading the actuator. Mechanical pre-stressing the actuator is usually done by some kind of spring mechanism. Pre-stressing actuators results in two other advantages than increasing the stiffness:

1. Improvement of the stroke of the actuator. Some materials show remarkable stroke enhancement on mechanical loading, whereas other types are rather insensitive to load variations.
2. Compensation for tensile stress to prevent damage of the actuator.

Piezo ceramics are very sensitive to tensile stresses. Only by applying mechanical preloading can piezo actuators be operated with high dynamics (pulsed operation). Preloading of approximately 10-20% of the specified maximum load of the actuator is optimal.

- Response time: Fast response is one of the desirable features of a piezo actuator. A PZT can reach its nominal displacement in approximately 1/3 of the period of the resonant frequency. Understand that this is a minimal value and it requires an amplifier with sufficient output current and rise time.

- Temperature: piezoelectric actuators require charge and discharge currents that increase with the operating frequency. The thermal active power P generated in the actuator during harmonic excitation (estimation):

\[ P = \frac{\pi}{4} \cdot \tan(\delta) \cdot f \cdot C \cdot U^2_{pp} \]

As can be seen, the power losses increase linear with frequency. This means that in high frequency ranges power losses will be considerable. Also the critical Curie temperature of the ceramic will be reached in a shorter amount of time. The only parameters that can be influenced, to prevent or delay this phenomenon, are the capacitance and the dielectric loss factor; they should be chosen as low as possible. In the next picture the temperature development as function of voltage and frequency can be seen (fig 9.3). The picture is valid for a small (AE0505D16) Tokin America PZT, but it gives an indication that the maximum drive frequency is very limited in AC drive conditions. The maximum drive frequency at 40 V is limited to approximately 5 kHz.
9.4 Consequences for high frequency operation

This research is especially interested in high frequency dynamic operation of piezoelectric actuators. PZTs can provide accelerations of thousands of g’s and are normally perfectly suited for dynamic applications. Because an output of approximately 40 kHz is necessary, this research will encounter some problems [32].

- Dynamic forces can limit the maximum drive frequency of the piezoelectric actuator. Every time the amplifier changes voltage, the piezo element changes its dimensions. Due to the inertia of the PZT mass, a rapid change will generate a force acting on the piezo. In sinusoidal operation with frequency f and amplitude ‘a’, peak forces can be estimated according:

\[ F_{\text{dyn}} = 4 \cdot \pi^2 \cdot m_{\text{eff}} \cdot a \cdot f^2 \]

Attaching a mass rigidly to the rapid moving actuator means creating high dynamic forces in the ceramic. These forces may limit the maximum drive frequency.

- The maximum operating frequency is also limited by the phase and amplitude response of the system. Especially the amplitude response at high frequencies under a certain mechanical lead is important. Are piezo ceramics capable of transforming high frequency electrical input signals into efficient mechanical output?

- The temperature. Suppose that dynamic forces are low and that the amplitude response remains flat until the resonant frequency. Will the temperature stay below the critical Curie temperature? (Probably the amplitude response will have some connection with the dielectric loss factor, namely efficiency. The dielectric loss factor directly influences the dissipated heat, and thus the temperature, in the actuator.)

This overview of piezoelectric technology is important to understand the limits of applying PZTs in high frequency operation. In the next chapter this knowledge of PZT technology will be used to introduce new ideas, how to apply PZTs in ultrasonics.
10 Applying PZTs in ultrasonic technology

10.1 Running PZTs at high frequencies

Piezoelectric elements are available in different forms. There are single sheet PZT, stacks (fig 10.1), PZT tubes etc. Each element has its own characteristics like size, stroke, capacitance, stiffness, resonance frequency, maximum (blocking) force, compression force and price. Our main concern is the relative large stroke and high frequency at which the element has to perform.

It is normally impossible to run any piezoelectric element at 40 kHz with a stroke of 7 micrometers, due to internal heat development. At some point namely the PZT will reach a critical temperature, which never may be exceeded. This critical temperature is reached pretty fast when the PZT is working at maximum amplitude (read maximum voltage). Normally a few kHz can be reached (See figure 9.3, critical temperature=95 °C).

So frequency responses and dynamic forces are not the first concern. The main problem is the excessive heat development in piezoelectric actuators. Solutions have to be found to avoid or control the heat development. Three ways will be discussed.

1. First of all, the PZT can be used to induce a natural frequency in a mechanical element. A PZT operates at approximately the natural frequency of the element, on which it is rigidly attached. The element, which vibrates at its natural frequency, is a booster. It is necessary to multiply the amplitude. So only a small amplitude of the PZT is necessary to create a relative large displacement. A possible construction is depicted in figure 10.2. The converter (nr. 1) transforms the high-frequency electric energy produced by the PZT into mechanical energy. The booster (nr. 3) serves as an amplitude transformer for the required amplitude range. Amplitude magnification is achieved by certain design features or the geometrical shape of the booster. The resonance frequency of the booster has to match the nominal frequency (working frequency) of the generator.
In this particular booster configuration a longitudinal vibration is used. A bending vibration can also be applied. By the use of stepped horns, or better-called, reduced diameter of a beam, fairly big amplitudes at high frequencies can be obtained. The PZT should be mounted on a bending point at a stiff portion of the beam. At this point, strains are relative low, so the power needed from the PZT to induce the vibration can be kept low. In this way high temperatures in the PZT can be avoided.

2. A second method to use a PZT to create high frequency, long stroke movements is probably active cooling of the PZT. Suppose a ring actuator as in picture 10.3. The efficient cooling surface in a ring actuator compared to a stack actuator is high. This means that, if properly designed, heat can be dissipated from the ring in a very effective way. It is possible that temperatures will remain low during high frequency, long stroke usage of the element. In this way the ring actuator can be used directly to create the movement, avoiding complex mechanical element resonating at a certain frequency.

3. A third method is to use several PZTs, and run them separately on the same frequency with a phase difference, which is set by the number of PZTs. If the PZTs would be steered with a sinusoidal wave, as normally, the different PZTs would interfere and a signal containing non-constant amplitudes would result. Though, if the PZTs would be fed with an impulse instead of a sinus, and the rise- and decay times are low enough, the output signals of the PZTs will not interfere. In this way the frequency of the incoming signal will be multiplied by the number of elements, resulting in a higher output frequency. The advantage of this system is that the separate elements can run at lower frequencies, avoiding high temperatures. Note that the amplitude of one element is equal to the amplitude of the total system, disregarding reaction forces on and deflection of other elements.

The first option has already been applied in different researches, as explained in PART I.
The technique, especially the application of the linear ultrasonic cutting technique, has been used successfully to cut different sorts materials. In order to cut elliptical patterns, bending modes in beams were used. As discussed in chapter 8 (Part I), there are some serious critics on this technique. Researchers can only guess what is happening at the actual tool tip. The beam could actually vibrate in a completely different frequency during cutting (due to cutting forces)! Another critic is that a vibrating beam is not flexible in changing variables. The cutting frequency for instance is fixed.
Because this research tries to estimate the influences of different parameters (including the frequency) on the ultrasonic cutting process, the option of designing a vibrating beam is not suitable.

The second option is to actively cool the PZT. This could be quite effective, but there are a lot of problems arising when this solution would be chosen. In the first place, there needs to be a transfer from the heat in the element to another medium. This medium could be air, but probably a closed cooling circuit with another medium like water is necessary. The PZT running at high voltages has to be strictly separated from the cooling fluid. This causes a lot of design problems. The technique is applicable, but it is very complicated.

The third option, running the different elements with impulse inputs, looks very promising.

10.2 Impulse operation of PZTs

As mentioned, the idea is to feed different PZTs with impulses, which are out of phase (Fig 10.4). If the response time of the PZTs is low enough, the output signals of the separate PZTs will not interfere, and a multiplication of the system output frequency will result. The frequency at which the PZTs have to run depends on the number of elements and the desired output frequency of the total system:

\[ f_{\text{drive}} = \frac{f_{\text{out}}}{n} \]  

[10.1]

The phase difference (\( \phi \)) between the impulses is also a function of the number of elements (n):

\[ \phi = \frac{2\cdot\pi}{n} \]  

[10.2]

The main concern in this issue is the response speed of the ferroelectric actuators. The response speed of ferroelectric actuators depends not only on the materials properties, but also on the mechanical resonance frequency of the device and the specification of the power drive. In a frequency range as high as the mechanical resonance frequency, the vibration amplitude is remarkably enhanced. Above the resonant frequency, the strain level is completely suppressed. Therefore, the theoretically fastest response is given by the resonance period. The minimum response time (rise and decay) of a single PZT on an impulse is limited by:
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\[ t_{\text{min}} = 2 \times \frac{1}{f_{\text{res}}} \]  

[10.3]

On the other hand, the maximum reaction time of a single PZT allowed, in order not to interfere with other elements, can be estimated according to:

\[ t_{\text{max}} = \frac{T_{\text{drive}}}{n} = \frac{1}{f_{\text{drive}} \cdot n} = \frac{1}{f_{\text{out}}} \]  

[10.4]

In practice, the minimum response time of an actuator is limited by the combination of the piezo and the power drive. Driving a ceramic actuator corresponds to the injection of charge to a large capacitor. The voltage rises or falls exponentially with a certain time constant. This time constant is called the cut-off frequency \([s]\) of the total system. It is given by the product of the capacitance of the actuator and the impedance of the amplifier [30].

\[ t_{\text{cut-off}} = C \cdot R_{\text{amp}} \]  

[10.5]

\[ f_{\text{cut-off}} = \frac{1}{t_{\text{cut-off}}} = \frac{1}{C \cdot R_{\text{amp}}} \]  

[10.6]

Rule 1: The lower the impedance of the amplifier, the higher the cut-off frequency.
Rule 2: Reduction of the capacitance leads to a higher cut-off frequency. However the required current of the amplifier must be kept in mind.

To reduce the capacitance of the ferroelectric actuator, the relative permittivity must be as low as possible. Applying a ferroelectric actuator means making a choice between piezoelectric and electrostrictive actuators. Electrostrictive materials have higher relative permittivity than piezoelectric materials. This means a higher capacitance, so lower response speed [30].

So piezoelectric materials are in favor regarding to response speed.

The piezoelectric actuator has to fulfill certain demands. As mentioned the resonance frequency has to be as high as possible. Also the capacitance of the piezo has to be as low as possible. The stiffness of each element has to be high. In this way the elements can be used as construction elements. In order to avoid expensive amplifiers the driving voltage must be low. This means that soft piezoelectric materials are in favor compared to hard piezoelectric materials. The main advantage of hard piezoelectric materials is, hysteresis rates are not as high as in soft piezoelectric materials. Thus heat development will be higher in soft piezoelectric materials.
11 Modeling an impulse driven PZT

11.1 Impulse response of a PZT

Because a pulse drives the PZTs, the reaction of the PZT as a change in length is very important. The output of a PZT as a function of its input can be modeled by using the dynamic equation of a longitudinal vibration mode [30]:

$$\rho \left( \frac{\partial^2 u}{\partial t^2} \right) = \left( \frac{1}{s_{33}} \right) \left( \frac{\partial^2 u}{\partial \xi^2} \right) \tag{11.1}$$

The Laplace transform is a powerful tool for treating a transient response to a pulse input:

$$U(s) = L[u(t)] = \int_0^\infty e^{-st} \cdot u(t) \cdot dt \tag{11.2}$$

The Laplace transform of a rectangular input pulse starting at \( t=a \) and ending at \( t=b \) (fig. 11.1):

$$g(s) = \frac{n}{s} \left( e^{-as} - e^{-bs} \right) \tag{11.3}$$

![fig11_1](image)

Fig 11.1: Impulse

The Laplace transformation for the dynamic equation with:

$$u(t = 0, x) = 0 \land \frac{\partial u(t = 0, x)}{\partial t} = 0 \land u(t, x = L/2) = 0 \land u(t, x = 0) = -u(t, x = L)$$

$$\rho \cdot s_{33} = \frac{1}{v^2} \quad (v = \text{sound velocity of the piezo ceramic}) \tag{11.4}$$

leads to the total displacement of the PZT:

$$U(s, x = L) = 2 \cdot d_{33} \cdot E^* \left( \frac{v}{s} \right) \left[ \frac{-e^{-sL}}{1 - e^{-sL}} \right] \left[ \frac{-e^{-sL}}{1 + e^{-sL}} \right] \tag{11.5}$$

where \( E^* \) is equal to:

$$E^* = K \cdot g(s) = \frac{K}{s} \left( e^{-as} - e^{-bs} \right) \tag{11.6}$$

when \( a = 0 \) and:

$$b = \frac{n \cdot L}{v} \tag{11.7}$$

\( b \) is chosen in this way because the resonance period of a piezoelectric actuator is equal to:

$$T_{res} = \frac{2 \cdot L}{v} \tag{11.8}$$

so if \( n \) is equal to 2; the piezoelectric actuator is vibrated exactly in its resonance frequency. With \( a \) and \( b \):
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\[ E' = \frac{K}{s} \left(1 - e^{-n \frac{s L}{v}} \right) \]  \[11.9\]

All the equation necessary to solve the output \( u(t) \) of the PZT as a function of a rectangular pulse with length \( b \), are available. Now let us obtain the displacement of the PZT for \( n=1 \) and \( 2 \).

For \( n=1 \):

\[ U(s, L) = 2 \cdot K \cdot d_{33} \cdot \left(\frac{v}{s^2}\right) \cdot \left[\frac{1 - e^{-\frac{-s L}{v}}}{1 + e^{-\frac{-s L}{v}}}\right]^2 \]  \[11.10\]

The last term in this equation is not directly solvable. To estimate the behavior of the PZT, Taylor expansion series have been made [30]. The inverse Laplace transform is solvable in brackets:

\[ u(t, L) = 2 \cdot K \cdot d_{33} \cdot v \cdot t \]  \[0 < t < \frac{L}{v}\]  \[11.11\]

\[ u(t, L) = 2 \cdot K \cdot d_{33} \cdot v \cdot \left[ t - 3 \left( t - \frac{L}{v}\right) \right] \]  \[\frac{L}{v} < t < 2 \cdot \frac{L}{v}\]  \[11.12\]

\[ u(t, L) = 2 \cdot K \cdot d_{33} \cdot v \cdot \left[ t - 3 \left( t - \frac{L}{v}\right) + 4 \left( t - 2 \cdot \frac{L}{v}\right) \right] \]  \[2 \cdot \frac{L}{v} < t < 3 \cdot \frac{L}{v}\]  \[11.13\]

for \( n=2 \), no Taylor expansion series have to be made:

\[ U(s, L) = 2 \cdot K \cdot d_{33} \cdot \left(\frac{v}{s^2}\right) \cdot \left[1 - 2 \cdot e^{-\frac{-s L}{v}} + e^{-\frac{-2 s L}{v}}\right] \]  \[11.14\]

thus,

\[ u(t, L) = 2 \cdot K \cdot d_{33} \cdot v \cdot t \]  \[0 < t < \frac{L}{v}\]  \[11.15\]

\[ u(t, L) = 2 \cdot K \cdot d_{33} \cdot v \cdot \left[ t - 2 \left( t - \frac{L}{v}\right) \right] \]  \[\frac{L}{v} < t < 2 \cdot \frac{L}{v}\]  \[11.16\]

\[ u(t, L) = 0 \]  \[t > 2 \cdot \frac{L}{v}\]  \[11.17\]
Now it is possible to plot these two responses against time:

![Graphs of impulse responses for different pulse widths](image)

As can be seen, only in the situation where the pulse width \(n=2\) is exactly adjusted to the resonance period of the piezoelectric actuator, the response in infinite time is zero. In all the other situations, so-called “ringing” occurs. Ringing of the PZT is not allowed in this situation because possible interference with other PZTs may occur. From this analysis can be concluded, that problems are expected with respect to interference with other PZTs, when the PZTs are running at a different frequency than the resonant. The analysis as presented above does not include damping factors. In reality, these factors will be present. The damping will play an important role, because it will determine the amount of time the ringing phenomenon will continue to be present in the system. From this point of view, high damping factors are a must, to prevent interference between the different signals.

In this paragraph only the response of the PZT was monitored. In reality the PZT is preloaded by some mechanism. The PZT together with the flexure form a system with a certain natural frequency. This raises the question how a mechanical system will respond to an impulse input.
11.2 Impulse response of mechanical flexure

Let's take the simplest mechanical model to estimate the response to an impulse input. This model is an undamped spring-mass system. The differential equation, which describes this model:

\[ x + \omega_n^2 x = f(t) \]  \[11.18\]

where \( \omega_n \) is equal to the natural frequency of this 1 DOF system. This model can be simulated in Matlab to obtain the output of the system \( x \) as a function of the input \( f(t) \).

To check if the theory described in paragraph 11.1 is valid for any mechanical system, an impulse will be modeled with a width equal to the time period of the resonant frequency.

If:

\[ \omega_n = 2 \cdot \pi \]

\[ T = \frac{2 \cdot \pi}{\omega_n} = 1 \text{ s} \]

So a step input of a length of 1 second should result in no ringing. This statement is checked and it appears to be valid according to figure 11.3.

![Linear Simulation Results](image)

Fig 11.3: adjusted impulse width simulation

This means that a combined system of a certain PZT and a mechanical flexure is expected to behave as figure 11.3, according to the theories described in this chapter.

Because this response cancels out any resonant ringing after an impulse hits the system, the two PZTs are expected not to interfere. In order to run a number of PZTs in series this is an absolute must.

The only way to check the behavior of PZTs in impulse operation is to do some experiments.
12 Experimental approach

12.1 Setup

To check the behavior of impulse driven PZTs, the following demands on the experiment have to be fulfilled:

• The experiment should be able to analyze the behavior of at least two piezoelectric elements.
• Each element (n) has to be driven at a certain frequency, specified by the desired output of the system divided by the number of elements (n).
• The driving mode is a pulse. The pulse width, height and frequency have to be changed during the experiment.
• The output of the system has to be measured. The output of the system consists of two parameters: temperature and displacement in time. The temperature of the piezoelectric element will be measured, preventing burning of the element.
• The displacement in time has to be measured on the fly. Measuring equipment has to be able to detect displacements with amplitudes of 15 $\mu$m and frequencies up to 40 kHz.
• The elements placed in series, have to be preloaded. The load pressure on the elements should be variable.

In order to fulfill these demands the following setup of the experiment is presented:

- Mechanical: The elements have to be mounted on a rigid wall with theoretical infinite stiffness. In order to preload them, some sort of flexural system has to present (fig 12.1):

![Diagram of experimental setup](image)

Fig 12.1: Proposal experimental setup

An elastic flexure and a rigid setting screw preload the elements. The number of elements can be changed quickly, by adjusting the setting screw.

To estimate the dimensions of the flexure, the type of PZTs used has to be known.

As already mentioned the PZT has to fulfill some criteria. A piezoelectric actuator, which satisfies these demands, and was available in stock, is the Tokin America AE0203D08 (fig 10.1, left side). Piezoelectric actuators from
Piezomechanik were also considered. The features seemed to be the same as the Tokin actuators, but they were not available in stock.

<table>
<thead>
<tr>
<th>Type</th>
<th>Dim (mm)</th>
<th>Stroke (µm)</th>
<th>Cap (nF)</th>
<th>Res. Freq (kHz)</th>
<th>Max force (N)</th>
<th>Compr. force (N)</th>
<th>Price ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AE0203D08</td>
<td>2x3x10</td>
<td>9/6</td>
<td>180</td>
<td>138</td>
<td>200</td>
<td>200</td>
<td>21.00</td>
</tr>
</tbody>
</table>

The maximum force generation of the PZT is 200 N (table 12.1). At this force generation the stroke of the piezo is equal to zero. A graph can be drawn with an approximation of the PZT's stroke as a function of the generated force (fig 12.2). In the same plot lines can be drawn, which represent certain flexure stiffness. A flexure for instance with a stiffness of 10e6 N/m, results in a generated force of 65 N and a stroke of 6.5 µm. Note that these numbers are valid when the PZT is running at a 150 V. If the applied voltage is less, the stroke will be less. The dimensions of the flexure can be estimated using an elastic beam deflection formula. For a beam supported at both ends and loaded at the center, the stiffness is estimated by [27]:

\[ S_1 = \frac{48 \cdot E \cdot I}{l^3} \]  \[ [12.1] \]

For a beam fixed at both ends and loaded at the center, the stiffness is represented by:

\[ S_2 = \frac{192 \cdot E \cdot I}{l^3} \]  \[ [12.2] \]

The right representation of the stiffness of the flexure will be in between these two values. This is due to the approximation of the supports of the flexure.

The length (l) and the width (b) of the flexure are known:

\[ l = b = 20 \times 10^{-3} \text{ [m]} \]

For the experiment aluminum will be used with a Young's modulus of 7e10 [N/m²] and a density of 2700 [kg/m³]. This results in a height of the beam of:

\[ h_1 = 2.4 \text{ [mm]} \]

\[ h_2 = 1.5 \text{ [mm]} \]
The setup will be designed with a flexure thickness of 2.5 mm (see appendix F). If this value is too stiff, material can be removed to reduce the thickness.

In the final setup not only the static behavior, but also the dynamic resonance is very important. To estimate the first natural frequency of the designed flexure the following equation is valid [23]:

$$f_1 = \frac{\lambda_1^2}{2 \cdot \pi \cdot L} \sqrt{\frac{E \cdot h^2}{12 \cdot \rho \cdot (1 - \nu^2)}}$$  \[12.3\]

For a clamped-free clamped-free rectangular aluminum plate $\lambda_1^2$ is equal to 22.27. So for the designed flexure the first natural frequency is estimated to be:

$$f_1 = 34 \text{ kHz}$$

For a supported-free supported-free rectangular aluminum plate $\lambda_1^2$ is equal to 9.63. The first natural frequency in this case is equal to:

$$f_1 = 15 \text{ kHz}$$

As with stiffness, it is difficult to estimate which boundary conditions exactly apply. Probably the true natural frequency will be in between these two values. What the influence of this natural frequency will be on the output of the system is unknown.

As mentioned the PZTs have to be preloaded. They will be preloaded at 20% of its maximal force generation. This means the preload force is 40 N.

Suppose one piezoelectric actuator has to be preloaded to the flexure. The resultant stiffness of the two springs in series is:

$$c_{res} = \frac{c_1 \cdot c_2}{c_1 + c_2}$$  \[12.4\]

If the stiffness of the PZT is estimated to be 40e6 [N/m], the resultant stiffness is equal to 8 N/μm. Preloading at 40 N, means a preload stroke of 5 μm.

Because this stroke is very small, a fine thread setting screw has to be used. In the final design an 80 threads/inch setting screw will be used. This is equal to 3.15 threads/mm. To obtain a 5 μm stroke, the setting screw has to be turned approximately 10 degrees.

Electrical: The amplifier should be able to provide impulse inputs to three separate PZTs. So three different output channels are needed. As mentioned, the amplifier has to be able to change pulse width, height and frequency. The range of the different variables in each channel are listed below (table 12.2):

<table>
<thead>
<tr>
<th>Table 12.2: Electrical features</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency range</td>
</tr>
<tr>
<td>1-40 kHz</td>
</tr>
</tbody>
</table>

Another important variable is the current (A), the amplifier has to supply to the PZT. The following equations describe the relationship between amplifier output current, voltage and operating frequency [32]. Note that the relationships are developed for sinusoidal operation, not for pulse drive methods!
Ultrasonic vibrations for single point diamond turning applications

\[ i_{\text{average}} \approx f \cdot C \cdot U_{pp} \]  
\[ i_{\text{max}} \approx f \cdot \pi \cdot C \cdot U_{pp} \]  

For instance, driving a 180 nF PZT at 100 V with a 40 kHz frequency requires approximately 2 A. This number gives an indication of the amount of current the amplifier has to generate.

Another factor is the total resistance of the amplifier. The resistance of the amplifier influences the cut-off frequency of the total system. Therefore the resistance has to be as low as possible (see 10.2).

Electricians at the Center for Precision Metrology will build the amplifier.

Measurement: In the experiment amplitudes up to 15 µm and frequencies of 40 kHz have to be measured. There are only a few possibilities to do this, because the need of a high bandwidth of approximately 100 kHz. The possibilities are:

1. An eddy current sensor (needs to be calibrated)
2. Optical sensor (CTD-sensor, fiber optic displacement sensor)
3. Piezoelectric sensor
4. Strain gauges

In the initial setup a cap gage probe with a bandwidth of 10 kHz and a -3 dB bandwidth of 20 kHz will be used. This will limit the possibility to run the PZTs at high frequency. If the theory works at lower frequencies, position sensors with higher bandwidths can be used.

12.2 Realization single PZT

The first experiments were conducted on a single PZT element (see fig 12.4). Understanding the process of pulsing a single PZT is necessary for interpreting the final experiment, in which a multitude of PZTs will be pulsed.

In the following figure (fig 12.3) the layout of the setup as a black box model will be described:

![Fig 12.3: Black box model of setup](image)

The cap gage-driver setup used in the experiment is a Lion Precision product. The sensitivity of the probe is 1.00 Volts/mil. This is equal to 40 mV/µm. The driver has an analog output, which can be coupled to a digital oscilloscope and a data acquisition board. The data acquisition board used is a PCMCIA type; it was configured on a
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This DAQ board (PCM-DAS16D/16) is developed by Computerboards, Inc. It is a 16-bit board with a maximum sampling frequency of 100 kHz. The readout of the board will be done in LabView. A special program has been written to read the different channels.

![Actual setup](image)

To avoid damaging the PZT due to excessive heat development, the temperature of the PZT is measured. Because the time constants of temperature fluctuations in such a small PZT as used in this experiment are very high, a highly sensitive temperature measurement method has to be used. The maximum temperature will stay below 120°C (250°F), therefore a scanning thermistor thermometer will be used. Because the probe of the thermistor is slightly bigger than the area of the PZT, the measured temperature will be lower than the actual one. Nevertheless, a good picture of the heat development in the PZT will be obtained.

12.3 Measurement and results

The first concern in the experiments is the frequency response of the flexure system. When the single PZT is driven at a certain frequency it can excite the flexure system in a natural frequency. This natural frequency will be dominant in the output signal. The consequence is, the true input signal will be distorted.

To measure the frequency response of the flexure system the impulse response technique has been used. The Fourier Transform Spectra can be constructed of the output signal, indicating the natural frequencies of the signal. The bandwidth of the impulse response measuring system is approximately 30 kHz and the maximum sampling frequency is 125 kHz. The system uses a small hammer as input signal and an accelerometer to measure the output. The following frequency response of the unloaded flexure has been measured:
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The first two dominant unloaded natural frequencies are at approximately 12 kHz and 17 kHz (fig 12.5). This means that two natural frequencies are in the range of 0-40 kHz.

![Unloaded flexure frequency response](image)

**Fig 12.5: Unloaded flexure frequency response**

The measurement above is made in an unloaded situation, so no preloaded piezo was attached to the flexure. It was not possible to make any frequency response plots of the preloaded flexure due to the high axial stiffness of the PZT. The only way to estimate the natural frequencies of the total system (so flexure including preloaded PZT) is, to scan the whole frequency range at constant voltage. Due to the bandwidth of the cap gage, the measurement is limited to 20 kHz. The following frequency response has been measured (fig 12.6):

![Frequency response at 25 Volts](image)

**Fig 12.6: Loaded flexure frequency response**

Due to the higher stiffness of the preloaded flexure the first natural frequency has been shifted from 12 kHz to 15 kHz. Probably the second natural frequency has also
been shifted to higher frequencies, but the cap gage probe is not able to detect this. Notice that the first natural frequency is still in the range of 0-40 kHz.

![Response flexure at 25 Volts, 5 kHz](image)

**Fig 12.7: Response flexure, 25 V, 5 kHz**

The picture above (fig 12.7) shows a response of the flexure on an impulse input in the voltage supplied to the PZT. The frequency of the impulse input is 5 kHz and the “on-time” is approximately 40 microseconds. Between the impulses there is a resonant ringing, the amplitude of the ringing seems to be constant so damping is minimal.

A Fourier analysis of this signal shows that there is a dominant frequency of 15 kHz in the signal. Because the peak in this spectrum is located at exactly the natural frequency of the system, the 15 kHz signal must be equal to the frequency of the resonant ringing. This means that after an impulse hit the system, the system is vibrating at its first natural frequency.

Changing the impulse width does not influence the output of the flexure. The 15 kHz resonant frequency is dominant in the output signal, no matter what the width of the impulse is. This means that the theoretical observations in chapter 11 cannot be confirmed at this moment.

All measurement presented above were done at 25 Volts. Increasing the input voltage does not change the output frequencies but only the amplitude. All the amplitude spectra are therefore valid through a wide voltage range.
12.4 Temperature

Increasing the input voltage does create another problem: temperature. As stated earlier the temperature of the PZT is a function of the applied frequency and the voltage.

![Temperature as function of frequency and voltage](image)

Increasing voltage or frequency leads to higher temperatures according the following graph (12.8):

- Increasing voltage with increasing frequency results in a dramatic increase in temperature.
- Because the voltage is limited the maximum stroke of the flexure is limited. When a broad range of frequencies needs to be tested, the test should be performed at 25 V (see fig 12.8).

12.5 Stroke

At the first resonance frequency the maximum stroke at 25 V is equal to 6 µm. At frequencies besides the resonant one, strokes fluctuate with frequency but are in the order of 2-3 µm (at 25 V). This is clearly too low, looking at the specifications. The solution is to increase voltage. Due to heat development this is impossible.

12.6 Rise times

Rise times of the PZT loaded by an impulse are very low, and are typically in the order of 20 µs.

12.7 Realization double PZTs

As mentioned, the maximum stroke of the flexure is limited because the maximum drive voltage, valid in a large frequency domain, is 25 V. If the technique of using two PZTs, both running at lower frequencies, is applicable; increasing the drive voltage to each separate PZT could increase the maximum stroke.
The critical factor is temperature, so experiments are performed where the output frequency of the total system is compared to the internal heat development of the PZTs.

As can be seen in the picture (fig 12.9), there is a big advantage using two PZTs. The range of drive frequencies can be increased drastically. In the single setup a maximum of 5 kHz could be reached at 50 Volts, now a 10 kHz output can be obtained at the same voltage.

![Temperature PZTs](image)

**Fig 12.9: Temperature in single – and double operation**

Of course the actual output is the most decisive factor. As said, the single setup had a resonant frequency of 15 kHz. This resonant frequency was interfering with the actual output. Driving the same flexure system with two PZTs will disturb the system again.

![Response flexure at 50 V, 3 kHz (double)](image)

**Fig 12.10: Response flexure, 50 V, 3 kHz, double operation**
in its natural frequency. This can be seen in figure 12.10. In between the two pulses there is again a 15 kHz signal present. Fourier analysis has verified this. If the two PZTs are running at 7.5 kHz, they actuate the resonant mode exactly. Because the PZTs are both running at lower frequency compared to the single PZT setup, the heat development is less. Therefore the applied voltage can be increased, resulting in a longer stroke. A maximum stroke of 10 μm has been obtained in resonant mode. One of the objectives of this research is to study the influence of the ultrasonic frequency on the cutting process. The frequency has to be changed continuously. Therefore it is not possible to use the resonant frequency.

Though the rise times of the PZTs look promising, natural frequencies of the total system are disturbing a clear output signal. To avoid this a flexure system has to be developed which has a much higher natural frequency than the first prototype. If for instance a flexure system with a first natural frequency of 50 kHz can be developed, the amplitude of the resonant ringing will be less. In this way, there will still be a resonant ringing in between the pulses, but it will not be so dominant as the 15 kHz signal in the foregoing pictures. This topic will be discussed in the next chapter.
13 Second generation of flexure systems

Designing a flexure system with an extremely high natural frequency and an in comparison fairly low stiffness is a challenge. A new idea will be presented, which will hopefully fulfill the demands.

13.1 Longitudinal vibrations

Traditionally researchers used longitudinal vibrations in beams to excite a tooltip in a linear way. This idea of using longitudinal vibration in a rod is used to develop a new flexure system. In this design the typical high natural frequency of a rod vibrating in its axial direction and the multi layer PZTs are used to create a compact flexure with an extremely high natural frequency (Fig 13.1). All the elastic deformation must be concentrated in the two rods. To accomplish this, an extremely rigid top block is necessary. So compared to the first design, the flexure is not dependent on any bending stiffness.

To use axial stiffness as a flexure the following setup is presented:

![Fig 13.1: Setup second-generation flexure system](image)

The problem that arises is to combine the desired stiffness in the design, because any rod is known to be extremely stiff in its axial direction.

The total stiffness of the flexure is equal to:

\[
\begin{align*}
    k_{rod} &= 2 \cdot k_{rod} \\
    k_{rod} &= \frac{E \cdot A_{rod}}{L} \\
    A_{rod} &= 4.76e-6 \text{ m}^2 \\
    d_{rod} &= 0.78 \text{ mm}
\end{align*}
\]

So suppose the rods are made of steel, the length is equal to two times the length of a single PZT and the desired stiffness is again $10^6$ N/m. This results in a diameter of the rod:

\[
A_{rod} = 4.76e-6 \text{ m}^2 \\
\]

With this value in mind, the dynamic behavior of the flexure will be examined. The following dynamical model is proposed (Fig 13.2):
The first natural frequency of this system is equal to:

\[ \omega_n = \sqrt{\frac{k_{tot}}{M_{eff}}} \]  \hspace{1cm} [13.3]

\[ M_{eff} = M + \frac{2}{3} \cdot M_{rod} \]  \hspace{1cm} [13.4]

Of course, this model can also be represented by:

Blevins [23] proposed a solution for longitudinal vibration of uniform beams with a stiffness \( k \), loaded by a top mass \( M \).

The first natural frequency of this system is equal to:

\[ f_1 = \frac{\lambda_1}{2 \cdot \pi \cdot L} \sqrt{\frac{E}{\rho}} \]  \hspace{1cm} [13.5]

Where \( \lambda_1 \) is the solution of:

\[ \cot(\lambda_1) = \left( \frac{M}{\rho \cdot A \cdot L} \right) \]  \hspace{1cm} [13.6]

If the solution of the first model (for different cases) is compared to the general accepted solution of the second model by Blevins, a maximum deviation of 5% is noticed. This difference is acceptable. So model 1 can be used for calculations, this is convenient because the non-linear cot-function has disappeared.
13.2 Actual design

Now the geometrical variables can be determined if the desired stiffness, the top mass and the desired natural frequency is known. The following procedure can be used (Fig 13.4):  

\[ k_{\text{tot}} = 2 \cdot \frac{E \cdot A_{\text{rod}}}{L} \]

\[ f_i = \frac{1}{2 \cdot \pi} \sqrt{\frac{k_{\text{tot}}}{M_{\text{eff}}}} \]

Fig 13.4: Mathematical design procedure

For different materials (see appendix F) the following table (table 13.1) can be created:

<table>
<thead>
<tr>
<th>Material</th>
<th>a=b [mm]</th>
<th>diameter [mm]</th>
<th>length [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mild steel</td>
<td>0,72</td>
<td>0,81</td>
<td>21,7</td>
</tr>
<tr>
<td>Aluminum</td>
<td>1,24</td>
<td>1,4</td>
<td>21,4</td>
</tr>
<tr>
<td>Titanium</td>
<td>0,97</td>
<td>1,1</td>
<td>19</td>
</tr>
<tr>
<td>Nylon</td>
<td>3,36</td>
<td>3,8</td>
<td>6,8</td>
</tr>
</tbody>
</table>

As already concluded; typical small diameters of the flexure are the result. Because these small values of area influence the bending stiffness of the flexure, the total stiffness of the flexure should be increased. This again means choosing another type of PZT. Tokin America Inc. produces a similar PZT as used in the first experiments. This PZT is capable of producing higher forces at the same displacement and same natural frequency (fig 10.1, right side). The features:

<table>
<thead>
<tr>
<th>Type</th>
<th>Dim (mm)</th>
<th>Stroke (µm)</th>
<th>Cap (nF)</th>
<th>Res. Freq (kHz)</th>
<th>Max. force (N)</th>
<th>Compr. force (N)</th>
<th>Price ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AE0505D08</td>
<td>5x5x10</td>
<td>9/6</td>
<td>750</td>
<td>138</td>
<td>850</td>
<td>850</td>
<td>59.00</td>
</tr>
</tbody>
</table>
The stroke of the PZT can be plotted (Fig 13.5) again as a function of the generated force. At this time a flexure with a stiffness of 30e6 N/m will result in a stroke of approximately 7 μm at 200 N.

The geometrical variables of the flexure can be calculated on basis of the new stiffness (Table 13.3). Note that in these new calculations the top mass is increased because of higher generated forces.

![Graph showing stroke as function of force generation](image)

**Table 13.3:** f = 40 kHz; kₜ₀ = 30e6 N/m; M = 3e-4 kg

<table>
<thead>
<tr>
<th>Material</th>
<th>a = b [mm]</th>
<th>diameter [mm]</th>
<th>length [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mild steel</td>
<td>1.25</td>
<td>1.41</td>
<td>21.7</td>
</tr>
<tr>
<td>Aluminum</td>
<td>2.14</td>
<td>2.4</td>
<td>21.4</td>
</tr>
<tr>
<td>Titanium</td>
<td>1.7</td>
<td>1.9</td>
<td>19</td>
</tr>
<tr>
<td>Nylon</td>
<td>5.9</td>
<td>6.6</td>
<td>6.8</td>
</tr>
</tbody>
</table>

Note: The length in these results didn’t change, because the top mass and the stiffness both were increased by a factor 3.

Note: The length of the flexure is almost equal to twice the length of a PZT for most materials. These results are purely coincidence but very convenient.

Because of the availability and fatigue behavior of steel, it is chosen to be the construction material. Nylon could also be considered, but the length of the flexures should be as long as 7 mm. This short length means several design problems, which can be eliminated by using steel.

When steel is being used, there are two options:

1. Designing a flexure system out of one piece of steel. By wire EDM the square flexures can be produced. Two flexures need to be produced, both with a different stiffness (see appendix G1).
2. Designing a flexure system, which uses "music wires" as a flexure. The thin steel wires should be mounted to a top and end mass. Changing the diameter of the wires means changing the stiffness of the setup.

Because of the flexibility in changing the flexure length, the variability in changing the stiffness quickly, and avoiding complex wire EDM operations, option 2 has been chosen.
In the experiment the natural frequency of the flexure should be flexible. This can be done easily by changing the active length of the wires. Because the length of two PZTs is fixed, the preload surface should be flexible too. This ends up in the following design (fig 13.6 and appendix G2):

![Diagram of final design]

Fig 13.6: Final design

The PZTs are placed between the top mass and the setscrew. The design of the top mass is important, because the weight of it should be as low as possible with a desired infinite stiffness. For the first experiment a rigid piece of aluminum will be taken.

![Actual realization design]

Fig 13.7: actual realization design

If it seems to be that the first natural frequency is still to low, the design can be changed. Applications of tiny U-beams may be a solution for a stiff, low weight top mass.

In picture 13.7 the result can be seen. The biggest PZT of the two is placed between the setscrew and the top mass. The wires are clamped to the big mounting block by a plate, preloaded by 4 screws.
13.3 Measuring techniques

In the first design a cap gage measurement device was utilized to detect the ultrasonic vibrations. The \(-3\, \text{dB}\) bandwidth of this system is equal to 20 kHz. Because the second design is expected to achieve a first natural frequency of 40 kHz, the measurement method has to be upgraded. If the cap gage is used to measure any vibrations with a frequency higher than 20 kHz, the amplitude of the vibrations will not be recorded correctly. The new measurement method is an opto acoustic sensor. The basic principle employed in the Fiber Optic Lever Displacement Transducer comes down to the use of an adjacent pair of fiber optic elements, one to carry light from a remote source to an object or target whose displacement or motion is to be measured and the other to receive the light reflected from the object and carry it back to a remote photo sensitive detector.

Fig. 13.8 depicts the interaction of adjacent transmit and receive fibers as the light is reflected from a target. It can be seen that at zero gap, the light in the transmit fiber would be reflected directly back into itself and little or no light would be transferred to the receive fiber. As the gap increases, some of the reflected light is captured by the receive fiber and carried to the photo-sensitive detector. As the gap increases, a distance will be reached at which a maximum of reflected light is transferred to the receive fiber. Further increases in the gap will result in a decrease in the light at the receiver fiber face and a corresponding drop in the output signal from the photo sensor (fig 13.8).

The gap at which the maximum, or zero slope occurs, provides a convenient and readily usable calibration reference position at which the output signal can be normalized in order to obtain a consistent sensitivity factor relatively independent of the color or finish of the surface of the target or object under measurement.

The device used in these experiments is a product of opto acoustic sensor, Inc. It is an Angstrom resolver series dual channel, model 201. The \(-3\, \text{dB}\) bandwidth of this system depends on the sensor and the signal amplification setting, but the maximum is equal to 5 GHz. In the range of these experiments the \(-3\, \text{dB}\) bandwidth is equal to 100 kHz, 5 times higher than the cap gage.

The analog output voltage of the system is measured with a digital Tektronix oscilloscope. It isn't feasible anymore to measure the output voltage with a data acquisition board installed on a laptop computer, as has been done in the first experiments. The sampling frequency of the DAQ board is limited to 100 kHz on a
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single channel. This means that the DAQ board will not monitor the highest frequencies correctly. It is possible to couple a printer directly to the oscilloscope, but then data processing is not possible. Another solution is to read out the GPIB (General Purpose Instrument Bus) on the laptop. In this way the oscilloscope acts as a measurement device (instead of the DAQ board), and the GPIB interface transfers data (communication device). The disadvantage of this system is that the measurement is never real time. Because there is no control function in this experiment this is not a problem. A National Instruments USB (Universal Serial Bus) controller has been purchased to achieve this communication (fig 13.9). Labview is needed to establish a connection between the instrument and the computer.

13.4 Measurement and results 2nd generation

Two types of flexures are used in the experiments, both having a different stiffness but the same top mass. A flexure with a stiffness of 1e6 N/m, using the steel wires of a diameter of 0.8 mm, and a flexure with a stiffness of 30e6 N/m, using steel wires of diameter 1.4 mm. The top mass in both cases is approximately equal to 9e-4 kg. Note that this is much higher than the mass used in the calculations in paragraph 13.3. This means that the natural frequency of the systems will be lower than the one predicted. Mass can always be removed to increase the natural frequency.

13.4.1 Low stiffness flexure

As mentioned, the flexure with the lowest stiffness of 1e6 N/m, is made of two wires with a diameter of 0.8 mm. One of the biggest problems encountered immediately is how to mount the wires to the top mass. The best strategy is to bend the wires around the top mass. The bend made acts as a restriction, and preloading the flexure creates a stiff fixture. Another problem encountered is the lateral stiffness of the system. Because all the stiffness is concentrated in the longitudinal direction, the lateral stiffness of the non-preloaded flexure is low. All of the lateral stiffness has to be created by the PZT itself. Inclining the PZT results in poor lateral stiffness and this results in a dominant low natural frequency in the longitudinal direction.

The main drawback from these two conclusions is that the design is extremely sensitive to the mounting procedure of the wires and the PZT. It appears that the setup is non-repeatable. This is especially the case when using the small PZT. Therefore the tests of the small PZT will not be presented. Only the big PZT seems to be useful in this experiment. Though these conclusions seem to be discouraging, some actual measurements of a good setup have been made.
For the flexure with a stiffness of 10e6 using a single big AE0505D08 PZT, running at 25 V at 5 kHz the following measurement has been made:

![Image 1](image1.png)

**Fig 13.10: measurements 25 V, 5 kHz**

The first picture in figure 13.10 is the rough signal obtained from the opto acoustic sensor. The second picture is the filtered signal. Filtering has been done in Matlab with a filter with a sharp cut-off frequency of 100 kHz. This cut-off frequency is set equal to the bandwidth of the opto-acoustic sensor in order to get the noise out of the signal. To analyze this signal, a FFT operation has been done on the raw data. The result shows which frequencies are dominant in the output signal (fig. 13.11). Besides the 5 kHz drive frequency, there is a 15 kHz signal present. This is again the first natural frequency of the system. It is clear why the natural frequency is that low; the top mass is too big compared to the stiffness of the flexure. According to the calculations made in paragraph 13.2, the used mass should be 9 times lower to achieve a 40 kHz bandwidth!

There are three options now; increasing the stiffness, reducing the mass or a combination of both. Increasing the stiffness by applying the bigger wires is the easiest solution. The top mass can always be changed later.

### 13.4.2 High stiffness flexure

Replacing the thin wires by wires of a diameter of 1.4 mm increases the total stiffness by a factor 3. The bigger wires seem to bring up another problem. They are so stiff, that they are very hard to bend. It is simply not possible to mount them to the top mass as done in the low stiffness flexure. The wires have to be flattened by a hammer, and then squeezed into the top mass.
The new system has a much higher lateral stiffness, mounting the small PZT into this flexure is fairly easy and repeatable. The problem is that the small PZT does not generate enough force to excite the flexure in a considerable way. It is very hard to get a good picture of the vibration due to the noise in the measurement system. The only practical way to excite the stiffer flexure is by using the bigger PZT’s. The results of the test can be seen in figures 13.12 and 13.13. The test conditions are exactly the same as in the last test. The first remark concerns the maximum stroke. The total stroke is decreased due to the higher stiffness. The FFT analysis shows that besides the drive frequency of 5 kHz, the second dominant frequency is equal to 25 kHz. Due to the increase in stiffness the natural frequency has been increased from 15 to 25 kHz.

From figure 13.12 can be concluded that the amplitude of the “ringing” in the signal is quite big, there is not enough damping in the system.

But the designed 40 kHz is still not achieved, because the top mass is a factor 3 too high. The only dimension in the top mass that can be changed is the height, because the dimension of the PZT fixes the width and the length. Reducing the height of the top mass reduces lateral stiffness. Reducing the height of the top mass has led to a maximum natural frequency of 32 kHz. At that time the mass was approximately equal to 6e-4 kg.

Although a natural frequency of approximately 30 kHz can be reached, there was no overall satisfaction with this type of flexure system. There are a few explicit reasons for this.

1. The experiments are mostly non-repeatable. This is due to the mounting procedure of the PZT.
2. Mounting the wires into the top mass is a critical process, though it can be optimized, the procedure itself is also non-repeatable.
3. The weight of the top mass could not be reduced to the numbers used in the design of the flexure. Especially in the case of the low stiffness flexure the top mass was way too heavy.

4. Lateral stiffness of the flexure is too low. It could never be used in an actual cutting process.

5. The longitudinal stiffness of 30e6 N/m is too high. The big PZT is not capable of generating more effective output in terms of deflection as the small PZT, because its heat development is higher.

Although the design seemed to have failed, a natural frequency of 30 kHz had been reached and some remarkable things were seen during the experiments. Because the first natural frequency is twice as high as in the first design, the bandwidth of the system has increased dramatically. It appeared that if the pulse width is almost equal to the time period of the first natural frequency of the total system, ringing in the system disappears.

### 13.5 Suppressing ringing

Until now there has been paid hardly any attention to the influence of the “on time” of the pulse on the system response. The reason is that the low natural frequency of the first generation was so dominant that it seemed that changing the impulse width did not influence the output signal. In chapter 11, a model was set up to estimate the output for a single PZT for different impulse widths. The major conclusion was that if the impulse width is equal to the resonance period of the PZT, ringing will disappear and a clear output signal will be the result.

In practice, it is simply not possible to pulse a single PZT without any mechanical preload. Therefore, the mechanical preload will influence the resonance behavior of the PZT and vice versa. The basic formula 11.1, which was used to model an impulse driven PZT, is simply a solution for every longitudinally vibrating system. Even when it is preloaded by some flexure system. This has been confirmed by the mechanical analysis presented in paragraph 11.2.

This is an important conclusion, and it reveals that if the PZT is pulsed by a pulse width exactly equal to the resonance period ($T_{res}$, see figure 13.14) of the total system, ringing will not be present in the output signal, no matter the value of the drive frequency (1/$T_{drive}$).
This conclusion has been checked with several experiments and it is valid. For one of the experiments the drive frequency was equal to 11 kHz and the pulse width was approximately 42 µs, this is equal to 24 kHz. The system had a first natural frequency of approximately 25 kHz. The results of this test can be seen in Figure 13.15.

According to the FFT analysis (Figure 13.16), there is only one dominant term, the drive frequency of 11 kHz. The magnitude of the first natural frequency of 25 kHz is much smaller. This confirms the theory that ringing can be suppressed by driving the PZT at a certain frequency with a constant pulse width equal to the time period of the first natural frequency of the system.

This leads to another conclusion; the higher the natural frequency of the system, the smaller the pulse width and this means that the drive frequency can be increased without interference between the adjacent pulses. The only limit to this is the response time of the PZT. If the pulse width becomes too small, there will simply be no output.

Theoretically it should now be possible to manipulate the output in such a way, that the output frequency is equal to the drive frequency for all frequencies up to the natural frequency of the system. Probably this conclusion is correct, but the pulse generator used in these experiments is only capable of generating a maximum pulse width equal to the inverse of twice the drive frequency. The reason is that the pulse generator is designed to activate two PZTs in series. If it would be possible to choose a pulse width bigger than twice the inverse of the drive frequency, the two pulses would overlap. So the maximum output frequency is limited by half of the natural frequency.

Because of the five disadvantages named in paragraph 13.4 and the limiting factor of the natural frequency on the maximum drive frequency, another attempt will be made to design a more applicable system with a higher natural frequency.
14 3rd generation flexure systems

14.1 Introduction
In the first and second-generation flexure systems, the types of flexure designed were of the translational type. In these types of flexure systems, the tool, the flexure mass and all masses attached to the flexure are required to follow the same high acceleration trajectory. Another option is to place the cutting tool at the end of a rotary arm. Using this system, only the cutting tool itself undergoes the highest acceleration while parts of the flexure are closer to the axis of rotation where they contribute less to the overall inertia. A lower overall inertia means a higher first natural frequency. To create this rotational system, a beam has to be designed which has a low torsional stiffness around its longitudinal axis. One of the main problems in this design is how to induce the force generated by the PZT into the beam.

14.2 Torsion of beams: theory
The formulas presented in this section are based on the following assumptions:
1. The bar is straight, of uniform circular section and of homogeneous material.
2. The bar is loaded only by equal and opposite twisting couples, which are applied at its ends in planes normal to its axis.
3. The bar is not stressed beyond the elastic limit.

Looking at the cantilever beam situation in figure 14.1 the twisting angle $\theta$ can be defined as:

\[
\theta = \frac{T \cdot L}{C \cdot G}
\]  

\[
C = \frac{\pi \cdot R^4}{2}
\]  

\[
G = \frac{E}{2 \cdot (1 + \nu)}
\]

The torque on the beam (T) has to be generated by the linearly vibrating PZT. This means that some sort of arm has to be attached to the beam to twist it. The dimensions of the arm are dependent on the choice of PZT and the dimensions of the beam. Imagine a situation as depicted in figure 14.2:
In this case the momentum $T$, exerted on the beam is equal to:

$$ T = F_p \cdot (R + \delta) \quad [14.4] $$

and the stroke ($x$) due to torsion:

$$ x = \sin(\theta) \cdot (R + \delta) \approx \theta \cdot (R + \delta) \quad [14.5] $$

Simple elastic beam equations indicated that the design as proposed in figure 14.2 is not feasible. The lateral stiffness of the beam will always be in the same order of magnitude as the torsional stiffness. This means that a force $F_p$ will not result in pure torsion but in a mix of torsion and lateral displacement. To increase the lateral stiffness of the beam the following setup is proposed (fig 14.3):

Because the lateral stiffness is increased the torsional stiffness will be increased too at constant diameter with respect to figure 14.2. The challenge is to find an optimum design point where a low torsional stiffness is combined with a high lateral stiffness and where the first natural frequency is as high as possible.

The main design parameters are; the length ($L$), the diameter ($D$), the length of the arm ($\delta$) and the material used. Due to fabrication problems, the height of the arm is assumed to be equal to the diameter of the beam. The width of the arm is assumed to be a fixed value, determined by the size of the PZT used. In the modeling of the beam it is assumed that shear can occur over the full length of the beam, so the extension arm included.

For the design in figure 14.3 equation 14.1 changes into:

$$ \theta = \frac{T \cdot (L + \frac{w}{2})}{2 \cdot C \cdot G} \quad [14.6] $$

Substituting formula 14.4 into 14.6 and converting to a torsional stiffness of a beam with length $L_s$: 
In series with this torsional stiffness is the lateral stiffness of the arm; this stiffness can be approximated by a cantilever beam with a length of $\delta$.

$$S_{\text{arm}} = \frac{3 \cdot E \cdot I_{\text{arm}}}{\delta^3} \quad [14.9]$$

The total stiffness at the end of the arm can now be defined as:

$$S_{\text{tot}} = \frac{F_p}{s} = \frac{S_{\text{tors}} \cdot S_{\text{arm}}}{S_{\text{tors}} + S_{\text{arm}}} \quad [14.10]$$

The system has to be designed in such a way that this stiffness is equal to a desired value. Special attention has to be given to the lateral stiffness of the total structure. The lateral stiffness of a beam of length $L_s$ can again be calculated from straightforward elastic beam formulas:

$$S_{\text{latbeam}} = \frac{192 \cdot E \cdot I_{\text{beam}}}{L_s^3} \quad [14.11]$$

Not only the stiffness has to fulfill certain demands, also the dynamical behavior of this system has to be optimal. Looking at the dynamical behavior of this design, two different design approaches can be distinguished. The first one neglects the mass of the extension arm and only models the torsional vibration of shafts. The second approach includes the influence of the arm-mass on the dynamical behavior, and models the beam as torsional springs. A quick calculation with fictive beam dimensions indicates that the first natural frequencies calculated by the second method are several factors lower than the ones calculated by the first method. The two methods overlap when the mass of the arm goes to zero. Because the mass of the arm seems to be critical in the calculation of the first natural frequency of the total system, the second method will be used and explained.

The dynamical model of figure 14.4:

The differential equation of motion for this undamped system becomes:

$$J \cdot \ddot{\theta} = -2 \cdot k \cdot \theta \quad [14.12]$$

The mass moment of inertia ($J$) is equal to the mass moment of inertia of the arm plus one third of the mass moment of inertia of each torsional beam.

$$J = J_{\text{arm}} + \frac{2}{3} \cdot J_{\text{beam}} \quad [14.13]$$

$$J_{\text{beam}} = m_{\text{beam}} \cdot R^2 \quad [14.14]$$

The calculation of the mass moment of inertia of the extension arm is slightly more complicated. To calculate it, the total arm has been divided into two sections. One section (I) with a length equal to $R+\delta$, and a second section (II) with a length of $R$. This yields:
The stiffness of each torsional beam with length \( L + w/2 \) is equal to:

\[
\kappa = \frac{G \cdot C}{L + w/2} = \frac{2 \cdot G \cdot C}{L_s} \tag{14.16}
\]

with the parameters \( G, C \) and \( L_s \) as defined in equations 14.2, 14.3 and 14.7.

From equation 14.12, the first natural frequency is known as:

\[
f_1 = \frac{1}{2 \cdot \pi} \sqrt{\frac{2 \cdot k}{J}} \tag{14.17}
\]

Now it is possible to calculate the total length \( L_s \) from a desired natural frequency. This length can in turn be used to calculate the stiffness of the beam. This has been done in Matlab. Figure 14.5 gives a summary of the mathematical model used.

Fig 14.5: Mathematical design procedure

Special attention has to be paid to other vibrational modes that can be excited. If the beam gets too long and slender, a lateral natural bending mode can occur, which is lower than the desired natural frequency. A second point of concern is the length of the arm. Again, if the arm gets too long and slender the cantilever beam may vibrate at a natural frequency, which is lower than the desired one. These first natural frequencies can be calculated from easy to apply formulas provided by Blevins [23].

The first lateral natural frequency of a clamped-clamped beam loaded by a center mass can be approximated by:

\[
f_1 = \frac{4}{\pi} \sqrt{\frac{3 \cdot E \cdot I_{beam}}{L_s^3 \cdot (m + 0.37 \cdot m_b)}} \tag{14.18}
\]

where \( m \) is equal to the mass of the arm of length \( \delta \) and \( m_b \) is equal to the mass of the total beam of length \( L_s \).
The first lateral natural frequency of a cantilever beam can be estimated by:

\[ f_l = \frac{\lambda_1^2}{2 \pi \cdot \delta^2} \sqrt{\frac{E \cdot I_{arm}}{p \cdot A_{arm}}} \]  

[14.19]

where \( \lambda_1 \) is equal to 1.875.

14.3 Torsion of beams: Simulation results

The model as described in paragraph 14.2 can now be used to design a torsional flexure system. The dimensions and parameters in table 14.1 are necessary input into the model:

The dimensions of the arm are chosen in such a way that any of the two types of PZTs can be used to vibrate the flexure system.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Mild steel</td>
</tr>
<tr>
<td>( \delta ) [mm]</td>
<td>4</td>
</tr>
<tr>
<td>w [mm]</td>
<td>3</td>
</tr>
<tr>
<td>h [mm]</td>
<td>D</td>
</tr>
<tr>
<td>( S_{tot} ) (N/m)</td>
<td>10e6</td>
</tr>
<tr>
<td>( f_1 ) (kHz)</td>
<td>40</td>
</tr>
</tbody>
</table>

Table 14.1: Model inputs

The calculation results in two major outputs as can be seen in figure 14.5, namely the diameter of the beam and the length of it.

To find the dimensions of the flexure where the desired natural frequency and the desired stiffness are attained, two plots have been created (fig 14.6). In the first plot the total stiffness (\( S_{tot} \)) as function of the diameter has been calculated, and in the second plot the length of the beam (\( L_0 \)) as function of the diameter has been plotted. Of course these two plots are interrelated because the stiffness is dependent on the length of the beam. This implicates that the y-values have to be determined by using the same frequency line in both plots.

One can notice that for a system with an infinite high natural frequency the length goes to zero and the diameter goes to infinity. This means that there should be some point where the flexure has specific dimensions, so it still has high manufacturability.
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on a standard lathe. From figure 14.6 can be concluded that a system with a first natural frequency of 40 kHz and a stiffness of 10e6 N/m has such dimensions. Mostly for all other cases, there will be a conflict between the width of the arm and the total length of the flexure.

Table 14.2 summarizes all data available of this design from the calculations as presented in paragraph 14.2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Equation</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Stiffness</td>
<td>14.10</td>
<td>N/m</td>
<td>10e6</td>
</tr>
<tr>
<td>First natural freq.</td>
<td>14.17</td>
<td>kHz</td>
<td>40</td>
</tr>
<tr>
<td>Diameter</td>
<td></td>
<td>mm</td>
<td>3.2</td>
</tr>
<tr>
<td>Total length</td>
<td></td>
<td>mm</td>
<td>9</td>
</tr>
<tr>
<td>Lateral Stiffness beam</td>
<td>14.11</td>
<td>N/m</td>
<td>287e6</td>
</tr>
<tr>
<td>Lateral Stiffness arm</td>
<td>14.9</td>
<td>N/m</td>
<td>80.6e6</td>
</tr>
<tr>
<td>Torsional Stiffness</td>
<td>14.8</td>
<td>N/m</td>
<td>11.8e6</td>
</tr>
<tr>
<td>Lateral 1st natural freq. beam</td>
<td>14.18</td>
<td>kHz</td>
<td>120</td>
</tr>
<tr>
<td>Lateral 1st natural freq. arm</td>
<td>14.19</td>
<td>kHz</td>
<td>167</td>
</tr>
</tbody>
</table>

The actual design has been machined and is depicted in figure 14.7 and appendix H. It can be used on the base of the 2nd generation design.

Looking at table 14.2 the following remarks concerning the design can be made:

1. The lateral stiffness of the beam is approximately 30 times higher than the desired stiffness of 10e6 N/m.
2. The lateral stiffness of the arm is approximately 8 times higher than the desired torsional stiffness of 10e6 N/m.
3. Remarks 1 and 2 indicate that the torsional stiffness of approximately 12e6 is absolutely the lowest one in the system.
4. The 1st lateral natural frequency of both the beam and the arm are at least 3 times higher than the expected torsional natural frequency of 40 kHz.
14.4 Measurements and results 3rd generation

14.4.1 Choice of PZTs

Again two types of PZT can be used in these experiments. The small PZT has the advantage of a small mounting surface, so no additional elements are needed to actuate the system (as shown in figure 14.7). The big PZT should have an extra element on top of the PZT to transfer the generated force to a point load. That element introduces extra mass to the system and drops the natural frequency of the system drastically. If no extra element is placed on top of the big PZT it appears that its maximized output is not any higher than the small PZT. Although the big PZT generates more force, it also heats up faster. If the maximized displacement is defined as a certain stroke at a certain frequency at which the PZT heats up to 80 °C, it appears that the maximized displacement for both the PZTs are the same. This can be seen in figure 14.8. Because the smallest PZT has the lowest moveable mass and capacitance, that one should be used preferably.

![Maximum displacement 2 PZTs](image)

Fig 14.8: maximized displacement as function of drive frequency

14.4.2 Frequency response

To determine the first natural frequency of this system the same sort of tests as used in the last 2 chapters are applied. When the system is hit by a pulse of any width (as long as it is not equal to the time period of the natural frequency of the system) ringing in the system will occur. When the drive frequency is below the first natural frequency, resonance will be dominant in the signal. The frequency of this resonant ringing is equal to the first natural frequency, and can be determined by an FFT analysis of the output signal.

In the following experiments a single small AE0203D08 PZT is used to actuate the flexure. Setting up the PZT in the flexure is a process with a high repeatability. This is due to the parallelism of the two surfaces the PZT is placed in between. Note that the measurements are conducted at 50V at 5 kHz instead of 25 V at 5 kHz as in the last two chapters.
The results of the test can be seen in figure 14.9. Again it is clear that there is a resonant ringing present throughout the whole signal. The frequency of this resonant ringing can again be estimated by looking at the FFT analysis of the raw data. Figure 14.10 shows the FFT analysis. Because 5 kHz is equal to the drive frequency, the second dominant peak in the plot has to be equal to the resonant ringing frequency. This indicates that approximately 50 kHz is the first natural frequency of the system. Note that this frequency is higher than the designed 40 kHz. It is possible that the model used is not correct but another obvious reason is the increased stiffness of the flexure due to the preloading of the PZT.

These results look extremely promising, but keep in mind that the magnitude of the resonant ringing is still considerable.

14.4.3 Stiffness
The first remark that has to be made regards the stiffness of the system. Theoretically it is possible to build a system with an infinitely high natural frequency, but it stiffness will be infinitely high too! So to compare the different flexures that have been developed until now, stiffness should always be kept in mind. This raises the problem how to measure the stiffness. The stiffness could be measured by placing a loadcell between the setscrew and the moveable mass, or by loading the moveable mass by gravity. Both methods bring up some problems.

Using a loadcell to measure the stiffness is simply not possible because there are no loadcells available, which are so small that they can fit in between the setscrew and the moveable mass.

Using masses to measure the stiffness could be applied, but note that a stiffness of 10e6 N/m requires 10 kg to excite it 1 μm! This means that the flexure has to be loaded by at least 20 kg to get only 2 data points in an acceptable range. A complete new experiment has to be designed to do it, and the time limit did not allow this.
The easiest way to compare the performance in terms of output of the different flexures is looking at the peak-to-peak deviation at a constant voltage and frequency. This method does not create absolute values of stiffness, but it makes it possible to compare the different flexures (see table 14.3).

<table>
<thead>
<tr>
<th>Type of flexure</th>
<th>Displacement (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2nd generation, wire system, big PZT</td>
<td>1</td>
</tr>
<tr>
<td>3rd generation, torsional system, small PZT</td>
<td>0.9</td>
</tr>
</tbody>
</table>

From table 14.3 can be concluded that the output of the third generation is approximately the same as the second generation. So compared to the second generation the third generation is favorable because its natural frequency is approximately 1.7 times higher. In this review the first generation is not discussed because the measurement method was different, instead of the opto acoustic sensor the cap gage was used.

### 14.5 Optimizing the output signal

As can be seen in paragraph 13.5, it is possible to actuate the system in such a way that the only dominant term in the output signal is the drive frequency itself. If the width of a single pulse is chosen to be equal to the time period of the first natural frequency of the structure, ringing will disappear.

Because the first natural frequency of the torsional flexure is equal to 50 kHz, a pulse width of approximately 20 µs should be used to excite the flexure.

In figure 14.11 the results are shown of a single small PZT loaded by a 50 V, 5 kHz pulse signal. The pulse width is equal to 25 µs.

The result is an output, which consists of clear pulses of a frequency equal to the drive frequency. The reason that the pulse width is slightly higher than expected probably has to do with the non-perfect shape of the pulse. The width of the pulse (T_{meas}) is measured from the two intersections with the x-axis.

Because the input signal, a pulse, has a certain rise and decay time, an average rectangular pulse would have a smaller pulse width of T_{eq} (see figure 14.12).
One of the main worries in impulse operation of PZTs, and especially with decreasing impulse width, is the response time of the system. If the rise time is too long, the PZT cannot generate its maximum force. When half of the impulse width is smaller than the rise time of the PZT, the optimal stroke will be aborted. The pulse is cancelled before the system has reached its maximum displacement. This results in a smaller stroke.

Paragraph 9.3 concluded that the rise time of a PZT is approximately equal to 1/3 of its resonant frequency. Let's assume that this statement is also true for PZT systems. The nominal resonant frequency of the system is equal to 50 kHz. So the expected rise time:

\[ t_{\text{rise}} = \frac{1}{3 \cdot 50e^3} = 6.7 \mu s \]

The nominal "on time" of half the pulse is equal to:

\[ t_{\text{allowed}} = \frac{1}{2 \cdot 50e^3} = 10 \mu s \]

It appears that the rise time of the system is fast enough. So in the adjusted pulse width case the maximum stroke is obtained at any drive frequency.

The maximum drive frequency is limited by the width of the pulse, as mentioned in paragraph 13.5. Note that this is a result of the design of the pulse generator, and it could be changed any time. For this design it results in a maximum drive frequency of 25 kHz in the pulse width adjusted case.

So up to 25 kHz, the signal can be adjusted in such a way that an almost exact pulse pattern, with a frequency equal to the drive frequency, will result. If an output frequency higher than 25 kHz should be obtained, there are two possibilities.

1. Apply the technique as proposed in paragraph 10.2. Put two PZTs in series, and run them both on a specific frequency. The two PZTs together generate an output frequency twice as high as the drive frequency. Because the PZTs can both run at a lower frequency, they can generate more force than one single PZT could do at the same output frequency.

2. Rebuild the pulse generator so the pulse width can be adjusted to be equal to the inverse of the drive frequency. In that way a single PZT can be run up to 50 kHz in pulse adjusted mode. The disadvantage of this idea is the performance of the PZT at high frequencies.

Because the advantages of point 1 are already clear from former sections like paragraph 10.2, the next paragraph will discuss the details of this technique applied to the torsional flexure system.
Another question is the maximum stroke as function of frequency. This question is already answered in figure 14.8, which was used to indicate the difference between the small and the big PZT. In figure 14.13 this figure can be seen again, but now the maximum drive voltage at a certain frequency is included. The maximum drive frequency in the plot is 20 kHz, above this value the output of the system is so low that measuring it becomes impossible due to the noise of the opto acoustic sensor. The maximum peak-to-peak displacement is equal to approximately 5 μm at 80 V. The minimum value is measured at 20 kHz and is equal to 1 μm at 18 Volts. At both these conditions the maximum temperature of the PZT was 80 °C. There is no information available about reliability of the PZT at this temperature.

14.6 Impulse adjusted input mode with two PZTs in series
In paragraph 10.2 and 12.7 the idea of running two PZTs in series was explained. In this paragraph that idea is combined with the idea presented in paragraph 13.5, adjusting the pulse width of the input signal to avoid ringing. It seems that the observation that ringing disappears when the pulse width is adjusted correctly, perfectly suits the theory of two PZTs in series. The appliance of two PZTs in series requires that the signals of the separate PZTs do not interfere. This requirement is fulfilled by adjusting the pulse width to be equal to the inverse of the natural frequency of the system.
In figure 14.14 the results of a test can be seen, in which two PZTs, both running at 5 kHz at 25 V, generate an output signal of 10 kHz. Because the pulse width has been fit to cancel out ringing, the output signals of each individual PZT do not interfere with each other. An almost perfect output signal is the result, “almost” because some slight ringing in between the pulses can be monitored. The ringing phenomena can be explained by looking at the mass a single PZT has to move. If a pulse hits PZT 2 (see fig. 14.14), it only has to accelerate the mass of the flexure, but when PZT 1 is activated, it has to move the mass of the flexure and the mass of PZT 2. So when PZT 2 hits the system it will have a higher natural frequency compared to when PZT 1 hits it. If a pulse generator can be designed with a separate variable pulse width on every channel, each PZT can be adjusted independently to cancel out ringing.

There also seems to be an irregularity in the maximum output voltage of each PZT. As can be observed, the maximum outputs of the PZTs are slightly different. It appears that PZT nr. 2, generates a more effective force than PZT nr. 1, the one attached to the setting screw. Interchanging the PZTs does not change this behavior; it is inherent to the setup. The explanation for this behavior should again be sought in the amount of moveable mass. PZT 1 has to move more mass than PZT 2. Because the amount of mass is higher, the acceleration is lower, so in a certain time interval PZT 1 generates less stroke than PZT 2. Generating more force in that time interval can compensate this; PZT 1 should be run at a higher voltage than PZT 2. So the solution is to make the drive voltage variable and independent on each channel. Because of the change in setup, the first natural frequency of 50 kHz cannot be achieved. It is not possible anymore to speak about one single natural frequency of the system, because of the irregularity in active moving masses. The most critical natural frequency, though, is still the lowest one, and that one appears when PZT 1 is activated. Note that also the natural frequency when PZT 2 is activated is lowered, this is due to the lower overall stiffness, a result of the introduction of PZT 1 to the system.

It appears that the natural frequency has dropped from 50 kHz to 35 kHz. The maximum drive frequency of a single PZT is limited to approximately 17 kHz, so the maximum output frequency is equal to the resonance frequency; 35 kHz. From figure 14.13 can be seen that the peak-to-peak displacement will be equal to approximately 1 μm at least.
14.7 Suggestions to optimize the current design

The current design can be pulsed in two different ways, both with their own performances:

1. Single PZT mode;
   a. Maximum drive frequency: 25 kHz
   b. Minimum displacement at maximum drive frequency: \( \approx 0.7 \mu m \)

2. Double PZT mode;
   a. Maximum drive frequency: 17 kHz
   b. Maximum output frequency: 35 kHz
   c. Minimum displacement at maximum output frequency: \( \approx 1.2 \mu m \)

Referring to paragraph 8.1 this yields for the cutting speed at the highest frequency:

\[
V_{1,\text{max}} = A \cdot 2 \cdot \pi \cdot f_{\text{max}} \approx 3.5 \text{ m/min} \quad [14.20]
\]
\[
V_{2,\text{max}} = A \cdot 2 \cdot \pi \cdot f_{\text{max}} \approx 8 \text{ m/min} \quad [14.21]
\]

In paragraph 8.1 the minimum cutting speed in diamond turning was defined to be equal to 3 m/min. The maximum cutting speed is 50 m/min. So it seems that both systems fulfill the requirements to do ultrasonic vibration cutting. But the maximum cutting speed is very low. Besides the low cutting speed in actual cutting experiments the output of the system is still not optimal (see paragraph 14.6). What can be done to optimize the output of this pulse width adjusted torsional system, in order to make it more applicable in ultrasonic vibration cutting?

1. Increase the natural frequency of the torsional system. This will lead to a higher bandwidth, so a higher maximum drive frequency of the system. A higher natural frequency might be more damped, so that the amplitude of ringing in the system (if present) will be lower. The main reason to increase the natural frequency is the extra mass on the system introduced by the diamond-cutting tool. All the analyses until now have been performed without an extra mass. The extra mass is unavoidable when actual cutting should be done. So increasing the natural frequency by changing the design should compensate the extra mass on the system. Increasing the natural frequency might be achieved by the following ideas:
   a. Use of materials with the same Young’s modulus, but a lower density. A look at appendix D shows that the use of carbon fibers is a good opportunity.
   b. Using hollow shafts and other type of extension arms. The idea is to create the same stiffness with less inertia or mass. Take a look at the rotational shaft. The torsional constant \( C_s \) is a function of the radius to the power four. This means that the most effective material is located at the outer side of the shaft. Taking material away from the center hardly influences the stiffness. But it does influence the moveable mass, so the torsional natural frequency. The same idea is valid for the extension arm. A rigid arm like used in this design is stiff, but also heavy. Other structures should be reviewed to use them to create a stiff but low extension arm. A simple solution would be to hollow the extension arm.
   c. There might be a better configuration of dimensions than used in this design. The model used to design this flexure seems to be reasonable, but has not been checked for several different dimensions. It only
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seems to approximate one situation. If the model is correct, this is the most optimal design. If it is not correct, there might be another flexure with other dimensions and a higher natural frequency at the same stiffness.

2. Redesign the pulse generator and amplifier electronics. When using two PZTs in series to actuate the flexure the effective output of each PZT is different, as concluded in paragraph 14.6. This means that the amplitude and impulse “on time” should be adjusted independently of each other. At the same time when redesigning the electronics, the impulse width should not be limited by twice the drive frequency, as is the case now. The maximum pulse width has to be equal to the inverse of the drive frequency. In this way it is virtually possible to create a constant voltage signal.

Note that if the natural frequency of the system is increased, the adjusted pulse width becomes smaller and smaller. This requires faster rise times of the pulse generator, resulting in a more perfect pulse.

3. The output in terms of stroke is still not sufficient. To obtain a bigger stroke there are mainly two solutions:

a. Decrease the stiffness of the flexure with constant natural frequency. The solution is the same as proposed under point nr. 1. Instead of increasing the natural frequency the same techniques can be used to achieve a lower stiffness with constant high natural frequency. Of course there has to be some trade off between the two design specifications. In an optimal design, the new flexure has a lower stiffness and a higher natural frequency. In that way a mass in the form of a cutting tool can be added without ruining the output signal and a high stroke can be obtained due to the lower stiffness.

b. Active cooling of the PZT. At this moment the heat development in the PZT is the limiting factor to increase the stroke at the present design. If a technique can be applied which conducts energy from the PZT to another media, the input voltage can be increased at a certain drive frequency, resulting in more force generation, so a bigger stroke. As already mentioned in paragraph 10.1, it is not easy to actively cool a PZT. Using a ring actuator is the best solution, but using that type of PZT requires an element on top of the PZT to transfer the generated force into the flexure. Such an element again has mass, reducing the natural frequency of the system.
14.8 Future work and uncertainties

Unfortunately, the flexure as it is now, cannot be applied in an ultrasonic vibration cutting process. That's why no actual cutting experiments have been conducted. Though, a new technique has been presented, which can be used to do fundamental research in the ultrasonic vibration-cutting field. The technique makes it possible to change frequency and amplitude, within their limits, during cutting.

In paragraph 14.7 several propositions have been made to actually use the technology to do cutting experiments. Looking at the number of propositions and their complexity, it is clear that the research is still in the first, exploration phase. At some time a decision has to be made if the research is going to be continued. What are the uncertainties that can influence this decision?

First of all, it is not certain if the desired amplitudes at higher drive frequencies can be obtained. For further researches this is going to be the biggest challenge. Combining a higher output of the system, with a higher natural frequency is the ultimate goal. The extra mass introduced to the system by the diamond tool is going to be a critical factor regarding frequency response.

Secondly, it must be understood, that it is still unclear if the impulse output of the signal really reduces tool wear. All former ultrasonic vibration systems were using sinusoidal waves to reduce tool wear. It is possible that the huge accelerations due to the impulses even worse the tool wear. The huge accelerations of the structure could increase chipping. The chipping is a result of impact forces, when the tool enters the material.

A third point of interest is the reliability of the system. The PZTs are pushed to the end at this moment. Operating temperatures up to 80°C are normal. For how long do they last in a setup, and is their output as force generation stable over time?

Over viewing this report leads to the conclusion that some new concepts have been developed regarding ultrasonic vibration cutting. Though the technique is not directly applicable in cutting experiments, a good basis has been provided for further research and developing.

The technique of double in series PZT operation, together with an impulse width adjusted input signal and a torsional flexure seems to be very promising to reveal the many questions still unanswered regarding ultrasonic vibration cutting.
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Conclusion part II

The main goal was to build an ultrasonically vibrating tool which stroke, frequency and mode can be changed within a certain range. The first assumption that has been made concerns the vibrating mode. This research has been focusing on linear vibrations only. Once the knowledge of linear vibration cutting has been obtained, the technique can be broadened to use it in elliptical vibration cutting.

In order to vibrate a flexure at any given frequency with a certain stroke, a flexure has to be designed with a low stiffness and an extremely high natural frequency (bandwidth>40 kHz). When the first natural frequency of the flexure is high enough, the amplitude of the resonant frequency in the output signal will be low so the output is approximately equal to the input. Three different sorts of flexures have been designed. It appeared that the third design, a torsional flexure is the one with the best stiffness-natural frequency ratio. Dependent on the setup, but bandwidths up to 50 kHz have been reached.

The second problem is the control of the flexure. Piezoelectric devices can be used to generate a force at high frequencies, except that normally the force generation at high frequencies is not big enough to excite a flexure. A possible solution is to use several PZTs in series, running at a certain frequency with a pulse as input signal. If the output signals of the separate PZTs do not interfere, a multiplication (dependent on the number of PZTs in series) of the drive frequency results. Because each PZT is running at a lower frequency than the actual output frequency, the heat development in the PZTs is lower. A higher output frequency and/or higher stroke is the result.

The third point of interest is the response of a certain flexure to an impulse input. The impulse input disturbs a very broad range of frequencies. This results in resonant ringing in between two impulses in the output signal. The frequency of this ringing is equal to the first natural frequency of the system. It appears, from theory and experiments that a pulse width adjusted to be equal to the time period of the first natural frequency of that flexure suppresses ringing almost completely. This theory can be used to generate an almost perfect output signal.

Combining the three ideas presented above; a flexure with a high natural frequency, pulsed PZTs in series and a pulse width adjusted input signal results in a system, which has a variable output frequency and adjustable stroke. The system designed has a maximum output frequency of 35 kHz. The minimal output stroke at that value is equal to approximately 1.2 μm.

No actual cutting experiments have been done, because the output signal of the system is not optimal. Some suggestions have been proposed to improve the output signal, so the tool can be used to actually perform cutting tests and derive relationships between cutting frequency and stroke on the one hand, and tool wear and cutting forces on the other hand.
Ultrasonic vibrations for single point diamond turning applications

Literature


Ultrasonic vibrations for single point diamond turning applications


Ultrasonic vibrations for single point diamond turning applications

[34] Piezo systems, *Online introduction to piezoelectricity*, www.piezo.com


**Companies visited:**
2. IPT fraunhofer institute, C. Bertalan and J. Hennig, Steinbachstrasse, 17 D-52074, Aachen, Germany.
Internet resources

Ultrasonic vibration cutting research published on the Internet:

2. http://www.lboro.ac.uk/departments/me/DYNMCS/page7.htm#Vibration
3. http://ima.berkeley.edu/research/index.html; research
5. http://www.mech.ubc.ca/~mal/hpages/shamoto.html; research

Corporations conducting research on the cutting process:


Corporations developing machines


Piezoelectric technology

Appendix A: Design of an ultrasonically linear vibrated tool

Fig. 3. Illustration of a newly fabricated ultrasonic vibration cutting tool system with high rigidity (1: bolted Langevin-type transducer, 2: ultrasonic vibration generator, 3: tool shank, 4: cutting insert, 5: tool holder, 6: tool fixing blocks, 7: clamping bolts, 8: tool post).
Appendix B: Cutting experiments: a comparison

(a) Cutting forces and cutting distances.

(b) Surface roughness and cutting distances.

Figure 6: Life of diamond tools. Workpiece: Hardened die steel (JIS: SUS420J2), HRC39. Feed rate: 10 μm/rev. Depth of cut: 10 μm. Cutting speed: 2.5 m/min. Tool: nose radius of 1 mm, rake angle of 0° and relief angle of 15°. Vibration: circular locus with radius of 3.5 μm, or linear locus with amplitude of 3.5 μm and at 20.4 kHz.
Appendix C: Understanding the forming of an ellipse

phase difference $1/16 \pi$

phase difference $1/8 \pi$

phase difference $1/4 \pi$

phase difference $1/2 \pi$

Principle of elliptical vibration cutting

\[
\begin{align*}
x(t) &= a \cos(\omega t) \\
y(t) &= b \cos(\omega t + \theta) \\
z(t) &= c t \\
r(t) &= x(t) + y(t) + z(t)
\end{align*}
\]
### Appendix D: Material properties

<table>
<thead>
<tr>
<th>Material</th>
<th>Elastic Modulus [GPa]</th>
<th>Density [Kg/m$^3$]</th>
<th>Poisson ratio</th>
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<tr>
<td>Aluminum 6061-T6</td>
<td>73</td>
<td>2700</td>
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<tr>
<td>Steel AISI C1020</td>
<td>210</td>
<td>7850</td>
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<tr>
<td>Titanium B120VCA</td>
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<td>4850</td>
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<tr>
<td>Thornel T-300C pan-based fiber</td>
<td>231</td>
<td>1760</td>
<td>0.3</td>
</tr>
<tr>
<td>Nylon 6/6 extruded</td>
<td>3</td>
<td>1134</td>
<td>0.3</td>
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## Appendix E: List of symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>ω</td>
<td>Angular frequency</td>
<td>rad/s</td>
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<tr>
<td>ρ</td>
<td>Density</td>
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<tr>
<td>ε</td>
<td>Permittivity</td>
<td>F/m</td>
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<tr>
<td>δ</td>
<td>Dielectric loss</td>
<td></td>
</tr>
<tr>
<td>φ</td>
<td>Phase difference</td>
<td>rad</td>
</tr>
<tr>
<td>ν</td>
<td>Poisson ratio</td>
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<tr>
<td>a</td>
<td>Amplitude</td>
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<tr>
<td>A</td>
<td>Area</td>
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<tr>
<td>C</td>
<td>Capacitance</td>
<td>F</td>
</tr>
<tr>
<td>D</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>D</td>
<td>Electric displacement</td>
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<tr>
<td>d</td>
<td>Strain coefficient</td>
<td>m/V</td>
</tr>
<tr>
<td>d</td>
<td>Damping factor</td>
<td>kg/s</td>
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<tr>
<td>E</td>
<td>Elastic modulus</td>
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<tr>
<td>F</td>
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<tr>
<td>f</td>
<td>Frequency</td>
<td>Hz</td>
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<tr>
<td>G</td>
<td>Shear modulus</td>
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<tr>
<td>g</td>
<td>Voltage coefficient</td>
<td>V m/N</td>
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<tr>
<td>I</td>
<td>Moment of inertia</td>
<td>m⁴</td>
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<tr>
<td>k</td>
<td>Relative permittivity</td>
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<tr>
<td>k</td>
<td>Coupling coefficient</td>
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<td>k</td>
<td>Stiffness</td>
<td>N/m</td>
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<tr>
<td>V</td>
<td>Cutting speed</td>
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<tr>
<td>v</td>
<td>Velocity</td>
<td>m/s</td>
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Appendix F: 1st generation flexure

Material: Aluminum 6061

Cameron applied research center
Ultrasonic vibration cutting

10-25-2001  Piezo mounting tool
<table>
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<th>Dim±0.1 mm</th>
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</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ultrasonic vibration cutting</td>
</tr>
</tbody>
</table>

| 10-25-2001 | Piezo preloading screw           |

\[
1/4" \times 80 = 80 \text{ threads/inch}
\]

\[
1/4" \times 80 = 3.15 \text{ threads/mm}
\]
<table>
<thead>
<tr>
<th>Dim. ±0.1 mm</th>
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Appendix G1: 2\textsuperscript{nd} generation flexure EDM operation

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<td></td>
<td>Ultrasonic vibration cutting</td>
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<p>| 12-17-2001 | Piezo AE0505D08 flexure          |</p>
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<th>Material</th>
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<tbody>
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Appendix G2: 2nd generation flexure wire system

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<table>
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<tbody>
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<td>Cameron applied research center</td>
</tr>
<tr>
<td>Ultrasonic vibration cutting</td>
</tr>
</tbody>
</table>

| 12-19-2001 | Piezo mounting block |

1/4\"×80
| Dim ±0.1 mm | Material: Mild steel |
| Dim. in mm | Cameron applied research center Ultrasonic vibration cutting |
| 12-19-2001 | Wire fixing plate |
Appendix H: 3rd generation flexure

| Dim±0.1 mm | Material: Mild Steel |
| Dim. in mm | Cameron applied research center Ultrasonic vibration cutting |
| 02-15-2002 | Torsion tool |